ANKARA YILDIRIM BEYAZIT UNIVERSITY

GRADUATE SCHOOL OF NATURAL AND APPLIED SCIENCES

INVESTIGATION OF THE PERFORMANCE OF SHORT FIBER REINFORCED PLASTIC GEARS AT DIFFERENT TEMPERATURES

M.Sc. Thesis By

Yasin AYDIN

Department of Mechanical Engineering

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ANKARA

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Yasin AYDIN

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M. Sc. THESIS EXAMINATION RESULT FORM

We have read the thesis entitled **" INVESTIGATION OF THE PERFORMANCE OF SHORT FIBER REINFORCED PLASTIC GEARS AT DIFFERENT TEMPERATURES"** completed by **Yasin Aydın** under supervision of **Prof. Dr. Selahattin Çelik** and we certify that in our opinion it is fully adequate, in scope and in quality, as a thesis for the degree of Master of Science.

Prof. Dr. Selahattin ÇELİK

Supervisor

Prof. Dr. Serkan TOROS

Jury Member

Assoc. Prof. Dr. Barış KALAYCIOĞLU

Jury Member

Prof. Dr. Sadettin ORHAN

Director

Graduate School of Natural and Applied Sciences

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Name & Surname: Yasin AYDIN

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August, 2024 **Yasin AYDIN**

INVESTIGATION OF THE PERFORMANCE OF SHORT FIBER REINFORCED PLASTIC GEARS AT DIFFERENT TEMPERATURES

ABSTRACT

In this study, polymer gears performances are improved with the help of high mechanical properties materials namely fibers. Short fibers which are made from varied materials and have different length and diameters to be added in POM and PA 66 polymers that are widely used polymers in gear technology.

Widely used polymer materials which are POM (polyacetal) and PA (polyamide) 66 are used as baseline studies. Polymer gears are studied theoretically by using VDI 2736 and analyzed in ABAQUS and KISSsoft. Due to VDI 2736 is not accepted as international standard, results need to compare with FEM. Short fibers have added, and elastic properties of polymers are obtained by using Digimat software. Representative Volume Element (RVE) is created via Digimat and can be used to analyze in analysis program namely ABAQUS which is most used program for composite materials.

Finally, short fiber reinforced plastic that show high mechanical properties are used and gear optimization studies (tip relief and profile shift coefficient) have been conducted to reduce tooth root bending stress and improve gear life.

In the last section, results and obtained values have been discussed. Thesis aims, results, and future works are mentioned.

Keywords; Plastic gears, fibers, Digimat, mechanical and thermal behavior, composite materials, SFRP, root bending stress analysis.

KISA ELYAF TAKVİYELİ PLASTİK DİŞLİLERİN FARKLI SICAKLIKLARDA PERFORMANSLARININ İNCELENMESİ

ÖZ

Bu çalışmada polimer dişlilerin performansları, fiber olarak adlandırılan mekanik özellikleri yüksek malzemeler yardımıyla artırılmaya çalışılmaktadır. Dişli teknolojisinde yaygın olarak kullanılan polimerler olan POM ve PA 66 polimerlerine farklı malzemelerden yapılmış, farklı uzunluk ve çaplarda kısa elyaflar eklenmiştir.

Temel çalışma olarak POM (Polyoxymethylene) ve PA (poliamid) 66 polimer malzemeleri kullanılmıştır. Polimer dişliler VDI 2736 kullanılarak teorik olarak çalışılmış ve ABAQUS ve KISSsoft'ta analiz edilmiştir. VDI 2736 uluslararası kabul görmüş bir standart olmadığı için, sonuçlar sonlu elemanlar analizi ile karşılaştırılmıştır. Kısa elyaflar eklenmiş ve Digimat yazılımı kullanılarak polimerlerin elastik özellikleri elde edilmiştir. Temsilci Hacim Elemanı (RVE) Digimat'tan alınarak kompozit malzemelerin analizleri için yaygın olarak kullanılan ABAQUS analiz programında analizler yapılmıştır.

Son olarak yüksek mekanik özellikler gösteren kısa elyaf takviyeli plastik dişli kullanılarak optimizasyon (diş üstü talaş alma, profil kaydırma) çalışmaları yapılmıştır. Modifiye edilmiş düz dişliler analiz edilmiştir.

Son bölümde referans dişli ile güçlendirilmiş malzeme kullanılan ve tasarımsal optimizasyon uygulanan dişli arasındaki farklar, elde edilen sonuçlar, gelecekte yapılması muhtemel işler bahsedilmiştir.

Anahtar Kelimeler; Plastik dişliler, fiber malzemeler, Digimat, mekanik ve ısıl davranışlar, kompozit malzemeler, diş kökü gerilme analizi.

CONTENTS

NOMENCLATURE

Roman Letter Symbols

℃	Celsius Degree
b	tooth width
d_p	pitch diameter
E	Elastic Modulus
E_{avg}	Average Elastic Modulus
E_{PACF30}	Average Elastic Modulus of PA66-CF-30
E_{PACF45}	Average Elastic Modulus of PA66-CF-45
E_{PACF60}	Average Elastic Modulus of PA66-CF-60%
E_{PAGF30}	Average Elastic Modulus of PA66-GF-30%
E_{PAGF45}	Average Elastic Modulus of PA66-GF-45%
E_{PAGF60}	Average Elastic Modulus of PA66-GF-60%
$E_{POMGF30}$	Average Elastic Modulus of POM-GF-30%
$E_{POMGF45}$	Average Elastic Modulus of POM-GF-45%
$E_{POMGF60}$	Average Elastic Modulus of POM-GF-60%
E_{11}	Elastic Modulus in x-direction
E_{22}	Elastic Modulus in y-direction
E_{33}	Elastic Modulus in z-direction
E_{12}, E_{21}	Shear Modulus in xy-plane
E_{13}, E_{31}	Elastic Modulus in xz-direction
E_{23}, E_{32}	Elastic Modulus in yz-direction
F_t	Tangential Load

g	gram
j	Gear Backlash
K	Kelvin
K_F	factor for tooth root load.
L	Contact length
m	Gear Module
MPa	Megapascal
Nm	Newton-meter
T	Torque
$p_{c(max)}$	Maximum Contact Stress
T_g	Glass Transition Temperature
T_m	Melting Temperature
x^*	Profile Shift Coefficient
Y_{Fa}	form factor,
Y_{Sa}	stress correction factor,
y_m	Material Factor
y_p	Tooth Shape Factor
z_p	Number of teeth of pinion
z_g	Number of teeth of gear

Greek Letter Symbols

α	Pressure Angle
σ_F	Tooth Bending Stress
σ_{Flim}	Fatigue Limit

α_w	Working pressure angle
Δa	The amount of addendum modification
ε_α	transverse contact ratio
μm^3	Micrometer
ϑ	Poisson Ratio
Y_ε	contact ratio factor,
Y_β	helix angle factor,

Acronyms

3D	Three Dimensional
AGMA	American Gear Manufacturers Association
CAE	Computer Aided Engineering
CFRP	Carbon Fiber Reinforced Polymer
DIN	Deutsches Institut für Normung
FEM	Finite Elements Method
HPSTC	Highest Point of Single Tooth Contact
PA	Polyacetal
PA-CF-30%	30% volume fraction carbon reinforcement PA 66
PA-CF-45%	45% volume fraction carbon reinforcement PA 66
PA-CF-60%	60% volume fraction carbon reinforcement PA 66
PA-GF-30%	30% volume fraction glass reinforcement PA 66
PA-GF-45%	45% volume fraction glass reinforcement PA 66
PA-GF-60%	60% volume fraction glass reinforcement PA 66
PEEK	Polyetheretherketone

PMC	Polymer Matrix Composite
POM	Polyoxymethylene
POM-CF-30%	30% Volume fraction carbon reinforcement POM
POM-CF-45%	45% Volume fraction carbon reinforcement POM
POM-CF-60%	60% Volume fraction carbon reinforcement POM
POM-GF-30%	30% volume fraction glass reinforcement POM
POM-GF-45%	45% volume fraction glass reinforcement POM
POM-GF-60%	60% volume fraction glass reinforcement POM
RVE	Representative Volume Element
S-N	Stress vs. Number of cycles
SFRP	Small Fiber Reinforced Plastic
VDI	Verein Deutscher Ingenieure

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CHAPTER 1

INTRODUCTION

With the increasing usage of polymer gears in gear technologies such as actuators, kitchen, office equipment, medical and automotive industries, so on, scientists and engineers consider the development of plastic gears. Plastic gears have many advantages which are mentioned as follows [1]:

- Low mass and inertia
- High corrosion resistance,
- No lubrication needed,
- Sound and vibration damping,
- Lower cost (mass production),
- Design freedom.

However, plastic gears have also many disadvantages such as low mechanical properties and performances, low thermal properties, poor manufacturing tolerances, and moisture absorption [1].

Widely used plastic gears are classified by gear transmission engineers and scientist as shown in Figure 1.1 below:

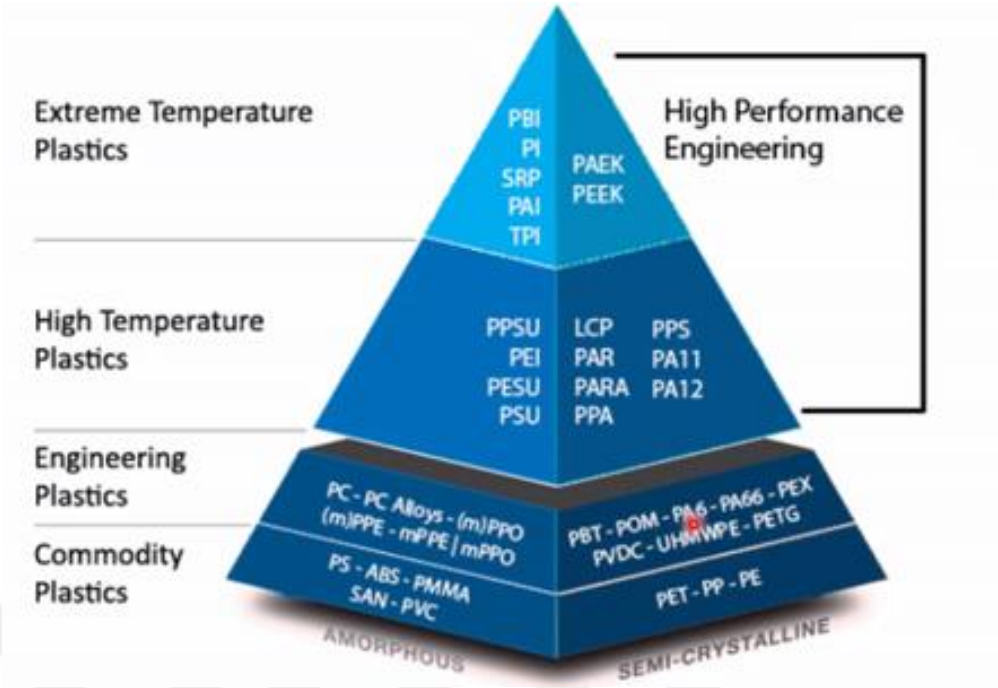


Figure 1.1 Classification of Plastic [2]

Engineers and scientists' study on failure modes for plastic gears. Thermal failures, fatigue failures, wear and deformation are examples of failures. Hence, besides the mechanical studies of plastic gear, wear and deformation types are also investigated in literature. Also, design and assemblies of plastic gears are examined. According to literature review, scientists and engineers try to improve the plastic gears load and performance capacity via development of materials & design, manufacturing, and assembly processes. Figure 1.2 below illustrates the classification of development processes for plastic gears.

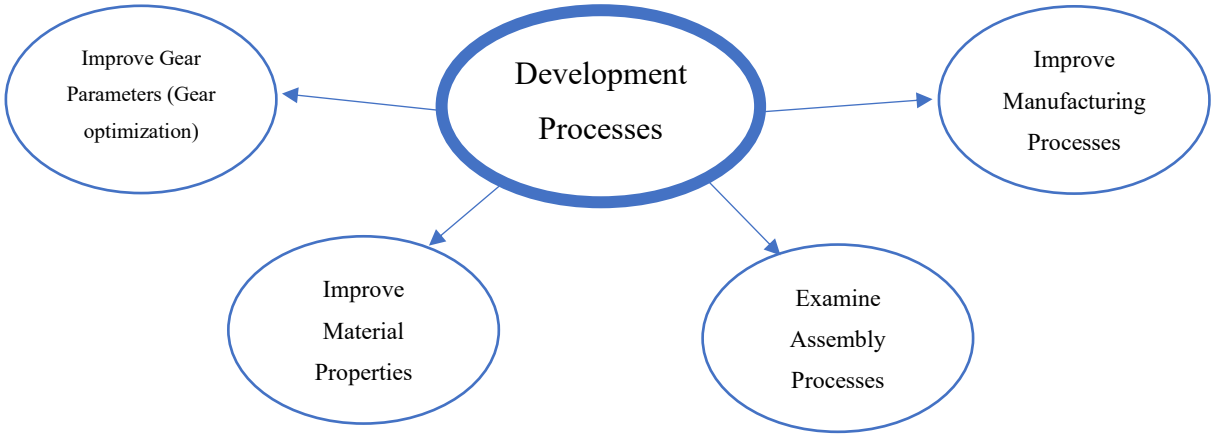


Figure 1.2 Classification of Plastic Gears Development Processes

For example, Zorko et. al [3], studied the effect of center distance error on service life of plastic gears. In this study, gears are meshed by using different center distance (CD) values. 3 different center distances are selected considering assembly and manufacturing errors. Also, by using VDI 2736, theoretical calculations were conducted. However, VDI 2736 does not include center distance error (nominal stress is constant for three cases). According to test results, when CD is increased nominal root stress also increases because of transverse contact ratio is increased. CD does not directly affect temperature, because of increasing of backlash then temperature will decrease. Also, comparison is made to show actual root stress differences between theoretical and experimental results. As a result, VDI 2736 shows higher root stress values than experimental values around 20 %.

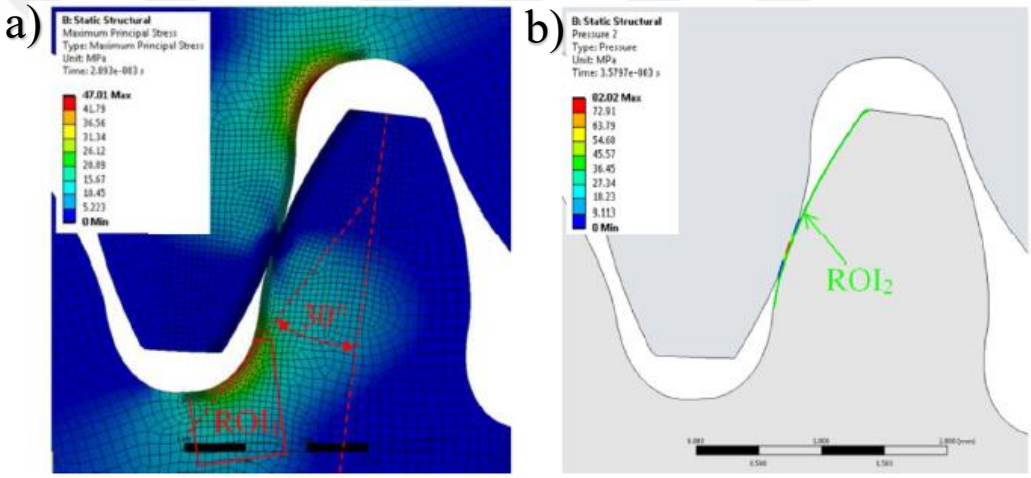


Figure 1.3 3D Stress Analysis of Plastic Gears a) Tooth Root Stress b) Flank Pressure [3]

The effect of backlash is shown in following figure. Figure 1.4a shows when gears are meshed in theoretical center distance and Figure 1.4b shows when center of gears are higher than theoretical value. Backlash has an important effect on gear design. In this study, center distance is increased as backlash increases. As gears are rotated, air is flowing between two gears. Due to high flow of air, the gears are cooled. This helps the service life of plastic gears.

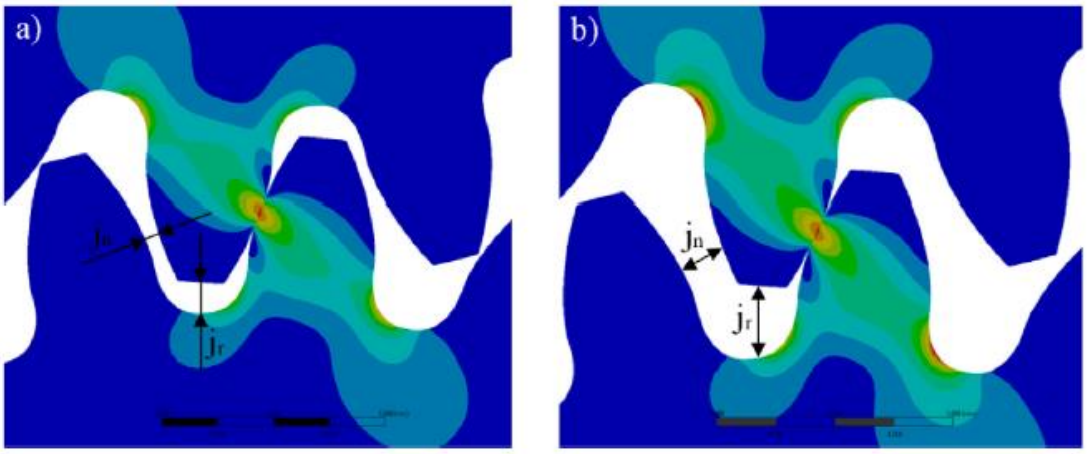


Figure 1.4 a) Theoretical Center Distance b) Higher Center Distance [3]

There is a comparison study conducted between VDI 2736 and JIS B 1759 standards [4]. This study shows that differences between standards for the bending load capacity of plastic gears. According to formulas which are mentioned in study, VDI 2736 can be a good standard for bending stress calculations. Because load factors can be determined, and this study helps to provide contact ratio factor. In this thesis, tooth fillet factor and rim thickness factor cannot be considered. Only effect of material types and mechanical properties are calculated [4,5].

In the other study [6], the effect of fiber materials on damping is examined for Nylon 6/6 plastic material which contains carbon fiber and glass fiber materials. Three materials are assessed separately. According to test results, the incorporation of glass and carbon fibers into the Nylon 6/6 material decreases the damping capacity of the composite. Reduction of the viscous component due to the reinforcement causes a reduction of material resilience. Gears mechanical properties are improved, and sound characteristics are also increased [7,8].

Zorko et al. [9] are studied on teeth profiles of plastic gears for improvement of mechanical behavior. Involute gear and S-gear are analyzed and assessed. Involute gear and S-gear are illustrated below. Figure 1.5a illustrates involute gear which is most used for gear. In this type, line of action of gears are straight and Figure 1.5b shows S-gear. When gears are rotated line of action is not straight. The theoretical tooth flank geometry of S-gears provides more rolling and less sliding between the matching flanks, compared to involute gears [9].

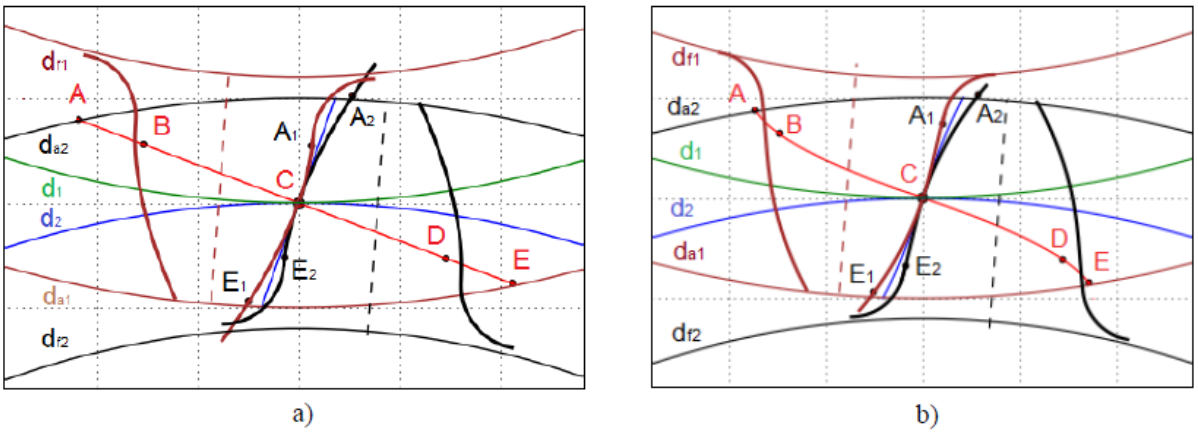


Figure 1.5 Gear profiles a) Involute Gear b) S-gear [9]

The results show that load carrier capacity of S-gear is higher than involute gear. S-gears show better mechanical performance because S-gear tooth profile improves contact conditions. Contact conditions depend on line of action, sliding velocity, so on. S-gears show lower contact stress, flash temperature and related heat generation. Hence, S-gears can carry high loads.

Düzcükoğlu [10] studied on tooth width modification. When gears transmit higher loads with large revolutions, the most common damage is thermal damage that is caused by heat on gear surface. The study aims to increasing contact area with using gear width. In theory, gears have forced each other with two loads which are Single tooth contact and double tooth contact. In single tooth contact region, load is transmitted at only one point. Hence, in this region, loads are higher. Düzcükoğlu tried to develop new gear model to reduce stress in the single tooth contact region. Single tooth contact region and double tooth contact region are shown in Figure 1.6. To reduce stress levels on gear flank and root, width of gear teeth is increased by a factor of two in single tooth contact region which is illustrated in Figure 1.7. Figure 1.7 a shows an unmodified gear view and Figure 1.7 b shows modified gear view.

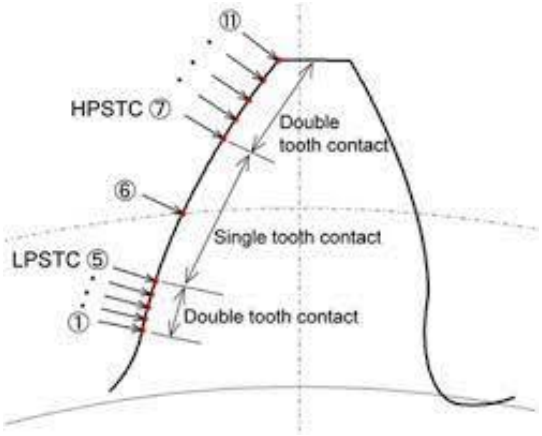


Figure 1.6 Single Tooth and Double Tooth Contact Regions [10]

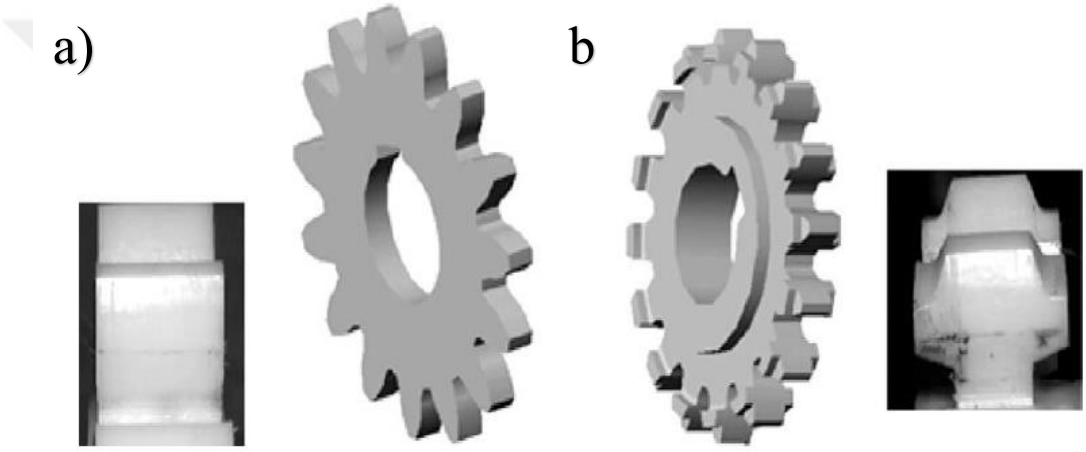


Figure 1.7 a) Unmodified Gear b) Modified Gear [10]

Taburdagitan and Akkok [11] focuses on the determination of surface temperature rise in spur gears using thermo-elastic analysis. Thermo-elastic analysis is used to investigate the surface temperature rise on gear surfaces. Thermo-elastic analysis is a method that combines thermal and mechanical analyses to study the effects of temperature on the mechanical behavior of materials. The study considers factors such as material properties, gear geometry, contact conditions, and operating parameters. Finite Element Method (FEM) is then employed to solve the thermo-elastic problem numerically. Simulations for analysis is conducted to analyze the temperature distribution and surface temperature rise in the gear teeth under different operating conditions. They investigate the influence of parameters such as load, speed, and lubrication on the temperature rise. The results of the study provide insights into the thermal behavior of spur gears and the factors that contribute to surface temperature

rise. The findings contribute to the understanding of gear performance and can inform the design and optimization of gear systems to improve their reliability and durability. In summary, the article presents a thermo-elastic analysis of spur gears to determine surface temperature rise, offering valuable insights for the design and optimization of gear systems [11].

Rae et al. [12] studied on exploring the mechanical properties of PEEK with a specific emphasis on its response to large compressive strain. The aim of the study is that involves subjecting PEEK specimens to controlled compression tests at different strain rates and temperatures to assess how these factors affect its mechanical behavior. The authors also investigate the influence of material processing methods, such as annealing and reinforcement techniques, on the mechanical properties of PEEK. According to results of the study, mechanical behavior of PEEK in scenarios involving large strains, many factors which are compressive response, including strain rate sensitivity, temperature dependency, and the effects of processing conditions are discussed.

When gears are meshed and rotated, power which is known as load is transmitted from one gear to another. In this transmission, every tooth is subjected to bending fatigue. After many load cycles, cracks initiate. Such fatal failures can be easily avoided if the gears are designed accordingly. To prevent root fatigue fractures during the required operating life, the maximum root stress must be lower than the material's fatigue strength limit for the selected cycle range, with an additional safety margin typically included. Usually also some safety margin is included. Figure 1.8 illustrates to show tooth root crack propagation. Crack is shown in the area indicated by the red circle.

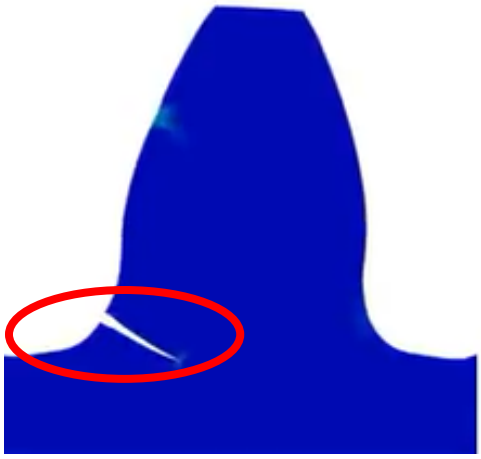


Figure 1.8 Tooth Root Crack

Tooth fatigue fracture which prevents normal operation and long gear life is an important problem, and it should be considered by designers and manufacturers. This problem can be foreseen easily anticipated by using stress analysis methods. For reliable operations, the stress values in the tooth root area should be lower than the material fatigue strength limit for related boundary conditions.

The material's fatigue strength can be represented by an S-N curve. The fatigue behavior of plastic materials is strongly temperature dependent. The formula is given below to calculate the relationship between fatigue limit and applied stress. Gear design parameters should be selected to satisfy for this inequality.

$$\sigma_F < \frac{\sigma_{Flim}}{S_{Flim}} \qquad \textbf{(Eq 1-1)}$$

According to ISO 6336 and DIN 3990 standards, and VDI 2736 guideline, the critical area which is also shown in Figure 1.8 can be determined with a 30° tangent method. The location thus determined also coincides well with the location of the maximum stress determined by the finite element method.

Another study is tried to develop performance of thermoplastic composite materials and focus on test [13]. The status in plastic gear production, failure modes, materials, lubrication methods and manufacturing are discussed in article. In particular, the advantages and disadvantages of gears made of PEEK and carbon fibers are explained. The graph is obtained to be shown below,

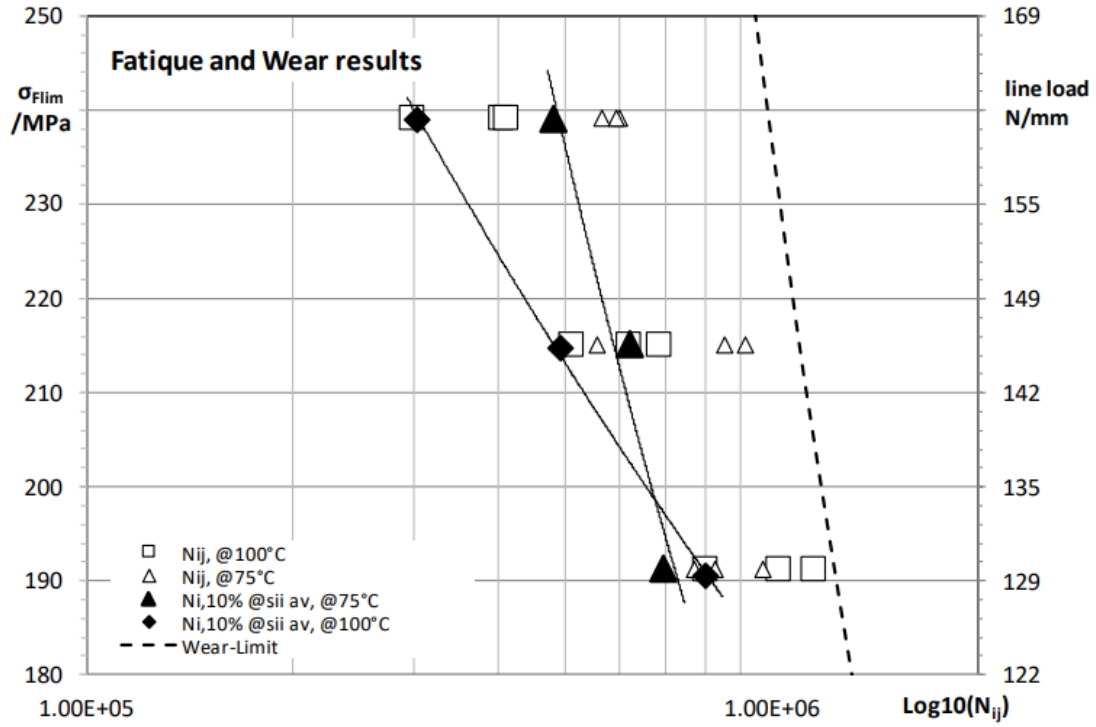


Figure 1.9 Comparison of Tested Gears at Different Temperatures [13]

Kurokawa et al. [14] are focused on the wear characteristics of fiber-reinforced polymers. The primary goal of the research outlined is to examine the wear behavior made from PMCs. The study aims to analyze the influence of several factors, including gear design parameters, material properties, lubrication conditions, and operational parameters, on the wear characteristics of these gears. The experimental results and findings related to the wear behavior of PMC gears under different conditions is presented and discussed. Effects of factors such as gear geometry, fiber orientation, sliding velocity, and lubrication on wear rates and mechanisms. The following curve is obtained.

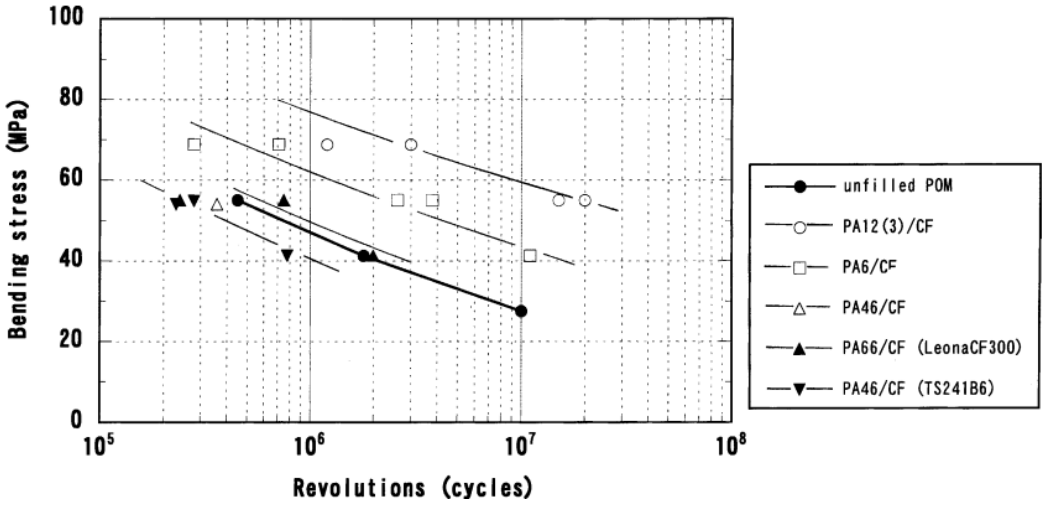


Figure 1.10 Load bearing characteristics of PA/CF gears in comparison with that of unfilled POM [14]

According to the test results, the failures of PA66-CF and PA46-CF gears were caused by the wear of teeth (wear failure), while PA12(3)-CF and PA6-CF gears seemed to be failed by fatigue since their wear amounts were quite small.

Digimat software is used to predict the micromechanical behavior of complex multiphase composite materials and structures. Different mesh sizes and RVE sizes are analyzed to determine the ideal sizes and elements. Digimat MF is a Mean-Field homogenization-based software which aims to predict the linear and nonlinear mechanical behavior of multi-phase materials based on constitutive properties of the base materials and the composite morphology such as filler content, length and aspect ratio, and orientation. Digimat FE is used to generate Representative Volume Element (RVE) for a large variety of material microstructures. According to mechanical properties of matrix and fiber materials and microstructure definition such as aspect ratio, fiber diameter and length, and fiber ply orientations, a finite element model is provided.

KISSsoft is an advanced gear program used to calculate and examine gear parameters, contact theories, meshing, and strength of materials such as Hertzian stress, root bending stress, tooth flank stress, and safeties.

For plastic gears, there are no international standards; only VDI 2736 and JIS B 1759 are used to calculate and predict plastic gear behaviors. However, these standards are not comprehensive for all gear parameters. VDI 2736 is based on ISO 6336-Method 3 and DIN 3990-3 standards. In this study, VDI 2736 is selected to calculate theoretical tooth root stress, and the provided values from theoretical calculations will be used to compare with analytical results.

By using fiber materials within the matrices, new materials are obtained in engineering programs. Mechanical and thermal properties of new materials is examined. Firstly, a gear pair is built, and root safeties are examined. Then fiber reinforced plastic materials are selected as gear materials and same analysis steps are applied on. Obtained values will be compared. Results to be discussed according to the thermal and mechanical properties. Digimat FE is used to determine mechanical properties of PA66-CF-60, and POM-CF-60 materials. Material behaviors with respect to temperature are examined. Obtained values are used both KissSoft and ABAQUS programs separately.

In Chapter 8 and 9, obtained values will be compared. Effects of fibers on plastic will be introduced. Advantages of fibers are examined. In last chapter, results are discussed. Test needs, aims of study is mentioned.

CHAPTER 2

INVESTIGATION OF POLYMER MATERIALS AND THEIR MECHANICAL PROPERTIES

Polymer materials are consisting of many units. Their molecules are made up of thousands of repeating chemical units.

Stiffness and strength are physical properties, and these properties are temperature dependent. To show the effect of temperature on polymers, Figure 2.1 is given below.

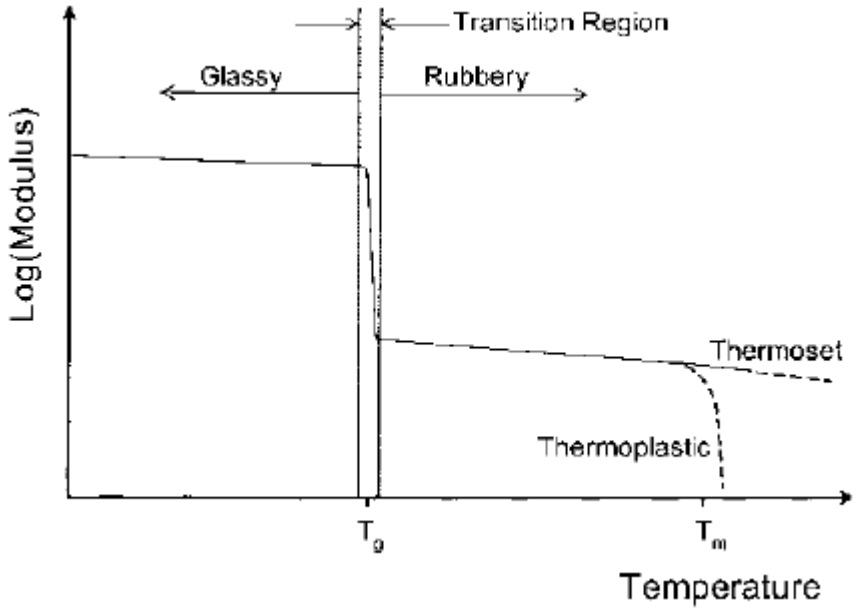


Figure 2.1 Temperature Effect on Polymers [15]

Thermoset and thermoplastic polymers exhibit the same behavior, except those high temperatures, thermoplastics cannot preserve their mechanical properties they can melt at high temperatures. According to the figure that is shown above, stiffness of polymers decreases dramatically at transition region. This region is called as glass transition temperature (T_g). In glassy region, stiffness decreases slowly when temperature increases. In this region, polymer shows brittle property. In rubbery region, polymers are ductile. Hence, the T_g denotes the transition between glassy and rubbery. This transition is associated with a sudden increase in mobility of segments within the molecular chain. In glassy region, polymer particulars are closely packaged

together and tightly bonded, and may not easily slide past each other. Also, polymers show high stiffness and strength but can be low ductility. Conversely, at temperatures well above the T_g, so in rubbery region, the molecular spacing is increased such that the molecular chains are mobile and can readily slide past each other. Consequently, the polymer is ‘‘rubbery’’ and exhibits lower stiffness and strength but higher ductility. T_m which is shown in Figure 2.1 is melting temperature. However, thermoplastics does not exhibit unique melting temperature. Furthermore, thermoset polymers cannot be melted, although the polymer is destroyed if temperature is raised to an excessive high level [15, 16].

POM, PA 66, PEEK materials are widely used polymers in gear industry. These are thermoplastic polymers and to be examined in this study. Plastic gears have many advantages such as corrosion resistance, low manufacturing cost, low weight, easy mass production and assembly processes, silent operations [15]. However, the reason of low load-carrying capacity, and performance at high temperatures, plastic gears need to improve. Poor heat conductance of plastic gears, polymers accumulate heat and soften [16].

According to given information, mechanical properties are affected at elevated temperatures.

2.1 Mechanical and Thermal Properties of POM (Polyoxymethylene)

POM material that is also known as acetal plastic is semi crystalline thermoplastic with high mechanical strength and rigidity. It is chosen to use in gear industry because it has high sliding properties, excellent wear resistance and moisture absorption is good. POM is a very versatile engineering material to use for complex components. Fatigue strength of the material is high. Thanks to its high degree of crystallinity, polyoxymethylene homopolymer exhibits high mechanical properties as strength, stiffness, and creep. In the case of POM copolymer, the strength and stiffness are slightly lower because its lower crystallinity degree. Furthermore, the use domain ranges from -30°C to 150°C that allows various applications. Furthermore, friction properties are excellent due to oxygen contained into POM monomer [17, 18].

General properties of POM thermoplastic are listed below in Table 2.1:

Table 2.1 Properties of POM

Properties	Values	Unit
Modulus of Elasticity	2750	MPa
Tensile Strength	73	MPa
Tensile Strength at Yield	12.5	MPa
Elongation at break (tensile test)	65	%
Flexural Modulus	2.58	MPa
Glass Transition Temperature	80	℃
Transient Working Temperature	140	℃
Thermal Expansion (23-100°C, long)	11	10^{-5}*1/K
Density	1.41	g/cm^3
Specific Heat Conductivity	0.31	W/m. K
Specific Heat Capacity	1.47	J/g. K
Coefficient of Thermal Expansion	110	10^{-6}/K
Poisson Ratio	0.44	-

2.2 Mechanical and Thermal Properties of PA 66 (Polyamide)

PA 66 is a polymer material which has high rigidity, good hardness, good abrasion and corrosion resistance and thermal dimensional stability. In addition, nylon 66 has outstanding wear resistance and low frictional properties. It also has particularly good temperature, chemical, and impact properties. Because of the absorb moisture properties of PA 66 which may affect the stability of dimension, engineers and scientists should be aware. In fact, PA 66 will tend to absorb more moisture at saturation than cast PA type 6. Mechanical properties of PA 66 are listed below table [15].

Table 2.2 Properties of PA 66

Properties	Values	Unit
Modulus of Elasticity	2810	MPa
Tensile Strength	82.5	MPa
Tensile Strength at Yield	8.6	MPa
Elongation at break (tensile test)	156	%
Flexural strength	110	MPa
Compression modulus	66	MPa
Glass Transition Temperature	78	℃
Service Temperature Range	-60 – 160	℃
Thermal Expansion (23-100°C, long)	12	10-5*1/K
Density	1.14	g/cm^3
Poisson Ratio	0.39	-

S-N curves of POM and PA 66 is given below:

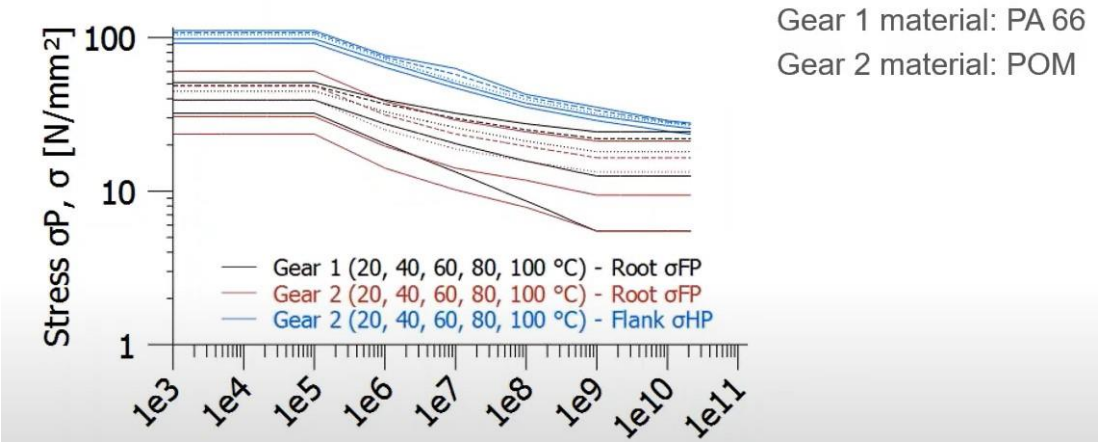


Figure 2.2 S-N Curves of POM and PA 66 [2]

CHAPTER 3

INVESTIGATION OF FIBROUS MATERIALS AND THEIR MECHANICAL PROPERTIES

Reinforcing fibers are used to improve mechanical properties of polymers that are major strengthening element for all polymeric materials [15]. Most widely used fibrous materials can be classified as glass, aramid, graphite, carbon, boron, and polyethylene (PE).

Fibrous materials affect the mechanical properties of composite materials directly. Fiber diameters, length, and ply angles are examined when mechanical properties of material are determined. Fiber diameters are selected as quite small ranging about 5 to 12 μm for glass, aramid, or graphite fibers. For polyethylene fibers, diameter range is 25 to 40 μm, for boron and silicon carbide fibers, diameter range is 100 to 200 μm.

Young Modulus ranges of glass fibers are about 70 GPa, for carbon fibers about 400 GPa.

Fiber diameter in composite materials affect elastic properties directly. Small diameters reduce the number and size of flaws in material, so increasing strength. As an example, glass fibers with a diameter of about 1 mm, have strength nearly 170 MPa. However, when the same glass is used as fiber material with 10 (ten) μm diameter, then strength of material is about 3450 MPa. For organic fibers, strengthening is accomplished by stretching the fiber and thereby aligning the polymer molecules. This produces fibers that are themselves anisotropic. For carbon or graphite fibers, strengthening is accomplished by aligning the basal planes of adjoining crystals, also producing an anisotropic fiber [15].

3.1 Glass Fiber

In this section, several types of fibers will be introduced according to their properties. For the first fiber material, glass fiber is examined which is fabricated by melting glass at nearly 1260 °C. There are two major types of glass fiber are called as “E-Glass” and “S-Glass.” E-Glass involves alumina borosilicate, so it has high electricity resistance.

S-glass involves magnesium aluminosilicate, so it has high tensile strength. Glass fibers are usually isotropic, and their typical properties are shown in Table 3.1.

Table 3.1 Properties of Glass Fibers (E-glass and S-glass)

Property	E-glass	S-glass
Specific Gravity	2.60	2.50
Young Modulus, MPa	72000	87000
Tensile Strength, MPa	3450	4310
Elongation, %	4.8	5.0
Coefficient of Thermal Expansion μm/m/°C	5.0	5.6
Poisson Ratio	0.21	0.22
Density (g/cm^3)	2.49	2.55

E-glass fibers are selected for low-cost applications. However, S-glass materials can be selected in applications requiring high mechanical strength and durability, such as aerospace, aviation, and military applications [19, 20].

3.2 Aramid Fiber

Aramid Fiber, which is also called as Kevlar is most widely used organic fiber. Aramid Fiber is based on poly (p-phenylene terephthalamide), which is a member of a family of polymers called ‘‘aramids.’’ They are formed by condensation / elongation process. The resulting fibers are anisotropic because covalent bonds are strong and formed in fiber direction. Weak hydrogen bonds are formed in transverse direction. Aramid fibers have high tensile strength, toughness, and stiffness values in axial direction because of strong covalent bonds.

Kevlar material have different grades which are Kevlar 149, Kevlar 49, Kevlar 129, Kevlar 29. In these grades, Kevlar 149 has the highest modulus and Kevlar 49 has highest strength. Kevlar 129 and Kevlar 29 with lower strength values but with a higher percent elongation [15].

Table 3.2 Typical Properties of Kevlar Fibers (in Axial Direction)

Property	Kevlar 149	Kevlar 49	Kevlar 129	Kevlar 29
Specific Gravity	1.44	1.44	1.44	1.44
Young Modulus, MPa	186000	124000	96000	68000
Tensile Strength, MPa	3440	3700	3380	2930
Elongation, %	2.5	2.8	3.3	3.6
Coefficient of Thermal Expansion μm/m/°C	-2.0	-2.0	-2.0	-2.0

3.3 Graphite and Carbon Fiber

The elemental carbon content of either type of fiber is above 90%, and the stiffest and strongest fibers have carbon contents approaching 100%. Some effort has been made to standardize these terms by defining graphite fibers as those that have:

A carbon content above 95%

Been heat-treated at temperatures more than 1700 °C.

Been stretched during heat treatment to produce a high degree of preferred crystalline orientation.

A Young's modulus on the order of 345 GPa.

Fibers that do not satisfy all the above conditions are called carbon fibers under this standard. However, as stated above, in practice, this definition is not widely followed, and the terms ‘‘graphite’’ and ‘‘carbon’’ are often used interchangeably. Both graphite and carbon fibers are produced by thermal decomposition of an organic (i.e., polymeric) fiber or ‘‘precursor’’ at high pressures and temperatures. The three most common precursors are:

Polyacrylonitrile (‘‘PAN’’)

Pitch (a by-product produced during the petroleum distillation process)

Rayon.

PAN fiber is extensively used across various industries. The specific steps in the fabrication process are proprietary, so only a general overview can be provided. During fabrication, the precursor material is initially drawn into a thread. It is then oxidized at approximately 260°C (500°F) to form crosslinks and an extended carbon network. The precursor is then subjected to carbonization treatment, during which noncarbon atoms are driven off. This step typically involves temperatures of approximately 700°C and is usually conducted in an inert atmosphere. Finally, during the graphitization process, the fibers are subjected to a combination of elevated temperature and tensile elongation.

Table 3.3 Typical Properties of Carbon Fiber

Property	**Low Modulus**	**Intermediate Modulus**	**Ultra-High Modulus**
Specific Gravity	1.8	1.9	2.2
Young Modulus, MPa	23000	370000	900000
Tensile Strength, MPa	3450	2480	3800
Elongation, %	1.1	0.5	0.4
Coefficient of Thermal Expansion µm/m/°C	-0.4	-0.5	-0.5

To analyze composite materials mechanical properties in Digimat, following values are used [21, 22]

- Modulus of Elasticity: 375 GPa
- Poisson Ratio: 0.22
- Thermal Expansion Coefficient:$5x10^{-7}$/°C
- Density: 1.80 g/cm^3

3.4 Polyethylene Fibers

Having high strength and high modulus polyethylene is called Spectra. It is based on ultra-high-molecular-weight polyethylene (UHMWPE). Spectra specific gravity has lower than water (0.97).

T_g of UHMWPE is in the range of -20 °C to 0 °C, and hence the fiber is in the rubbery state at a room temperature condition and exhibits time dependent behavior. It allows high impact resistance and toughness but creep effects under long term loading cannot be desirable. The melting temperature of the fiber is 147 °C so this fiber cannot be used in many industries [15].

Table 3.4 Typical Properties of Polyethylene Fiber

Property	Spectra 900	Spectra 1000	Spectra 2000
Specific Gravity	0.97	0.97	0.97
Young Modulus, MPa	70000	105000	115000
Tensile Strength, MPa	2600	3200	3400
Elongation, %	3.8	3.0	3.0
Coefficient of Thermal Expansion µm/m/°C	>70	>70	>70

CHAPTER 4

MATERIAL DEVELOPMENT

4.1 Development of New Materials by Using POM, PA 66 and Fibers

Digimat MF and FE is used to provide mechanical properties of composite materials. In this section, Digimat is examined and new composite materials their mechanical properties to be provided.

Digimat is a commonly used application in engineering materials area. Interface of Digimat MF graphical user interface is illustrated in Figure 4.1 Digimat MF GUIFigure 4.1 below:

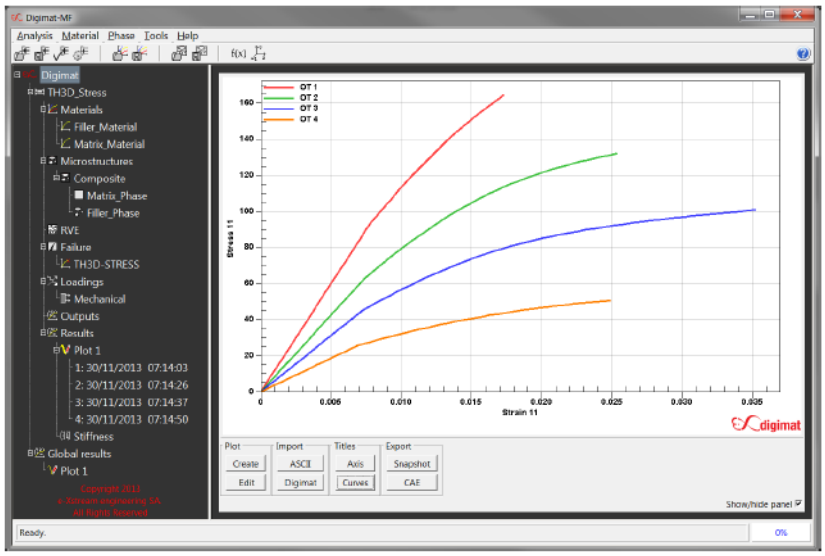


Figure 4.1 Digimat MF GUI

The right side of the screen involves analysis steps which are Materials, Microstructures, RVE, Loadings, Outputs, and results.

There are several methods in analysis tab which are mechanical, thermo-mechanical, thermal, and electrical. In this thesis study, mechanical and thermo-mechanical analyses to be performed. Stress-strain curves, stiffness, and thermal expansion will be provided by using RVE taking from Digimat output.

In this section, RVE size, mesh independence to be examined. Different RVE sizes such as 25x25x25, 50x50x50, 75x75x75, and 100x100x100 (in µm) will be arranged and stiffness values will be calculated. Then, by using the best choice analysis will be done with different mesh sizes. After that, different composite materials which will be mentioned after will be examined. A block diagram of the processes is given below.

Figure 4.2 Block Diagram of Steps

Composite materials which to be analyzed are listed in Table:

Table 4.1 Mixing Properties

Composite Material	Matrix Material	Fiber Material	Volume Fraction
POM-GF-30%	POM	Glass Fiber	30
POM-GF-45%	POM	Glass Fiber	45
POM-GF-60%	POM	Glass Fiber	60
POM-CF-30%	POM	Carbon Fiber	30
POM-CF-45%	POM	Carbon Fiber	45
POM-CF-60%	POM	Carbon Fiber	60
PA66-GF-30%	PA 66	Glass Fiber	30
PA66-GF-45%	PA 66	Glass Fiber	45
PA66-GF-60%	PA 66	Glass Fiber	60
PA66-CF-30%	PA 66	Carbon Fiber	30
PA66-CF-45%	PA 66	Carbon Fiber	45
PA66-CF-60%	PA 66	Carbon Fiber	60

Production processes of CFRP materials are listed below [23-24]:

- Preparation of fiber reinforcement materials: Fibers are cut to specific lengths that are analyzed in design phases. Then, fibers are mixed with polymers.
- Preparation of polymer materials.
- Extrusion Process: The dried mixture which includes polymer and fibers fed into an extruder. In this process, polymer material is melted under elevated temperature and pressure and homogeneously mixed with the fibers. During this phase, the molten polymer and fiber material mixture is continuously stirred by the extruder’s screws.
- Cooling and Pelletizing: The material exiting the extruder is cooled and solidified in a water bath. The solidified material is pelletized. These pellets are used to produce final products through injection molding to other methods.

- Injection molding: Pellets are fed into an injection molding machine. They are melted at high temperature. After this process, pellets injected into molds to obtain the desired shape (spur, helical, worm gears, splines, so on).

4.1.1 Investigation of RVE Sizes

In this section, RVE sizes to be examined which to be mentioned below. PA-GF-30% material is selected to calculate stiffness values. Stiffness results, elastic modulus of materials will have compared each other and discussed. The analysis to be conducted is given in Figure 4.3.

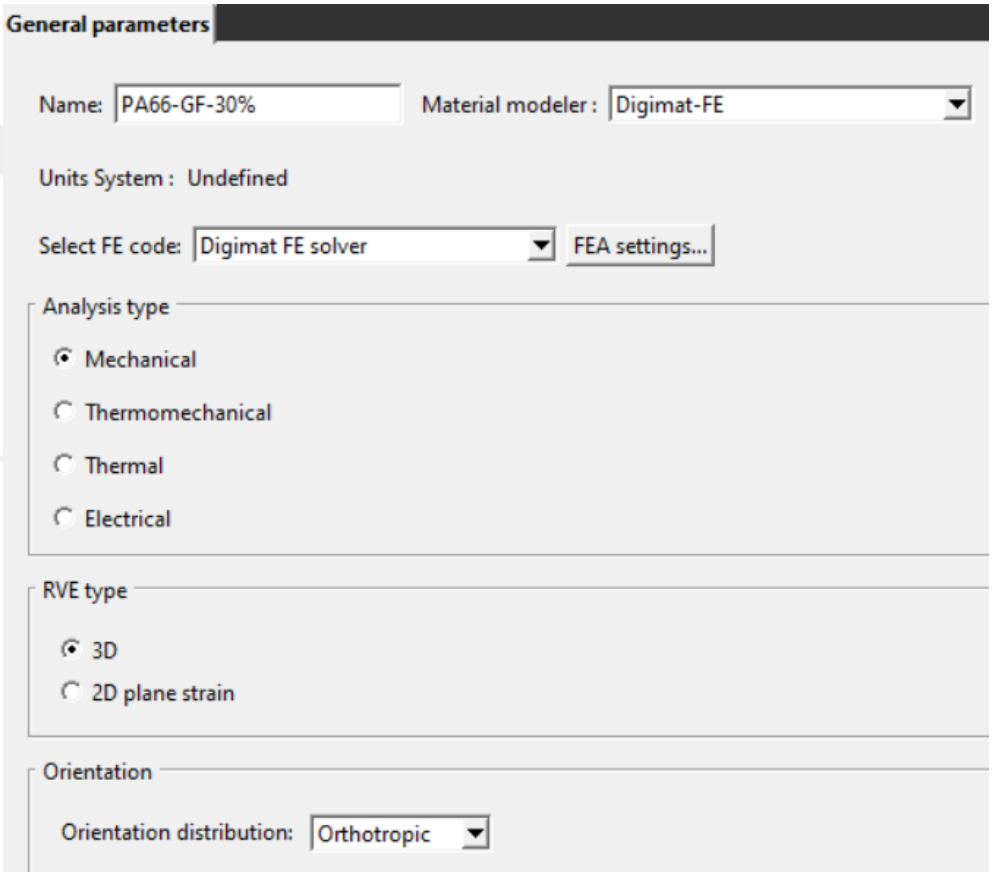


Figure 4.3 Analysis Properties

For the first step, materials properties are set in the materials section. Glass behavior is defined as Elastic and isotropic. PA66 is defined as Elastoplastic. It is illustrated below:

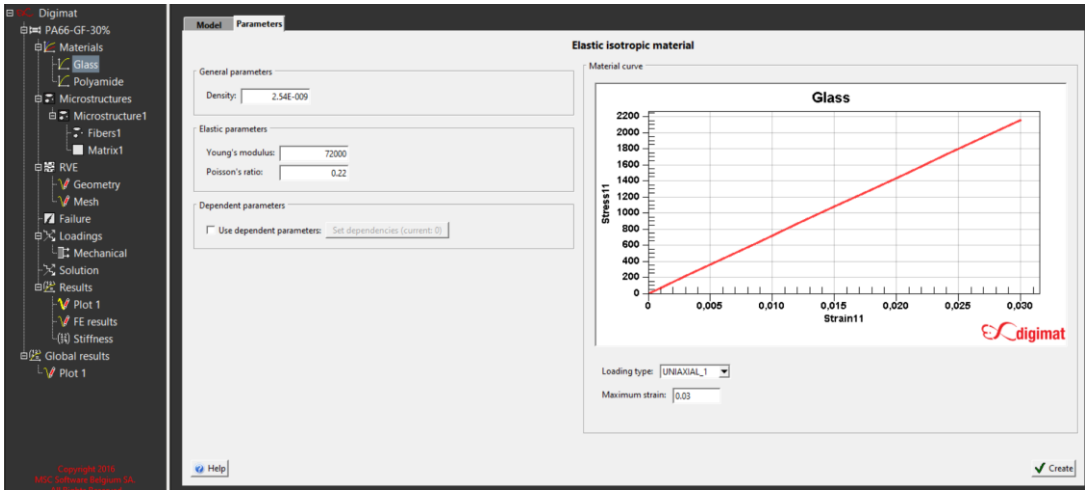


Figure 4.4 Mechanical Properties of Glass (Fiber Material)

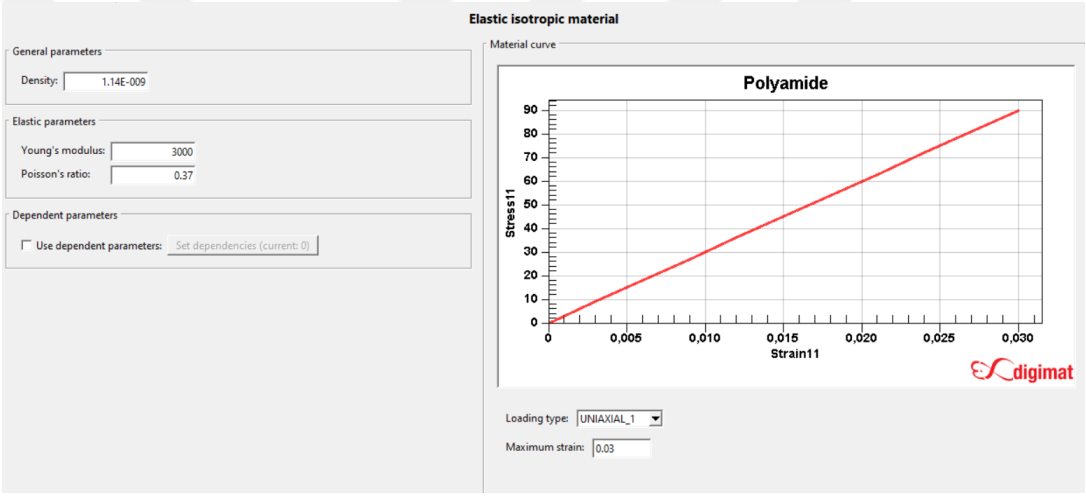
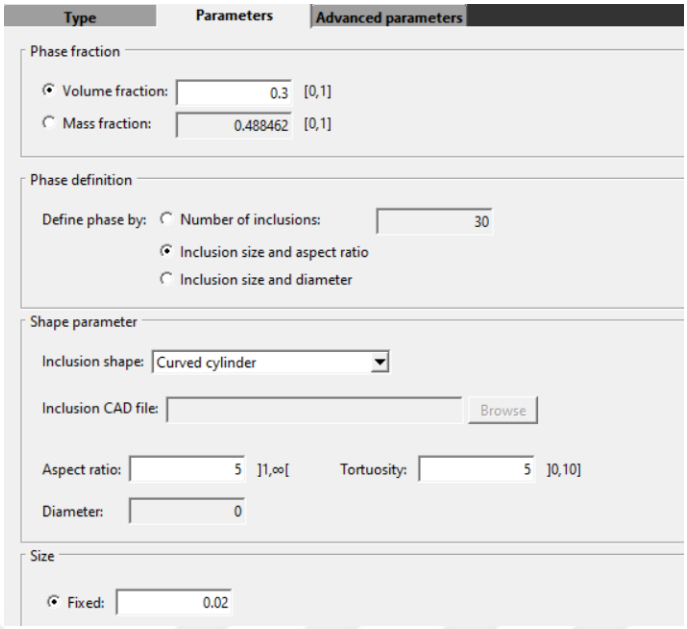


Figure 4.5 Mechanical Properties of Polyamide (Matrix Material)

Then, microstructure is defined as given below:

a)



b)

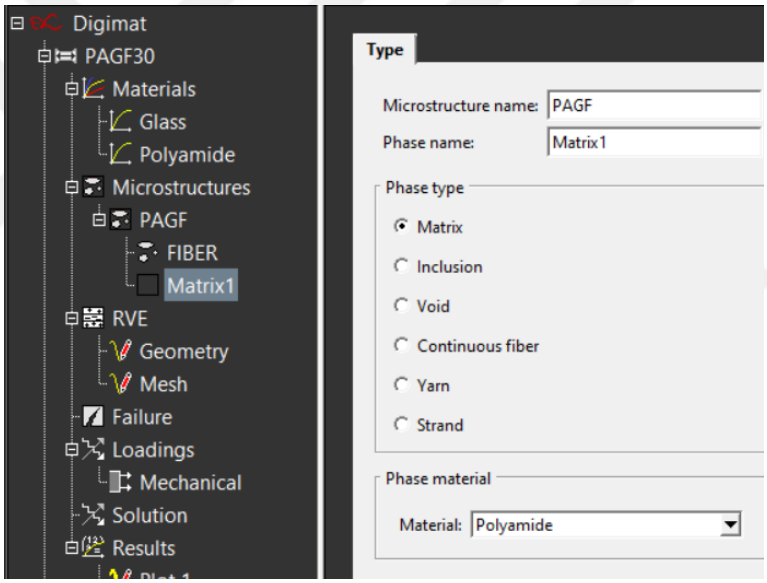


Figure 4.6 Microstructure Properties a) Fiber Diameter Shape and Ratios b) Matrix type.

Diameters of fibers are set as 0.002 mm. Then, RVE size is set as 25x25x25 μm^3.

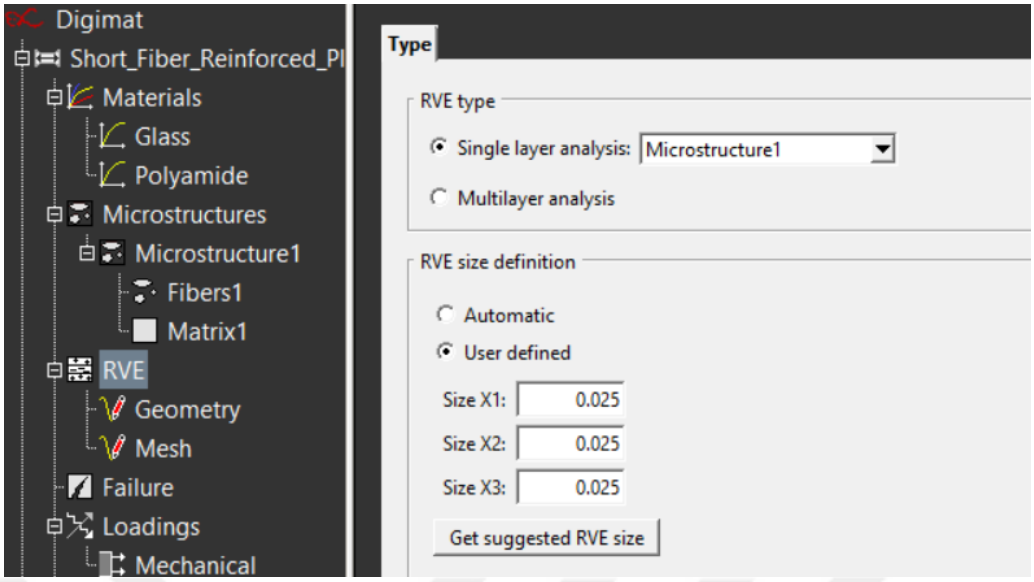


Figure 4.7 Mesh Size

RVE and mesh are given as follows. Mesh element size is arranged as 25x25x25 voxel element. These mesh elements size will also use for other studies which will represent different RVE sizes.

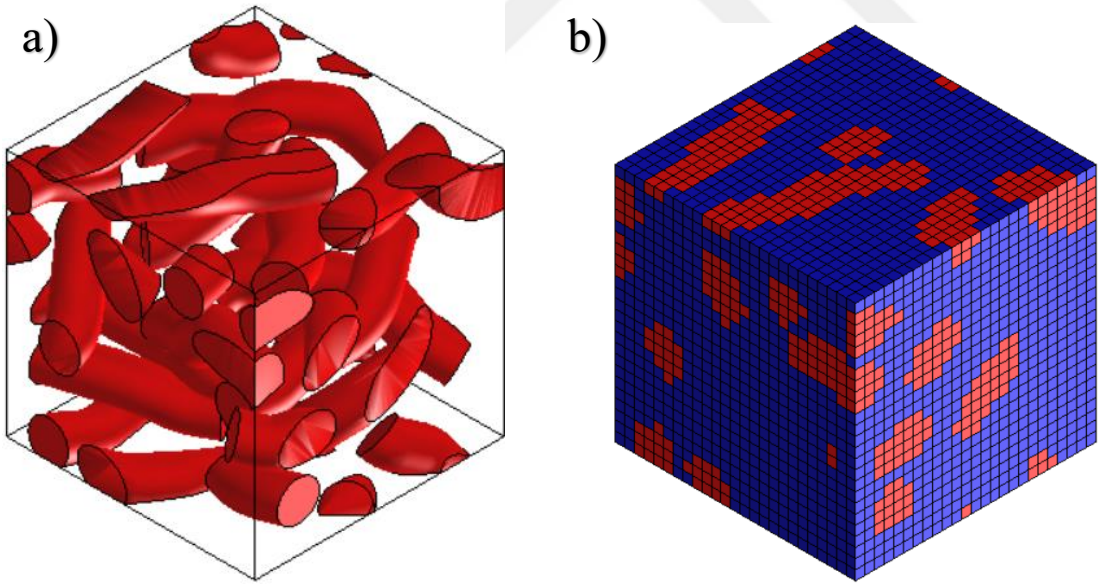


Figure 4.8 25x25x25 RVE Size a) RVE of PA-GF-30% b) Mesh.

Analysis is completed in 373 seconds. Results values are obtained as listed below:

$$\begin{pmatrix} E_{11} & E_{12} & E_{13} \\ E_{21} & E_{22} & E_{23} \\ E_{31} & E_{32} & E_{33} \end{pmatrix} = \begin{bmatrix} 6991.84 & 2615.48 & 2403.34 \\ 2615.48 & 6840.79 & 2276.05 \\ 2403.34 & 2276.05 & 7869.58 \end{bmatrix} \quad \textbf{(Eq 4-1)}$$

$$\begin{pmatrix} \nu_{12} & \nu_{13} \\ \nu_{21} & \nu_{23} \\ \nu_{31} & \nu_{32} \end{pmatrix} = \begin{bmatrix} 0.317586 & 0.332059 \\ 0.31725 & 0.333175 \\ 0.297371 & 0.304958 \end{bmatrix} \quad \textbf{(Eq 4-2)}$$

The stiffness matrix is obtained as follows:

	11	22	33	12	23	13
11	9845.61	4494.57	4297.73	-33.9079	12.2632	-22.1431
22	4494.57	9697.44	4283.79	-298.519	31.6151	15.9337
33	4297.73	4283.79	8848.31	-4.93668	83.4475	-16.5224
12	-33.9078	-298.518	-4.93655	2628.42	4.5411	44.4422
23	12.2632	31.6151	83.4475	4.5411	2277.39	-32.4847
13	-22.1431	15.9337	-16.5224	44.4422	-32.4847	2404.77

As a second analysis, RVE size of SFRP is changed as 50x50x50 μm^3. Voxel element size, material types and mechanical properties, phase fraction ratio, diameter of small fibers take constant. The results are obtained as follows in 19 seconds:

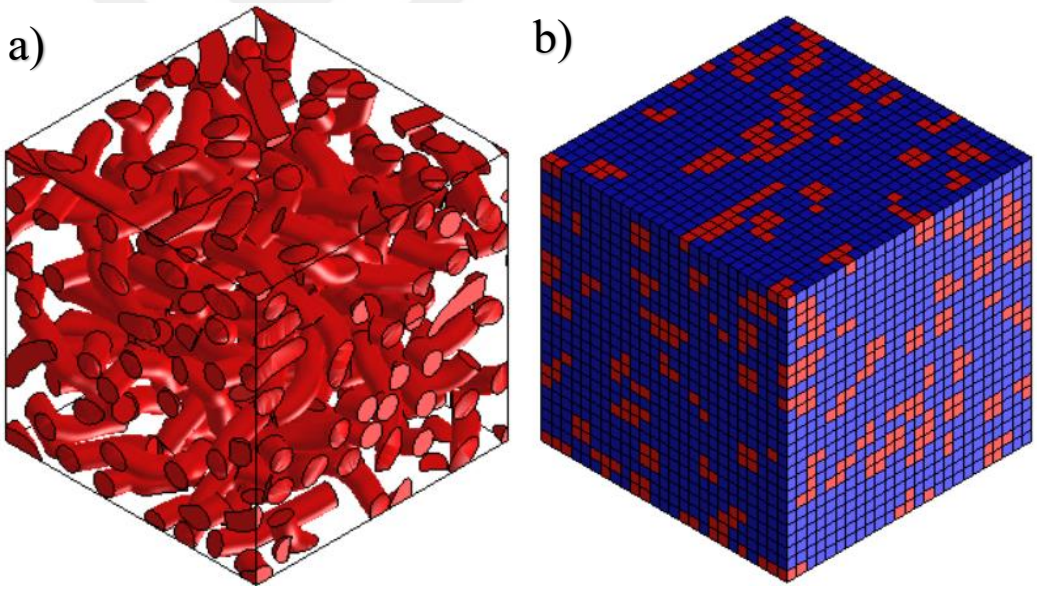


Figure 4.9 50x50x50 RVE Size a) RVE of PA-GF-30% b) Mesh.

$$\begin{pmatrix} E_{11} & E_{12} & E_{13} \\ E_{21} & E_{22} & E_{23} \\ E_{31} & E_{32} & E_{33} \end{pmatrix} = \begin{bmatrix} 8028.35 & 2819.43 & 2631.66 \\ 2819.43 & 8020.14 & 2587.55 \\ 2631.66 & 2587.55 & 7206.3 \end{bmatrix} \quad \textbf{(Eq 4-3)}$$

$$\begin{pmatrix} \nu_{12} & \nu_{13} \\ \nu_{21} & \nu_{23} \\ \nu_{31} & \nu_{32} \end{pmatrix} = \begin{bmatrix} 0.288690 & 0.319220 \\ 0.288395 & 0.315433 \\ 0.286534 & 0.283424 \end{bmatrix} \quad \textbf{(Eq 4-4)}$$

The stiffness matrix is obtained as following result:

	11	22	33	12	23	13
11	10694.2	4449.34	4325.63	-45.3192	-16.3412	36.0228
22	4449.34	10659.9	4297.5	-49.9602	-52.1314	28.0105
33	4325.63	4297.5	9666.18	-29.6153	-89.6476	8.70923
12	-45.3192	-49.9602	-29.6153	2820.03	14.8125	-23.7872
23	-16.3412	-52.1314	-89.6476	14.8125	2589.04	-35.6606
13	36.0228	28.0105	8.70922	-23.7872	-35.6606	2632.52

The last analysis of this section is that RVE size is arranged as 75x75x75 μm^3. Voxel element size, material types and mechanical properties, phase fraction ratio, diameter of small fibers take constant. The results are obtained as follows in 2145 seconds:

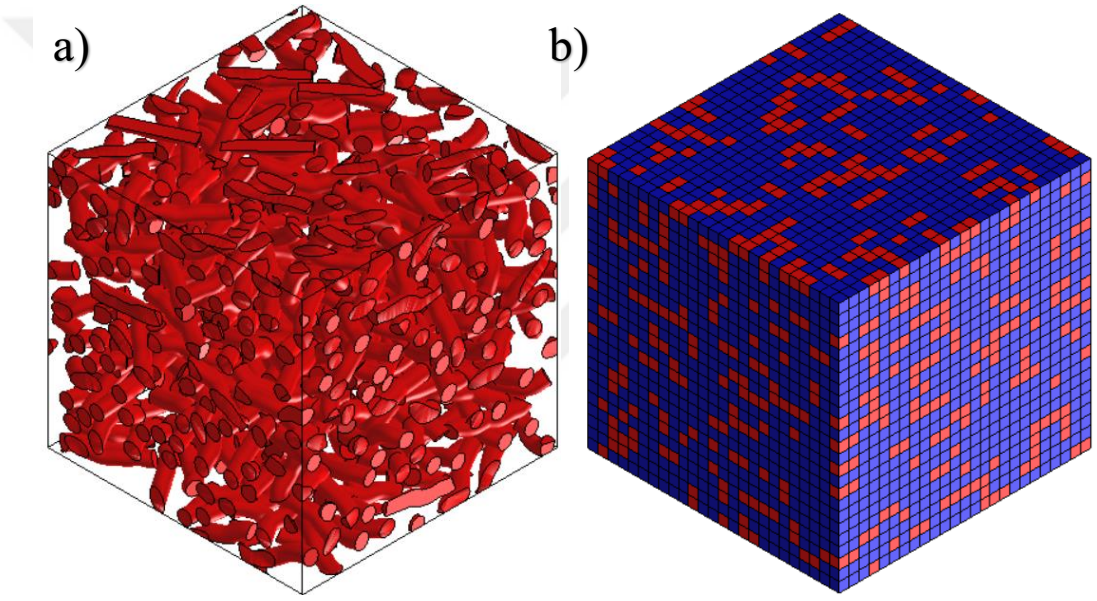


Figure 4.10 75x75x75 RVE Size a) RVE of PA-GF-30% b) Mesh.

$$\begin{pmatrix} E_{11} & E_{12} & E_{13} \\ E_{21} & E_{22} & E_{23} \\ E_{31} & E_{32} & E_{33} \end{pmatrix} = \begin{bmatrix} 8518.95 & 2835.85 & 2633.1 \\ 2835.85 & 8479.93 & 2604.3 \\ 2633.1 & 2604.3 & 7651.88 \end{bmatrix} \quad \textbf{(Eq 4-5)}$$

$$\begin{pmatrix} \nu_{12} & \nu_{13} \\ \nu_{21} & \nu_{23} \\ \nu_{31} & \nu_{32} \end{pmatrix} = \begin{bmatrix} 0.280592 & 0.303801 \\ 0.279307 & 0.303569 \\ 0.272880 & 0.273926 \end{bmatrix} \quad \textbf{(Eq 4-6)}$$

The stiffness matrix is obtained as following result:

	11	22	33	12	23	13
11	11015.1	4350.97	4197.75	6.53869	13.6286	10.027
22	4350.97	10968	4191.76	31.8585	16.7048	1.80357
33	4197.75	4191.76	9945.78	4.59379	23.7686	14.3491
12	6.53869	31.8585	4.59379	2836.01	3.48428	12.5513
23	13.6286	16.7048	23.7687	3.48429	2604.37	4.47947
13	10.0271	1.80357	14.3491	12.5513	4.47947	2633.19

Taking results which have been obtained from analyses to mention above will have compared each other. Elastic modulus in x-direction, y-direction and z-direction are taking as example. The graph is given below to show elastic modulus of materials. According to the graph, elastic modulus depends on RVE size. The ideal RVE is chosen with respect to high elastic modulus property. Analysis times to mention in analysis will not consider because effect of analysis time can be negligible. As a result, 50x50x50 μm^3 will be used because of the highest modulus of elasticity values except in x-direction and duration of analysis time may be ideal.

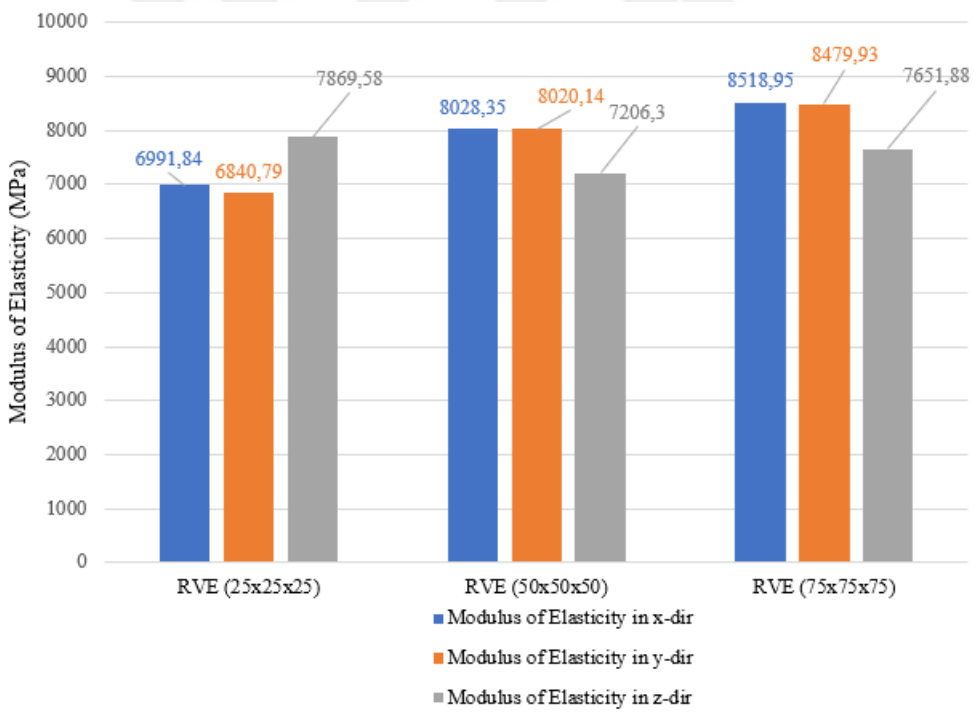


Figure 4.11 Comparison of Elastic Modulus of Three Materials (RVE Size)

4.1.2 Investigation of Mesh Sizes Dependency

In this section, the effect of mesh size to be examined and analyzed. As examined last section, 50x50x50 μm^3 can be suitable to use because it has ideal modulus of elasticity values. In this part, different size of number of voxels which are 25x25x25, 50x50x50, 100x100x100 will be examined.

First, materials, microstructure are set same as mentioned in the last section. Steps are listed below:

Glass material is defined as elastic and its mechanical properties such as density, Young's Modulus, and Poisson's Ratio.

Fiber material is defined as elastic and its mechanical properties such as density, Young's Modulus, and Poisson's Ratio.

Microstructures are defined as Glass is Fiber material and diameter of fiber is defined as 0.02.

RVE sizes is set as 50x50x50 μm^3.

The number of voxel mesh is set as:

- ➔ In first analysis: 25x25x25
- ➔ In second analysis: 50x50x50
- ➔ In third analysis: 75x75x75

After analysis, $E_{11}, E_{22}, E_{33}, E_{12}, E_{13}, E_{23}$ to be provided then E_{11}, E_{22}, E_{33} will compare.

Furthermore, stiffnesses of SFRP's are obtained.

Results will discuss and ideal mesh size to be selected.

25x25x25 number of voxel elements is set as first analysis. Illustration is given below:

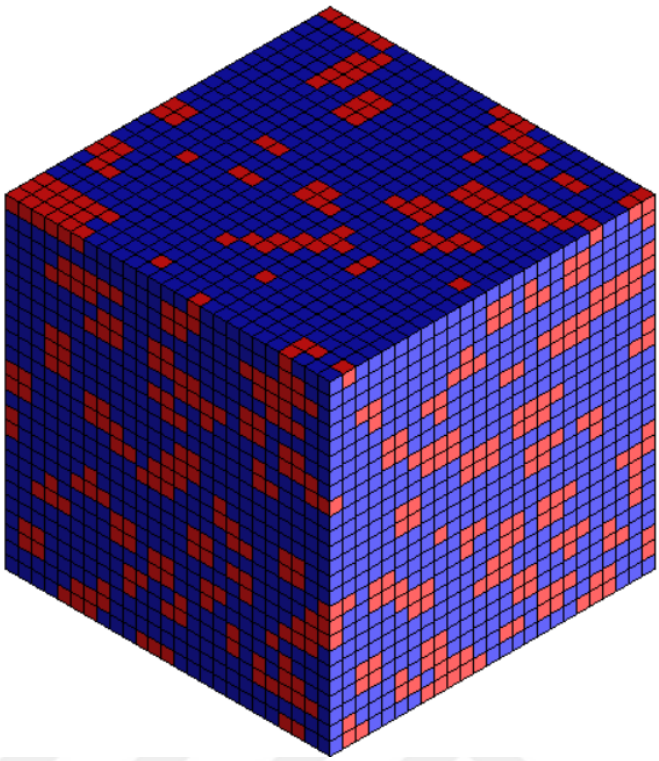


Figure 4.12 Mesh of RVE 25x25x25 Voxel Element

Results of the analysis is given below:

$$\begin{pmatrix} \mathbf{E_{11}} & \mathbf{E_{12}} & \mathbf{E_{13}} \\ \mathbf{E_{21}} & \mathbf{E_{22}} & \mathbf{E_{23}} \\ \mathbf{E_{31}} & \mathbf{E_{32}} & \mathbf{E_{33}} \end{pmatrix} = \begin{bmatrix} \mathbf{7793.77} & \mathbf{2770.27} & \mathbf{2549.38} \\ \mathbf{2770.27} & \mathbf{7856.04} & \mathbf{2559.43} \\ \mathbf{2549.38} & \mathbf{2559.43} & \mathbf{6971.88} \end{bmatrix} \quad \textbf{(Eq 4-7)}$$

	11	22	33	12	23	13
11	10448.6	4422.76	4269.4	63.9549	10.6343	45.0278
22	4422.76	10524.7	4278.16	62.51	20.0304	-15.3493
33	4269.4	4278.16	9423.05	14.1433	-29.5104	33.6358
12	63.955	62.51	14.1433	2770.92	-10.7922	2.84255
23	10.6343	20.0304	-29.5104	-10.7922	2559.72	-2.52368
13	45.0278	-15.3493	33.6358	2.84255	-2.52368	2549.8

50x50x50 number of voxel elements is set as the second analysis. Results are illustrated below:

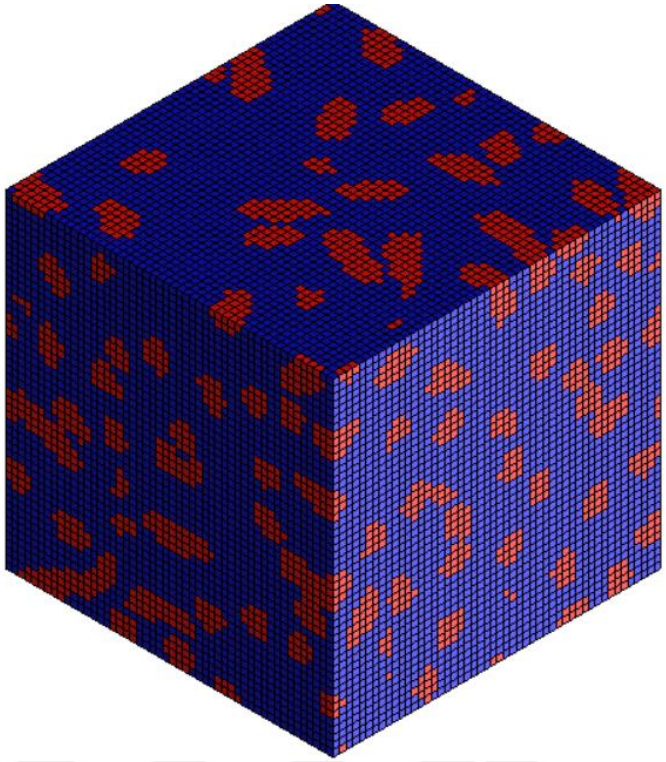


Figure 4.13 Mesh of RVE 50x50x50 Voxel Element

Results of the analysis which are provided in 270 seconds are given below:

$$\begin{pmatrix} E_{11} & E_{12} & E_{13} \\ E_{21} & E_{22} & E_{23} \\ E_{31} & E_{32} & E_{33} \end{pmatrix} = \begin{bmatrix} 7300.35 & 2500.17 & 2297.39 \\ 2500.17 & 6602.63 & 2245.15 \\ 2297.39 & 2245.15 & 6040.9 \end{bmatrix} \quad \text{(Eq 4-8)}$$

	11	22	33	12	23	13
11	10118.7	4381.72	4248.68	3.19073	4.78356	26.34
22	4381.72	9350.11	4246.74	3.88574	-21.1457	-14.5048
33	4248.68	4246.74	8601.87	1.66293	-37.0286	11.8118
12	3.19073	3.88574	1.66293	2500.22	-1.87115	10.419
23	4.78356	-21.1457	-37.0286	-1.87115	2244.42	7.98783
13	26.34	-14.5048	11.8118	10.419	7.98783	2297.63

75x75x75 number of voxel elements is set as first analysis. The illustration is given below:

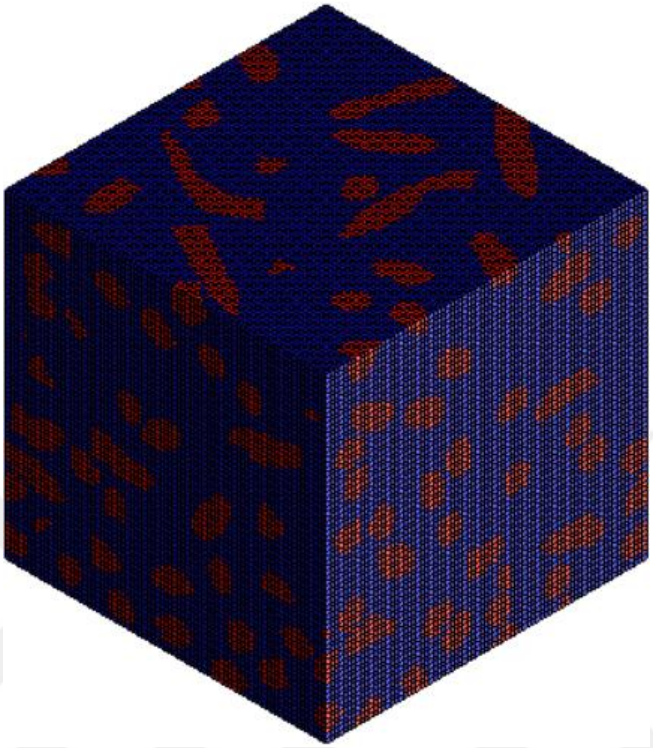


Figure 4.14 Mesh of RVE 75x75x75 Voxel Element

Results of the analysis is given below which are provided in 1946 seconds:

$$\begin{pmatrix} E_{11} & E_{12} & E_{13} \\ E_{21} & E_{22} & E_{23} \\ E_{31} & E_{32} & E_{33} \end{pmatrix} = \begin{bmatrix} 6069.47 & 2249.05 & 2041.35 \\ 2249.05 & 5960.66 & 2036.95 \\ 2041.35 & 2036.95 & 5458.43 \end{bmatrix} \quad \textbf{(Eq 4-9)}$$

	11	22	33	12	23	13
11	8889.28	4281.95	4108.11	97.3531	-4.75121	84.2691
22	4281.95	8761.58	4103.61	78.0617	-54.3876	18.5918
33	4108.12	4103.61	8031.67	24.9625	-11.239	59.4335
12	97.3531	78.0617	24.9625	2250.53	12.105	-6.86642
23	-4.75121	-54.3876	-11.239	12.105	2037.91	31.0878
13	84.2692	18.5918	59.4335	-6.86643	31.0878	2042.86

Results are obtained and compared. Because of analysis time and mechanical properties results, the ideal mesh element number is selected as 50x50x50 (total number of voxel elements are 125000).

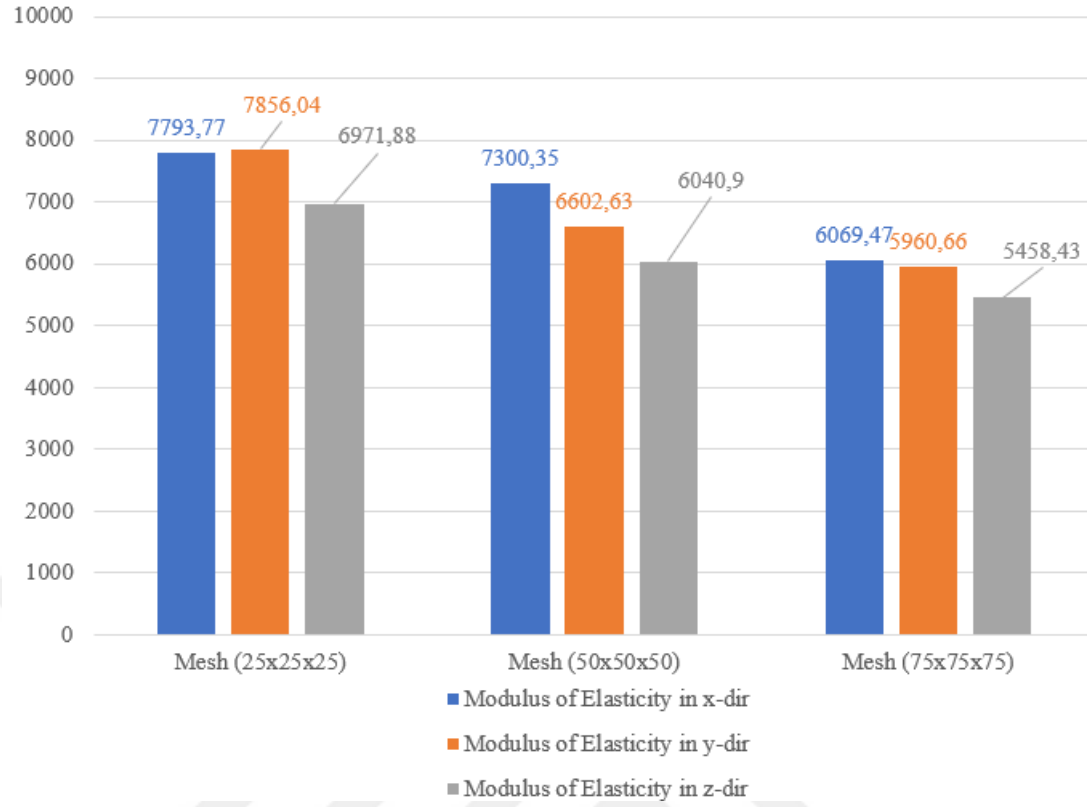


Figure 4.15 Comparison of Elastic Modulus of Three Materials (Number of Mesh)

4.2 Investigation of Mechanical Properties Of Fiber Reinforced PA 66

Mechanical properties of POM are examined in Section 2.1 above. By using Digimat, different types of fibers to be used with different length and diameter. Glass fiber, aramid fiber, carbon fiber and polyethylene fiber can be added and mechanical properties to be provided by Digimat. However, because of the high mechanical properties of carbon and glass fibers, these materials are used as fibers. Fibers and matrix materials mechanical properties are set at ideal conditions. PA 66, glass fiber, and carbon fiber densities and Young Moduli values are set at 20 ℃.

Design parameters of materials are given as following figures. Fibers are randomly oriented.

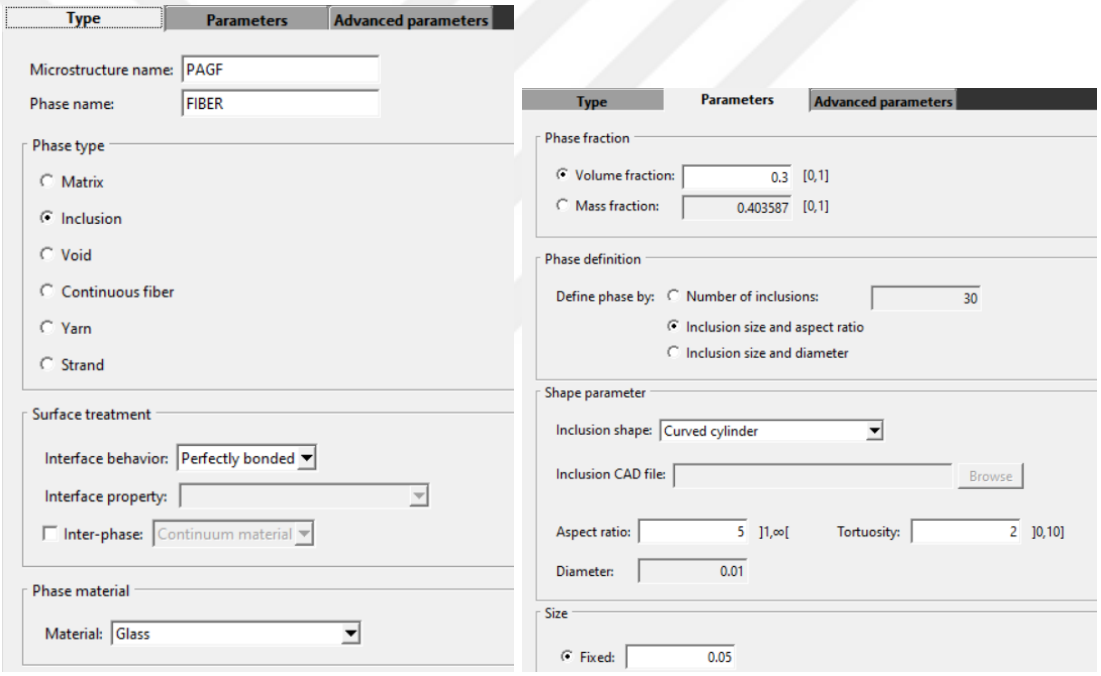


Figure 4.16 Fiber Design Parameters

4.2.1 PA 66 and Glass Fiber

Stiffness matrices and modulus of elasticity of 3 types of materials (SFRP) will be obtained at ideal conditions which RVE size is 50x50x50 μm^3, 50x50x50 number of voxel element, isotropic material behavior, fiber diameter is 0.05 µm.

Table 4.2 PA 66-Glass Fiber Materials

New Material	Fiber Material	Matrix Material	Volume Fraction	Homogenization
PA66-GF-30%	Glass Fiber	PA 66	30	Mori-Tanaka-First Order
PA66-GF-45%	Glass Fiber	PA 66	45	Mori-Tanaka-First Order
PA66-GF-60%	Glass Fiber	PA 66	60	Mori-Tanaka-First Order

Modulus of elasticity in 3D of three different materials are given as follows:

PA66-GF-30%

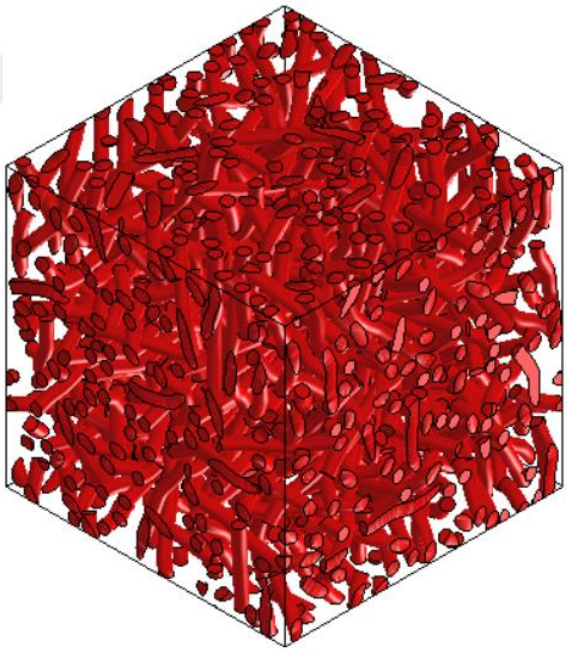


Figure 4.17 RVE of PA 66-GF-30

$$\begin{pmatrix} E_{11} \\ E_{22} \\ E_{33} \end{pmatrix} = \begin{bmatrix} 6503.04 \\ 6422.26 \\ 5586.97 \end{bmatrix} \qquad \textbf{(Eq 4-10)}$$

PA66-GF-45%

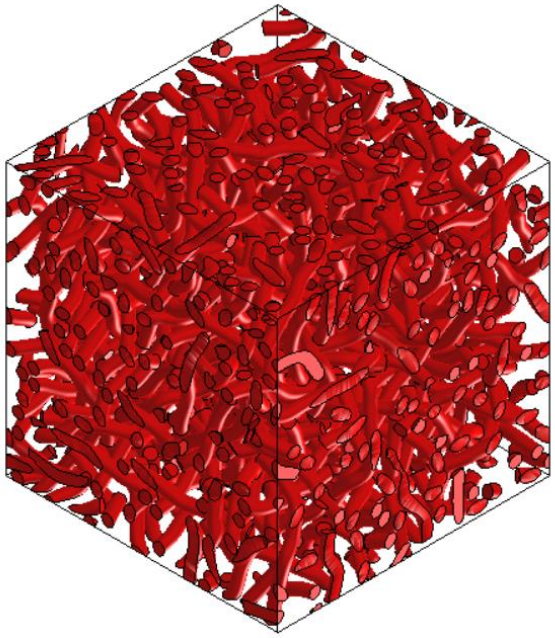
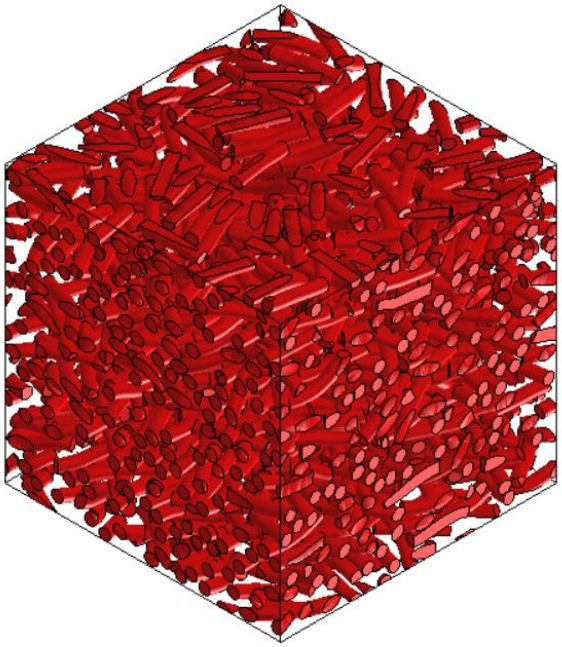


Figure 4.18 RVE of PA 66-GF-45

$$\begin{pmatrix} E_{11} \\ E_{22} \\ E_{33} \end{pmatrix} = \begin{bmatrix} 9613.42 \\ 6189.88 \\ 6091.71 \end{bmatrix} \qquad \textbf{(Eq 4-11)}$$

PA66-GF-60%



$$\begin{pmatrix} E_{11} \\ E_{22} \\ E_{33} \end{pmatrix} = \begin{bmatrix} 8501.58 \\ 6144.86 \\ 6139.1 \end{bmatrix} \qquad \textbf{(Eq 4-12)}$$

E_{11}, E_{22}, E_{33} values are compared for all materials as given in following figure.

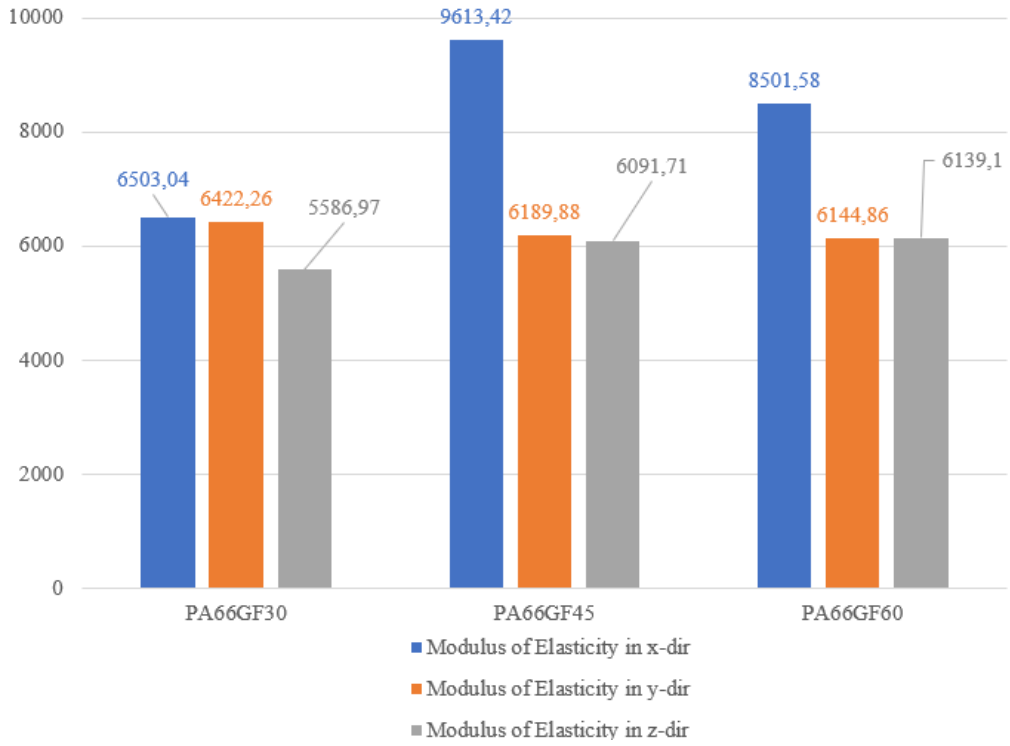


Figure 4.19 Comparison of Elastic Modulus of PA66-GF-30%, PA66-GF-45% and PA66-GF-60%

According to obtained values when fiber volume ratio increases then elastic moduli are also increased. Elastic moduli in directions depends on the fiber orientation. As seen in RVE figures above, fibers are oriented randomly.

According to randomly oriented fibers which is examined in this section, average Elastic Modulus of materials are obtained from equation as follows:

$$\boldsymbol{E_{avg} = \frac{E_x + E_y + E_z}{3}} \qquad \textbf{(Eq 4-13)}$$

By using this formula, average Elastic Modulus are calculated as follows.

$$E_{PAGF30} = 6170.75\ MPA$$

$$E_{PAGF45} = 7298.33\ MPA$$

$$E_{PAGF60} = 6928.52\ MPA$$

4.2.2 PA 66 and Carbon Fiber

Stiffness matrices and modulus of elasticity of 3 types of materials (SFRP) will be obtained at ideal conditions which RVE size is 50x50x50 μm^3, 50x50x50 number of voxel element, isotropic material behavior, fiber diameter is 0.001 mm.

Table 4.3 PA 66-Carbon Fiber Materials

New Material	Fiber Material	Matrix Material	Volume Fraction	Homogenization
PA66-CF-30%	Carbon Fiber	PA 66	30	Mori-Tanaka-First Order
PA66-CF-45%	Carbon Fiber	PA 66	45	Mori-Tanaka-First Order
PA66-CF-60%	Carbon Fiber	PA 66	60	Mori-Tanaka-First Order

Modulus of elasticity in 3D of three different materials are given as follows:

PA66-CF-30%

$$\begin{pmatrix} E_{11} \\ E_{22} \\ E_{33} \end{pmatrix} = \begin{bmatrix} 8498.13 \\ 10141.7 \\ 6744.82 \end{bmatrix} \qquad \textbf{(Eq 4-14)}$$

PA66-CF-45%

$$\begin{pmatrix} E_{11} \\ E_{22} \\ E_{33} \end{pmatrix} = \begin{bmatrix} 10570.9 \\ 6044.63 \\ 6406.07 \end{bmatrix} \qquad \textbf{(Eq 4-15)}$$

PA66-CF-60%

$$\begin{pmatrix} E_{11} \\ E_{22} \\ E_{33} \end{pmatrix} = \begin{bmatrix} 9512.31 \\ 11408.7 \\ 7085.53 \end{bmatrix} \qquad \textbf{(Eq 4-16)}$$

Global density of PA66-CF-60 % is calculated as 1536 kg/m^3.

E_{11}, E_{22}, E_{33} values are compared for all materials as given in following figure.

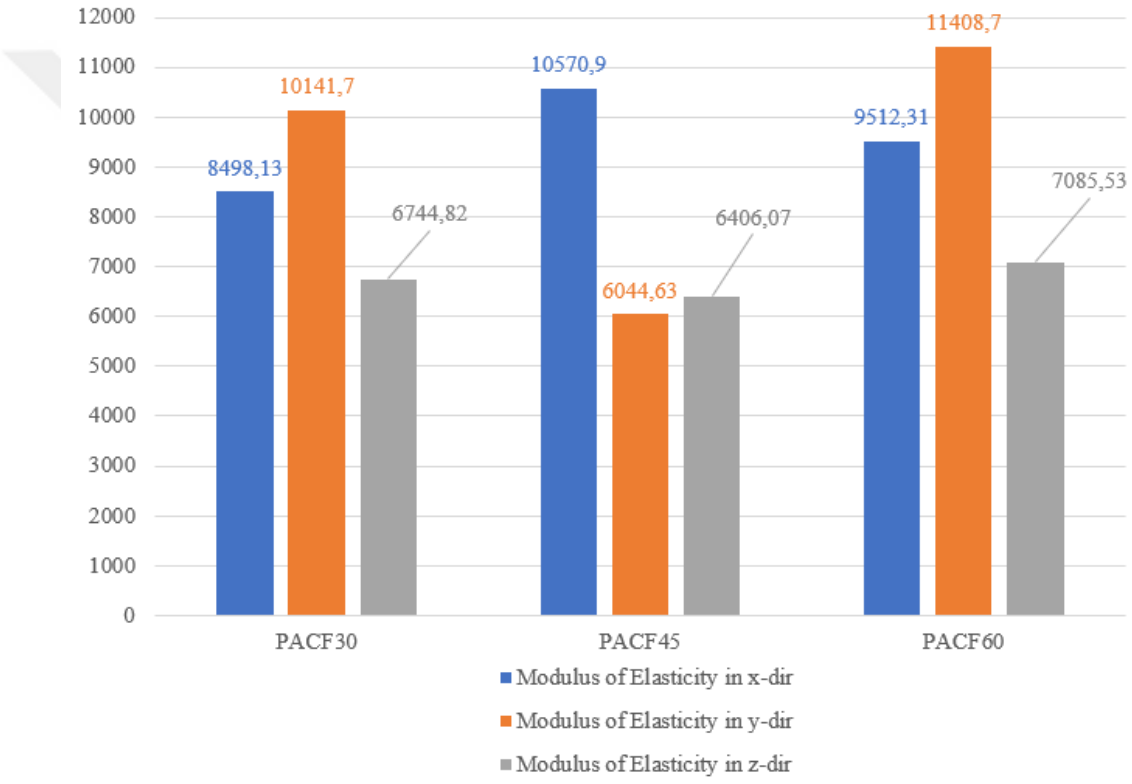


Figure 4.20 Comparison of Elastic Modulus of PA66-CF-30%, PA66-CF-45% and PA66-CF-60%

Considering two graphs which are Figure 4.20 and Figure 4.21 then we can obtain following list:

- Elastic modulus of material is affected by fiber volume fraction.
- Stiffness of material is affected by fiber volume fraction.

- Carbon reinforcement has high elastic modulus, hence materials which include carbon fiber have high Young Modulus values.
- Average value of elasticity moduli is calculated and will be used.

$$E_{PACF30} = 8461.55\ MPa$$

$$E_{PACF45} = 7673.87\ MPa$$

$$E_{PACF60} = 9335.52\ MPa$$

4.3 Investigation of Mechanical Properties of Fiber Reinforced POM

4.3.1 POM and Glass Fiber

Stiffness matrices and modulus of elasticity of 3 types of materials (SFRP) will be obtained at ideal conditions which RVE size is 50x50x50 μm^3, 50x50x50 number of voxel element, isotropic material behavior, fiber diameter is 0.001 mm.

Table 4.4 POM-Carbon Fiber Materials

New Material	Fiber Material	Matrix Material	Volume Fraction	Homogenization
POM-GF-30%	Glass Fiber	POM	30	Mori-Tanaka-First Order
POM-GF-45%	Glass Fiber	POM	45	Mori-Tanaka-First Order
POM-GF-60%	Glass Fiber	POM	60	Mori-Tanaka-First Order

Stiffness matrices and 6x6 modulus of elasticity of three different materials are given as follows:

POM-GF-30%

$$\begin{pmatrix} E_{11} \\ E_{22} \\ E_{33} \end{pmatrix} = \begin{bmatrix} 7248.44 \\ 5652.67 \\ 5568.84 \end{bmatrix} \qquad \textbf{(Eq 4-17)}$$

POM-GF-45%

$$\begin{pmatrix} E_{11} \\ E_{22} \\ E_{33} \end{pmatrix} = \begin{bmatrix} 7479.33 \\ 7775.66 \\ 6016.35 \end{bmatrix} \qquad \textbf{(Eq 4-18)}$$

POM-GF-60%

$$\begin{pmatrix} E_{11} \\ E_{22} \\ E_{33} \end{pmatrix} = \begin{bmatrix} 7875.33 \\ 7783.04 \\ 8180.47 \end{bmatrix} \qquad \textbf{(Eq 4-19)}$$

E_{11}, E_{22}, E_{33} values are compared for all materials as given in following figure.

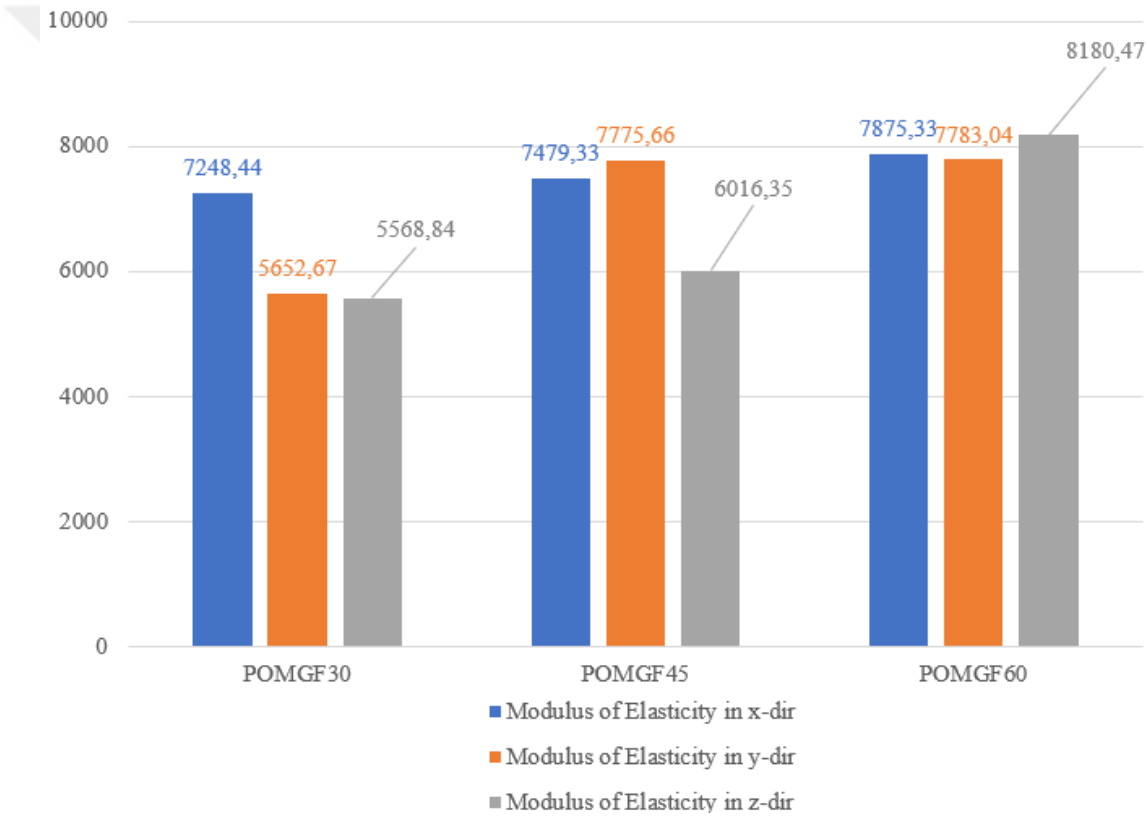


Figure 4.21 Comparison of Elastic Modulus of POM-GF-30%, POM-GF-45% and POM-GF-60%

By using Eq 4.13 formula average Elastic Modulus are calculated as follows.

$$E_{POMGF30} = 6156.65\ MPA$$

$$E_{POMGF45} = 7090.45\ MPA$$

$$E_{POMGF60} = 7946.28\ MPA$$

4.3.2 POM and Carbon Fiber

Stiffness matrices and modulus of elasticity of 3 types of materials (SFRP) will be obtained at ideal conditions which RVE size is 50x50x50 μm^3, 50x50x50 number of voxel element, isotropic material behavior, fiber diameter is 0.001 mm.

Table 4.5 POM-Carbon Fiber Materials

New Material	Fiber Material	Matrix Material	Volume Fraction	Homogenization
POM-CF-30%	Carbon Fiber	POM	30	Mori-Tanaka-First Order
POM-CF-45%	Carbon Fiber	POM	45	Mori-Tanaka-First Order
POM-CF-60%	Carbon Fiber	POM	60	Mori-Tanaka-First Order

Stiffness matrices and 6x6 modulus of elasticity of three different materials are given as follows:

POM-CF-30%

$$\begin{pmatrix} E_{11} \\ E_{22} \\ E_{33} \end{pmatrix} = \begin{bmatrix} 10816.3 \\ 6006.42 \\ 5875.09 \end{bmatrix} \qquad \textbf{(Eq 4-20)}$$

POM-CF-45%

$$\begin{pmatrix} E_{11} \\ E_{22} \\ E_{33} \end{pmatrix} = \begin{bmatrix} 10854 \\ 5873.94 \\ 5848.56 \end{bmatrix} \qquad \textbf{(Eq 4-21)}$$

POM-CF-60%

$$\begin{pmatrix} E_{11} \\ E_{22} \\ E_{33} \end{pmatrix} = \begin{bmatrix} 11310.1 \\ 5997.32 \\ 6359.59 \end{bmatrix} \qquad \textbf{(Eq 4-22)}$$

Global density of the material is 1600 kg/m^3.

E_{11}, E_{22}, E_{33} values are compared for all materials as given in following the figure.

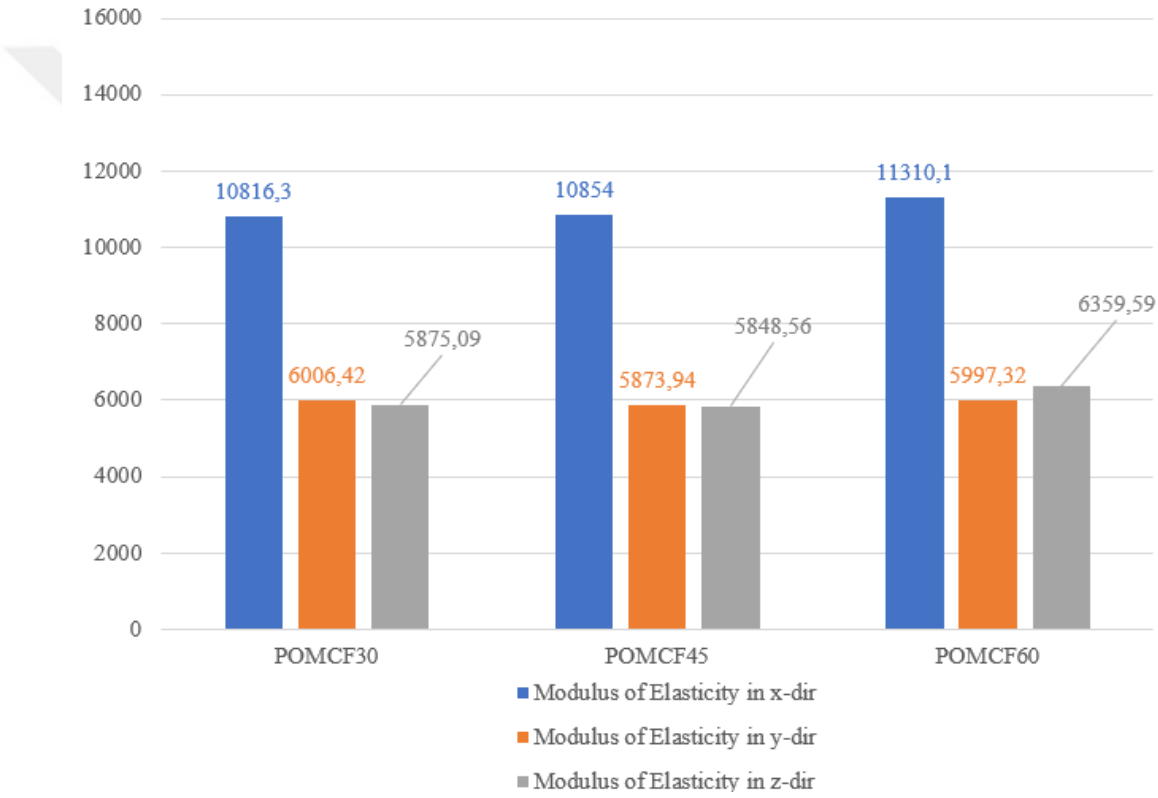


Figure 4.22 Comparison of Elastic Modulus of POM-CF-30%, POM-CF-45% and POM-CF-60%

By using Eq 4.13 formula average Elastic Modulus are calculated as follows.

$$E_{POMCF30} = 7565.94\ MPa$$

$$E_{POMCF45} = 7525.5\ MPa$$

$$E_{POMCF60} = 7889.01\ MPa$$

4.4 Comparison of Obtained Elastic Modulus Values

Using Digimat MF Software, we examine the mechanical properties of thermoplastic polymer materials such as POM, and PA 66. Glass fiber and carbon fiber are selected as reinforced materials. We examine various mixing ratios, specified by volume fractions.

Gears are used to transmit power from one gear to another gear. In these applications, margin of safety, elastic properties of materials are important. Therefore, a high modulus of elasticity should be chosen for high-load transmission." For 3D analysis, carbon fiber reinforcement material is used because it has high modulus of elasticity in three directions and resistant to corrosion and wear. These parameters affect the life of gears or gear meshes.

Effect of RVE size, number of voxel element and material types are examined in section 4.1, 4.2, 4.3, and 4.4. In this part, PA 66 and POM materials are used with carbon fiber reinforced material. The following are the advantages and reasons for the selection are listed below:

- ✓ 50x50x50 RVE size will be used because analysis time and iterations.
- ✓ Number of voxel element is 50x50x50.
- ✓ Carbon fiber reinforced material exhibits high mechanical properties. Hence, carbon fiber will be a reinforced material.
- ✓ 60 % volume fraction will be used because elastic modulus is higher than others.
- ✓ Eq 4-15 and 4-27 will be used to analyze as mechanical properties of materials.
- ✓ Due to the relation between fiber orientation and elasticity moduli, three different orientations are used to detect ideal mechanical properties.

Based on the listed parameters and advantages, POM-CF-60% and PA66-CF-60% materials to be used in analysis. ABAQUS CAE will be used as analysis program. Linear and static analysis will be conducted.

The mechanical behaviors of plastic materials depend on temperature changes. Selected materials which are POM-CF-60 % and PA66-CF-60 % are analyzed. Elastic moduli of materials are examined with respect to temperature variance. Hence, the

results obtained are illustrated. The graphs show how elastic modulus vary with respect to temperature.

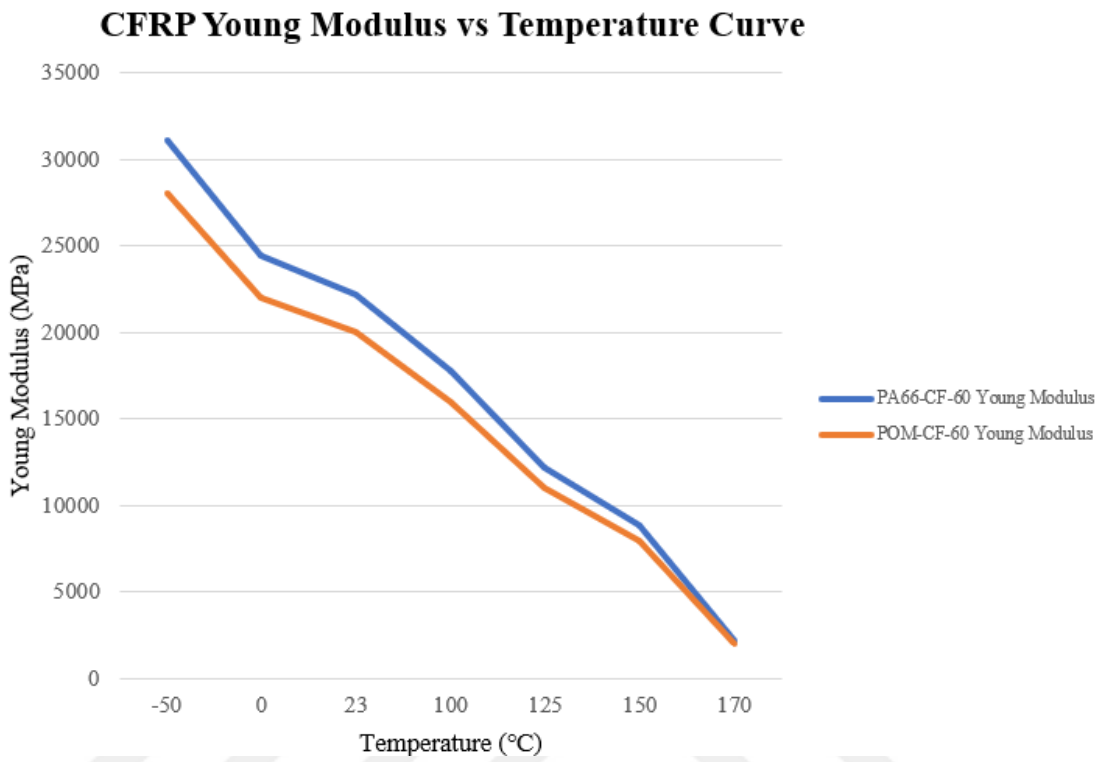


Figure 4.23 Elastic Modulus vs Temperature of PA66-CF-60 % and POM-CF-60

As shown in Figure 4.23, elastic modulus of CFRP materials depend on temperature. When temperature increases the elastic modulus of material decreases. When gears are in contact with each other, because of contact stress and friction to be increases. Hence, at elevated temperatures, the mechanical behavior of gears will decrease. Bending stress at tooth root will not change because it depends on applied load, transmitted power, speed, and gear design parameters. For this reason, the factor of safety will decrease when temperature increases [11]. It is important to analyze gear behavior at elevated temperatures. Elastic modulus values with respect to temperature is listed in Table 4.6 below.

Table 4.6 Materials Elastic Modulus Values with Temperature

Material	Elastic Modulus	Temperature
POM-CF-60 %	29000 MPa	-50 °C
POM-CF-60 %	8000 MPa	140 °C

PA66-CF-60 %	32000 MPa	-50 °C
PA66-CF-60 %	8900 MPa	140 °C

CHAPTER 5

INTRODUCTION TO GEAR THEORIES

Gears which are classified as machine elements are used to transmit power from input (power source) to target output. A gearbox is a system which includes gears, bearings, shafts, splines, etc. There are many gearboxes used industries. Planetary and parallel shafts gearboxes are examples. Gearboxes are used to increase rotational speeds or decrease rotational speeds. Metallic gears are widely used gears in industries.

Gear theory is an important topic to improve gearboxes. Widely known standards are used to design gears such as AGMA, DIN, JIS, JGMA, and ISO. To understand gear theory and to design a gear, gear parameters to be examined.

In this study, spur gear, which is widely used gear for aerospace will be examined. Spur gears are used for high rotational speed applications. Firstly, nomenclature of spur gear teeth is introduced. Figure 5.1 is given below to show gear parameters.

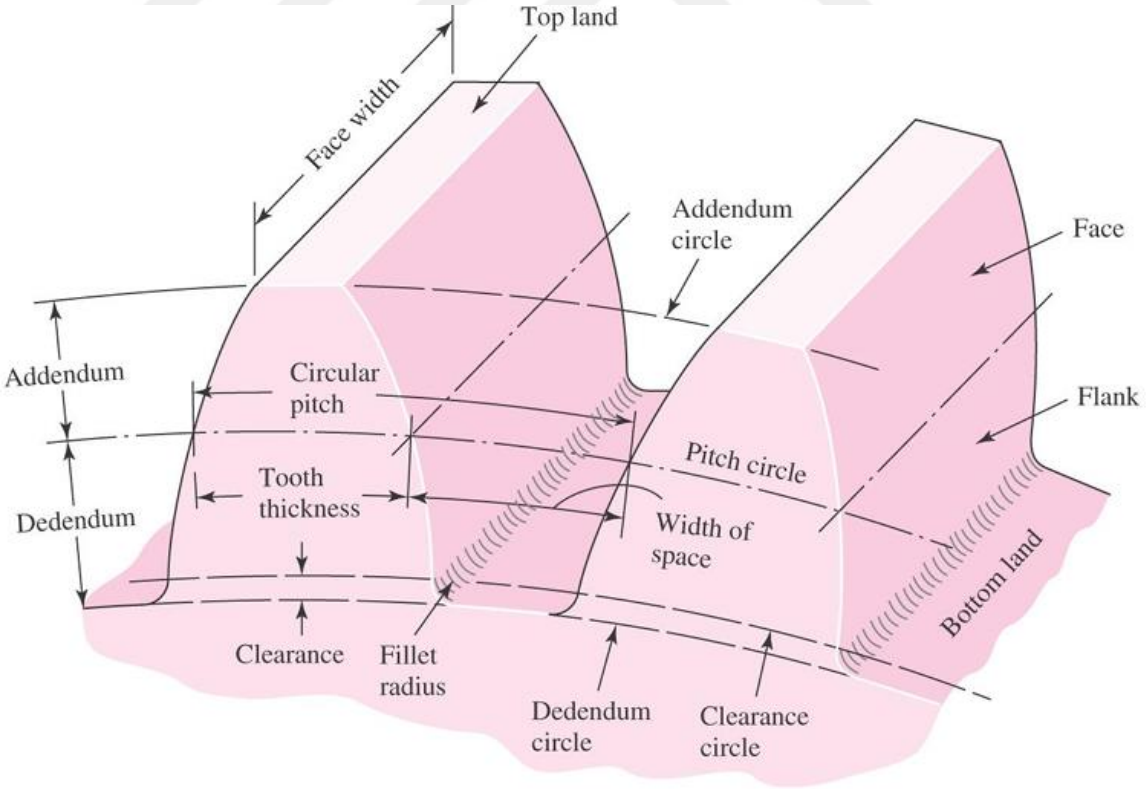


Figure 5.1 Spur Gear Nomenclature [25]

The pitch circle means that a theoretical circle upon which all calculations are usually based.

The circular pitch is the distance, measured on the pitch circle, from a point on one tooth to a corresponding point on an adjacent tooth.

The module (m) is the ratio of the pitch diameter to the number of teeth.

The addendum (a) is the radial distance between the top land and the pitch circle.

The dedendum (b) is the radial distance from the bottom land to the pitch circle.

The clearance circle is a circle that is tangent to the addendum circle of the mating.

The pressure angle is between the line of action and tangent to the pitch circles.

The backlash is the amount by which the width of a tooth space exceeds the thickness of the engaging tooth measured on the pitch circles [24, 25].

When surfaces move each other, such as roll and/or slide movements, gears produce constant angular velocity ratio. This situation is called Conjugate Action. All gear meshes are available for this law. Forces are transmitted on the line of action which is normal to the contacting surfaces. In this law, angular velocity is inversely proportional to the radii to Point P, the pitch point. This law is also known as the Law of Gearing.

$$\left|\frac{\omega_1}{\omega_2}\right| = \frac{r_2}{r_1} \qquad \textbf{(Eq 5-1)}$$

Conjugate action law which is defined in [16] is illustrated in Figure 5.2.

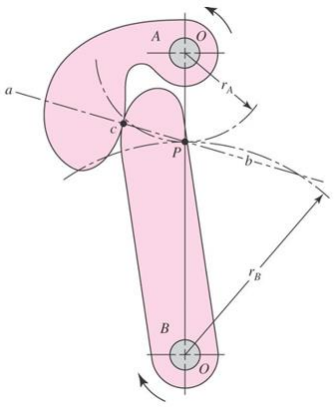


Figure 5.2 Conjugate Action

The most common conjugate profile is involute profile. The most commonly known method is that involute profile can be generated by unwrapping a string from a cylinder, keeping the string taut and tangent to the cylinder. Involute profile is shown in Figure 5.3 [24, 25, 26].

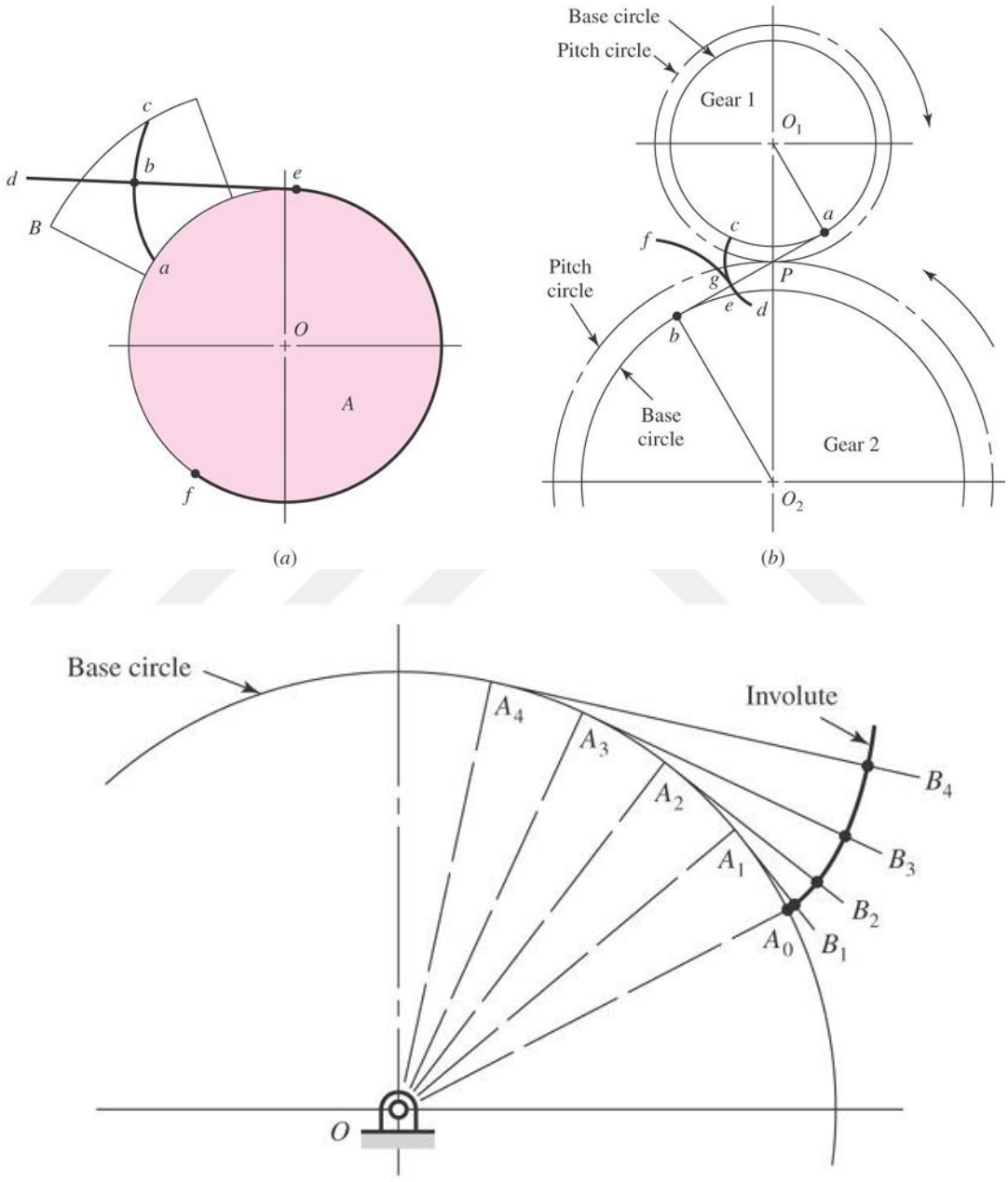


Figure 5.3 Involute Profile Method [24]

Gear layout circles are illustrated below.

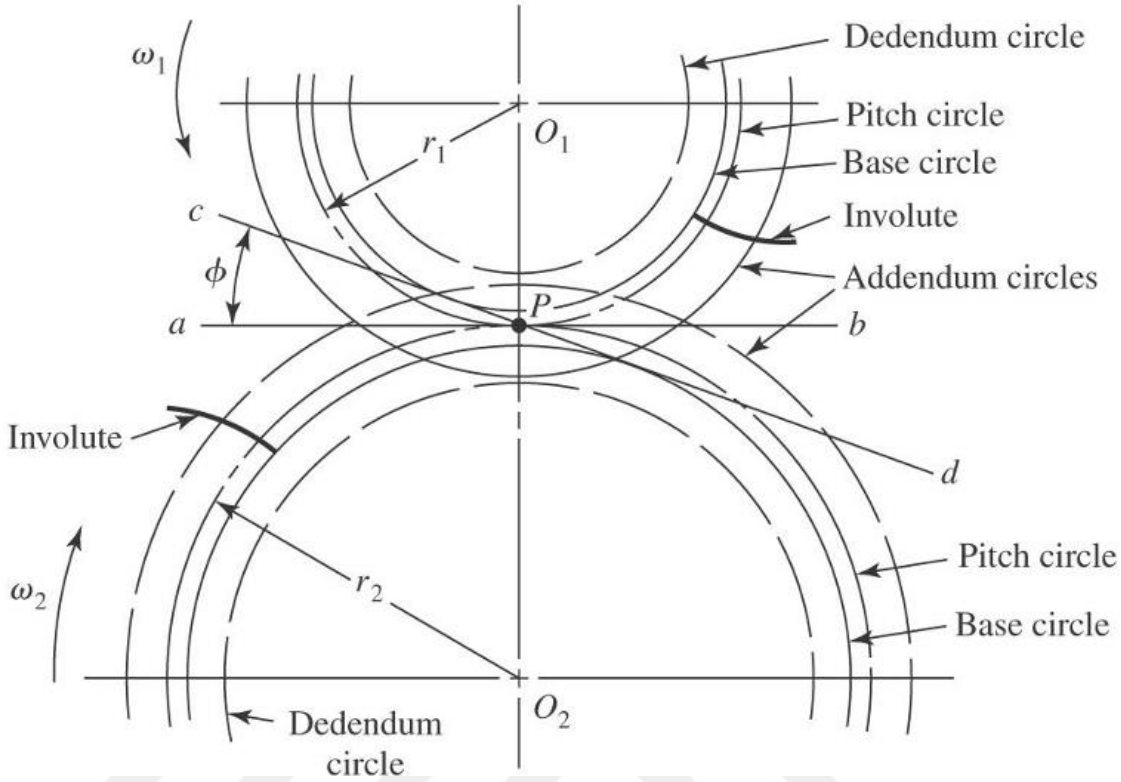


Figure 5.4 Gear Layout and Related Terms [25]

Sequence of gear layout is listed below [25]:

- Pitch circles are in contact.
- Pressure line at desired pressure angle.
- Base circles tangent to pressure line
- Involute profile from base circle
- Cap teeth at addendum circle at 1/P from pitch circle.
- Root of teeth at dedendum circle at 1.25/P from pitch circle
- Tooth spacing from circular pitch, $p = \pi/P$

Contact Ratio is another important gear parameter to use in design processes. It is shown in Figure 5.5 [25].

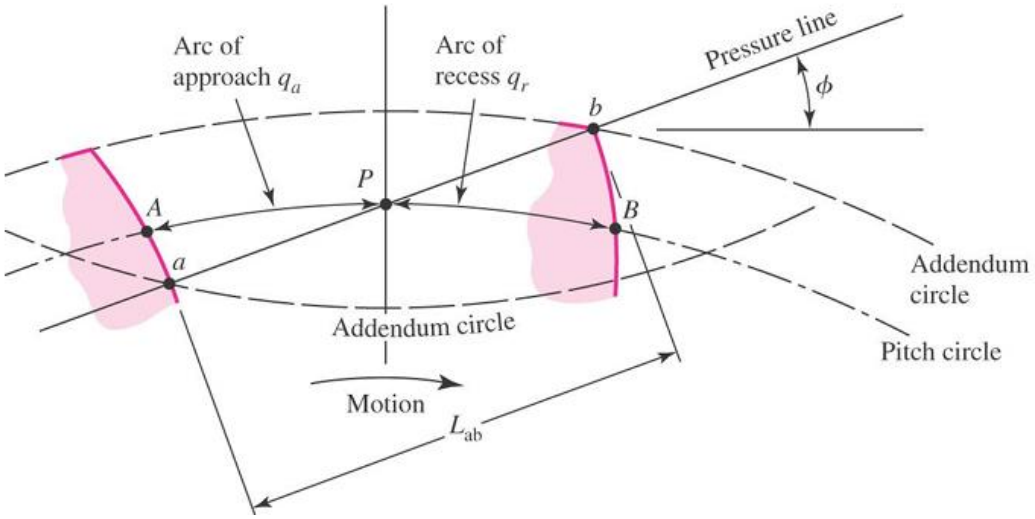


Figure 5.5 Contact Ratio [25]

Gear analysis is important to conduct gear design and development phases. There are many diverse types of analysis which are root bending stress, tooth contact stress, lubrication analysis, micro modification analysis, alignment, vibration, and noise analysis, etc. In this thesis, root bending stress to be examined with effect of materials. Theoretically, bending stress depends on applied load, gear form factor, stress correction factor, load factor, etc. All the parameters of root bending stress are examined in the following section. VDI 2736 will be used as a standard [24, 25].

The region is known the double tooth pair contact region starts Point A which is the meshing of gears starts at Point A, the second pair is in contact at the point of “D.”. When the first gear pair reached point “B,” the second gear pair contact ends. Thus, this point is specifically named the Lowest Point of Single Tooth Contact (LPSTC). After point “B,” one tooth pair continues to contact until point “D.” The region between B – D is defined as the single tooth contact region. Also, the point “D” is named as the Highest Point of Single Tooth Contact (HPSTC). After the HPSTC, the meshing process continues with two gear pairs until the end of the contact point “E.” [26, 27]. Line of action of gears and related points are illustrated in Figure 5.6.

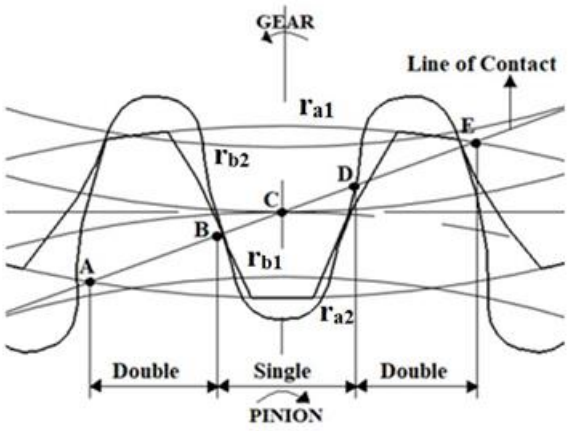


Figure 5.6 Line of Action of Spur Gears [27]

When gears have meshed each other, the following figure can be obtained.

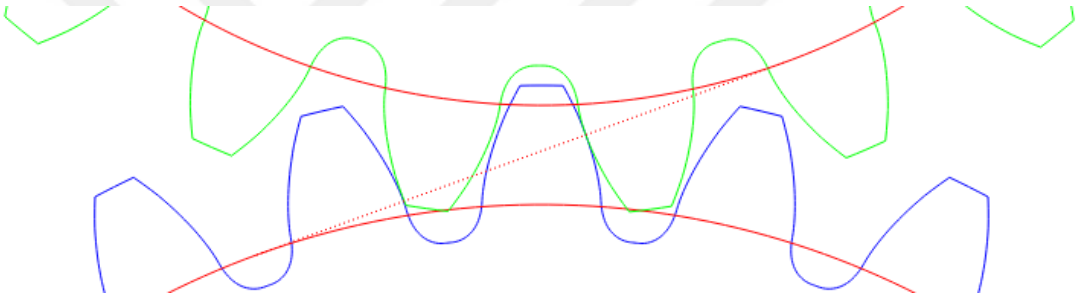


Figure 5.7 Gear Mesh

In this figure, dotted Red Line represents line of action of gears. It is also called as path of action. Red arcs are base diameters of gears. Green and blue profiles are gear tooth profiles.

5.1 Examination of Theoretical Root Bending Stress

The load-related increase in the contact length and ratio of gears made of polymers has a significant influence on root stress. The maximum tooth root stress σ_F is calculated by using VDI 2736 [28].

$$\sigma_F = \frac{F_t}{b m_n} Y_{Fa} Y_{Sa} Y_\varepsilon Y_\beta K_F \qquad \textbf{(Eq 5-2)}$$

Where;

F_t is the nominal tangential load at the cylindrical gear,

b is the tooth width,

m_n is the module,

Y_{Fa} is the form factor,

Y_{Sa} is the stress correction factor,

Y_ε is the contact ratio factor,

Y_β is the helix angle factor,

K_F is the factor for tooth root load.

Calculation of tangential load:

$$F_t = \frac{2000 \times T}{d_p} \qquad \textbf{(Eq 5-3)}$$

Where:

T is applied torque in Nm to provide from $P\ (kW) = T \times n/9548$ (n is speed of gear in rpm)

d_p is pitch diameter of cylindrical gear.

Calculation of Stress Correction Factor

$$Y_{Sa} = (1.2 + 0.13L_a) \times q_s^{\left(\frac{1}{1.21+\frac{2.3}{L_a}}\right)} \qquad \textbf{(Eq 5-4)}$$

Where;

$L_a = S_{Fn}/h_{fa}$

Calculation of Contact Ratio Factor;

$$Y_\varepsilon = 0.25 + \frac{0.75}{\varepsilon_\alpha} \qquad \textbf{(Eq 5-5)}$$

Where;

ε_α is transverse contact ratio.

Calculation of Form Factor;

$$Y_{Fa} = \frac{\frac{6h_{Fa}}{m_n}\cos\alpha_{Fan}}{\left(\frac{S_{Fn}}{m_n}\right)^2\cos\alpha_n} \qquad \textbf{(Eq 5-6)}$$

Calculation of Tooth Root Load Factor;

$$K_F = K_A K_V K_{F\beta} K_{F\alpha} \qquad \textbf{(Eq 5-7)}$$

5.2 Numerical Calculation of Root Bending Stress

In this section numerical results to be provided via KISSSoft advanced gear design program. Gear parameters are selected as follows:

Module (m_n) = 2

Pressure angle (α) = 20 °

Number of teeth of pinion (z_p) = 27

Number of teeth of gear (z_g) = 23

Thickness (b) = 10 mm

Profile shift coefficient (x^*) = 0

Dedendum Coefficient $(h_f) = 1.25$

Addendum Coefficient $(h_a) = 1.00$

No lubrication

According to literature reviews, the number of teeth is selected based on hunting teeth. Hunting teeth in gear pairs is widely selected methods for gear design which is means that there is no common denominator between the number of pinion and gear teeth. For this reason, every tooth of the pinion will mesh with every slot of the gear. After all teeth and slots have been rolling with each other, the cycle repeats. The cycle

repetition happens after the gear performs a few revolutions, equal to the number of pinion teeth. This is of course also true if the pinion performs several revolutions equal to the number of gear teeth. The number of revolutions to achieve “one tooth hunting sequence” is independent from the fact if the numbers of teeth are prime numbers or if simply one number is even and the other number is odd [29].

Module is also selected based on literature reviews. Engineers and scientists try to improve plastic gear performance, and many studies are done with using 1 module.

14.5°, 20°, and 25° pressure angles are widely selected angles for plastic gears. However, 20° is chosen because stronger tooth shape, reduced undercutting. 25° pressure angle has the highest load carrying ability but is more sensitive to center distance variation and hence less quiet.

Thickness and profile shift coefficient are selected to provide easy calculation.

According to literature reviews, the same material for both pinion and gear shows better performance [3, 9, 24, 25]. Several types of materials will be analyzed. POM-POM and PA 66-PA 66 to be examined. These materials show higher mechanical and thermal properties, and these are commonly used plastics in industry.

5.2.1 POM-POM Gear Pair Numerical Analysis

Gears are designed by using KissSoft program via gear design parameters which are mentioned above. The following figure is shown to gear design screen. POM is selected for both pinion and gear.

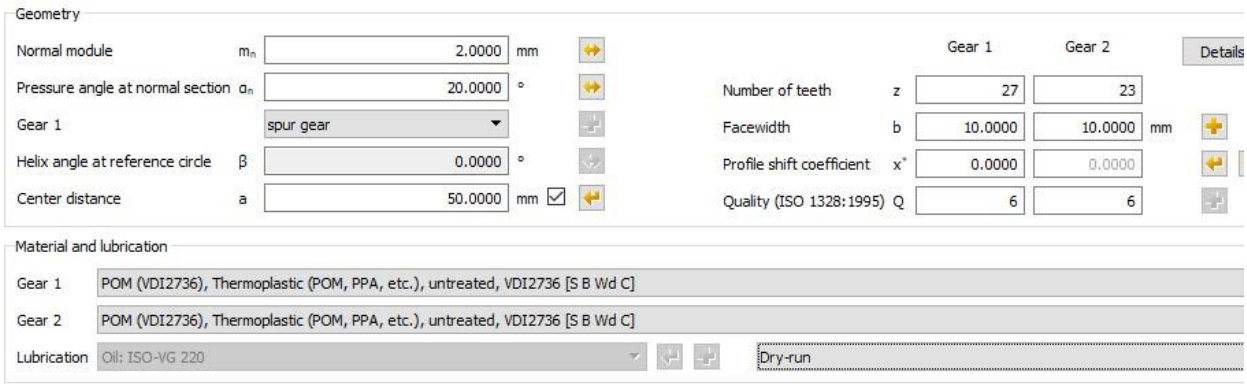


Figure 5.8 Design Parameters of POM-POM Gear Mesh

3 kW power is transmitted to gear pair with 5000 rpm rotational speed. Hence, applied torque can be calculate as 5.73 Nm. Tooth root and flank temperatures are 70 ℃.

Gear pair system, tooth forms of pinion and gear are given in Figure 5.9.

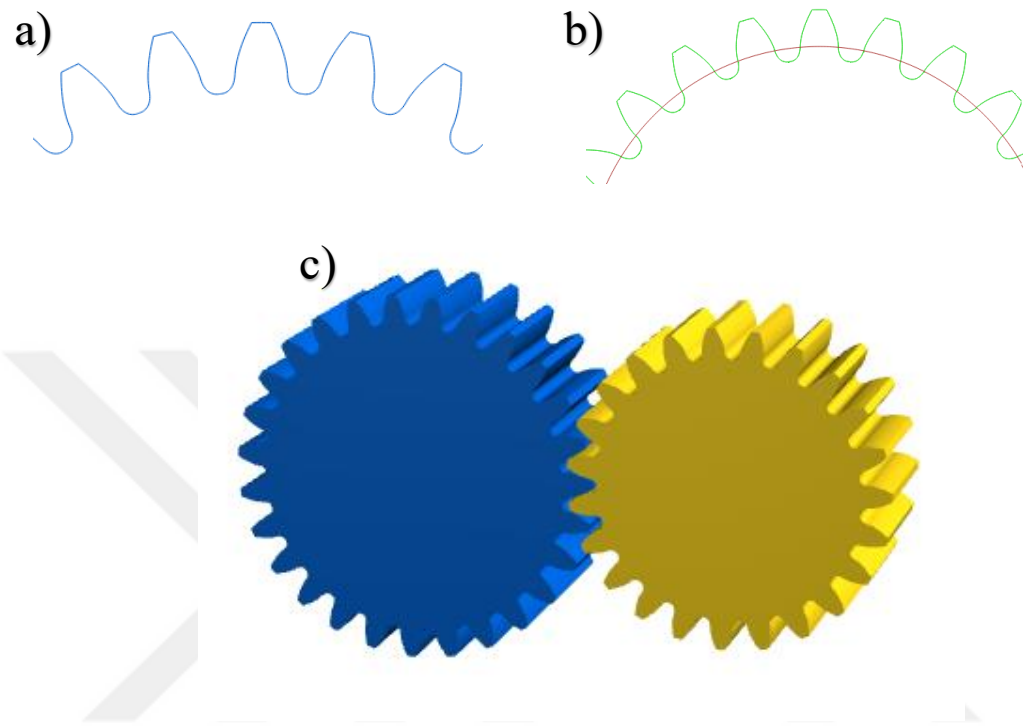


Figure 5.9 a) Pinion Tooth Form b) Gear Tooth Form c) System Isometric View

VDI 2736:2013 (modified YF Method B) is selected as calculation method. YF Method B is generally used for constant loads [5, 26]. Dynamic factor and transverse load factor are chosen as 1.0. After analysis is done, following results are obtained:

Table 5.1 POM-POM Gear Pair Analysis Results (@ 50 ℃)

Results	Pinion	Gear
Radial Force	77.2 N	77.2 N
Normal Force	225.8 N	225.8 N
Root Stress	39.25 MPa	39.60 MPa
Nominal Contact Stress	45.66 MPa	45.66 MPa
Root Stress Safety	1.299	1.288
Tooth Deformation	127.589 µm	127.589 µm

Also, due to the elasticity of plastic materials depending on temperature, analysis needs to examine with the different temperature conditions. Table 5.2 is given to show stress values of gear at 100 ℃.

Table 5.2 POM-POM Gear Pair Analysis Results (@ 100 ℃)

Results	Pinion	Gear
Radial Force	77.2 N	77.2 N
Normal Force	225.8 N	225.8 N
Root Stress	39.25 MPa	39.60 MPa
Nominal Contact Stress	45.66 MPa	45.66 MPa
Root Stress Safety	0.357	0.354
Tooth Deformation	176.839 µm	176.839 µm

Specific sliding which can be used to reduce flash temperature and contact temperature is shown in Figure 5.10. As mentioned in Section 5, line of action is the common tangent line of the base circles of a pair of involute gears in contact. When a pair of involute gears is meshed, the contact occurs on the line of action and the contact point moves along the line of action.

In this gear mesh (z_p is 27 and z_g is 23) sliding velocities are not same and C point which is seem in following figure is not at the middle of line because of differences of number of gears.

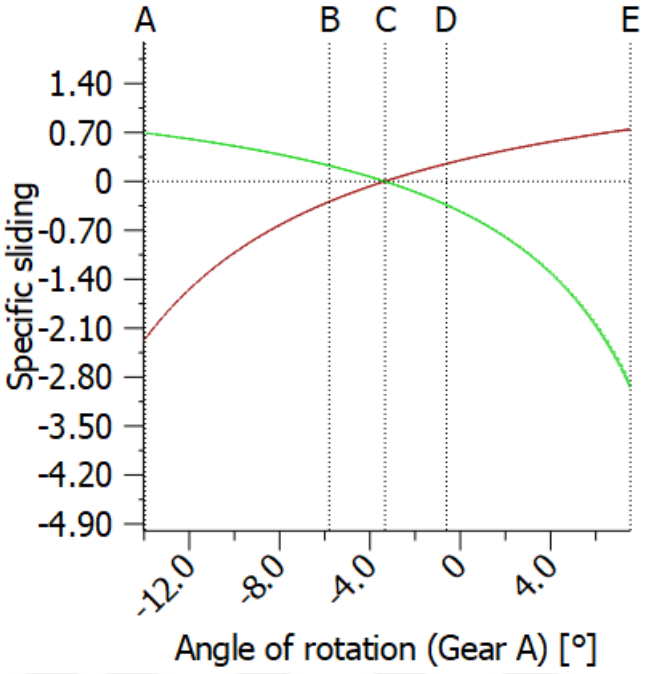


Figure 5.10 Specific Sliding of POM-POM Gear Mesh

Heat development curve which to be used to determine thermal analysis is given in Figure 5.11.

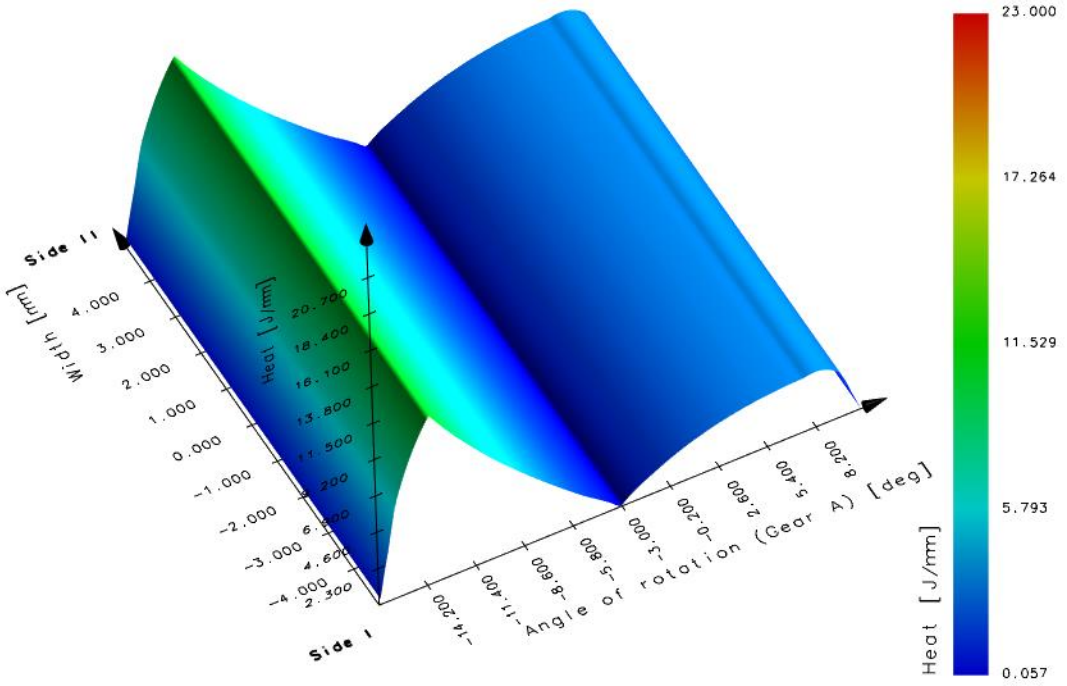


Figure 5.11 3D Heat Development Curve of POM-POM Gear Mesh

Tooth root stress is provided in KissSoft FEM module. 3D tooth root stress distribution is illustrated in Figure 5.14.

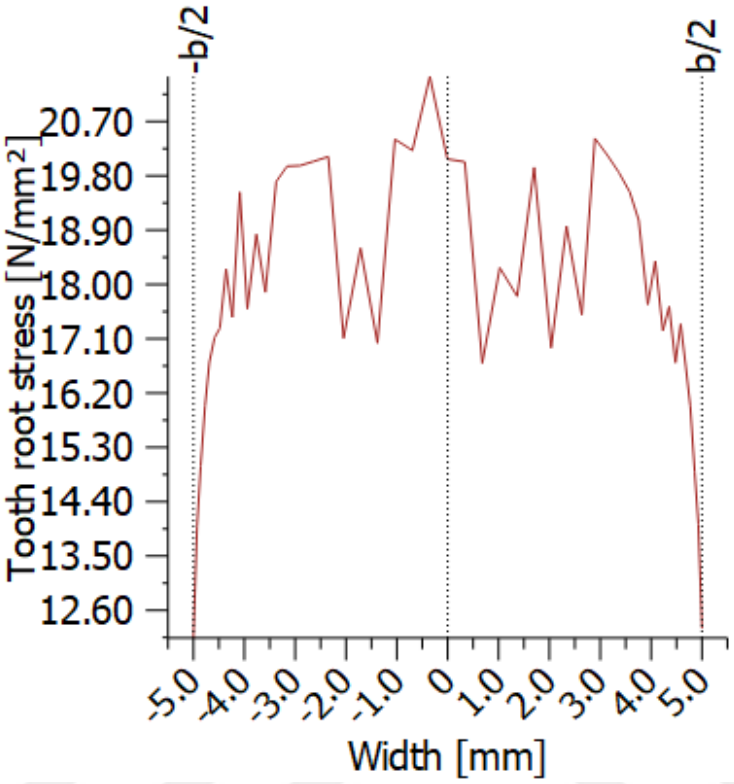


Figure 5.12 3D Tooth Root Stress vs Width Curve of POM-POM Gear Mesh

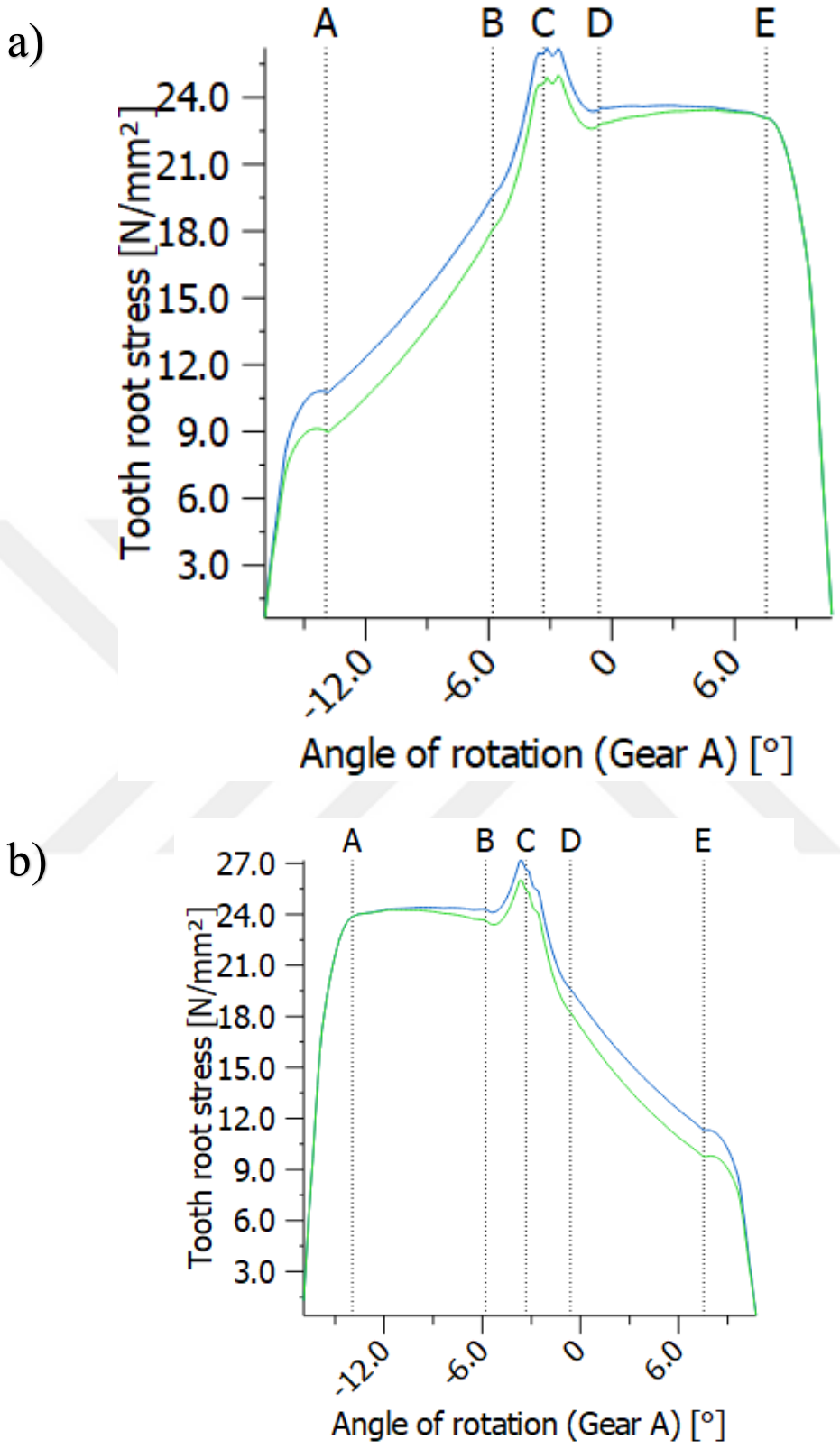


Figure 5.13 Tooth Root Stress vs Width Curve of POM-POM Gear Mesh a) Pinion b) Gear

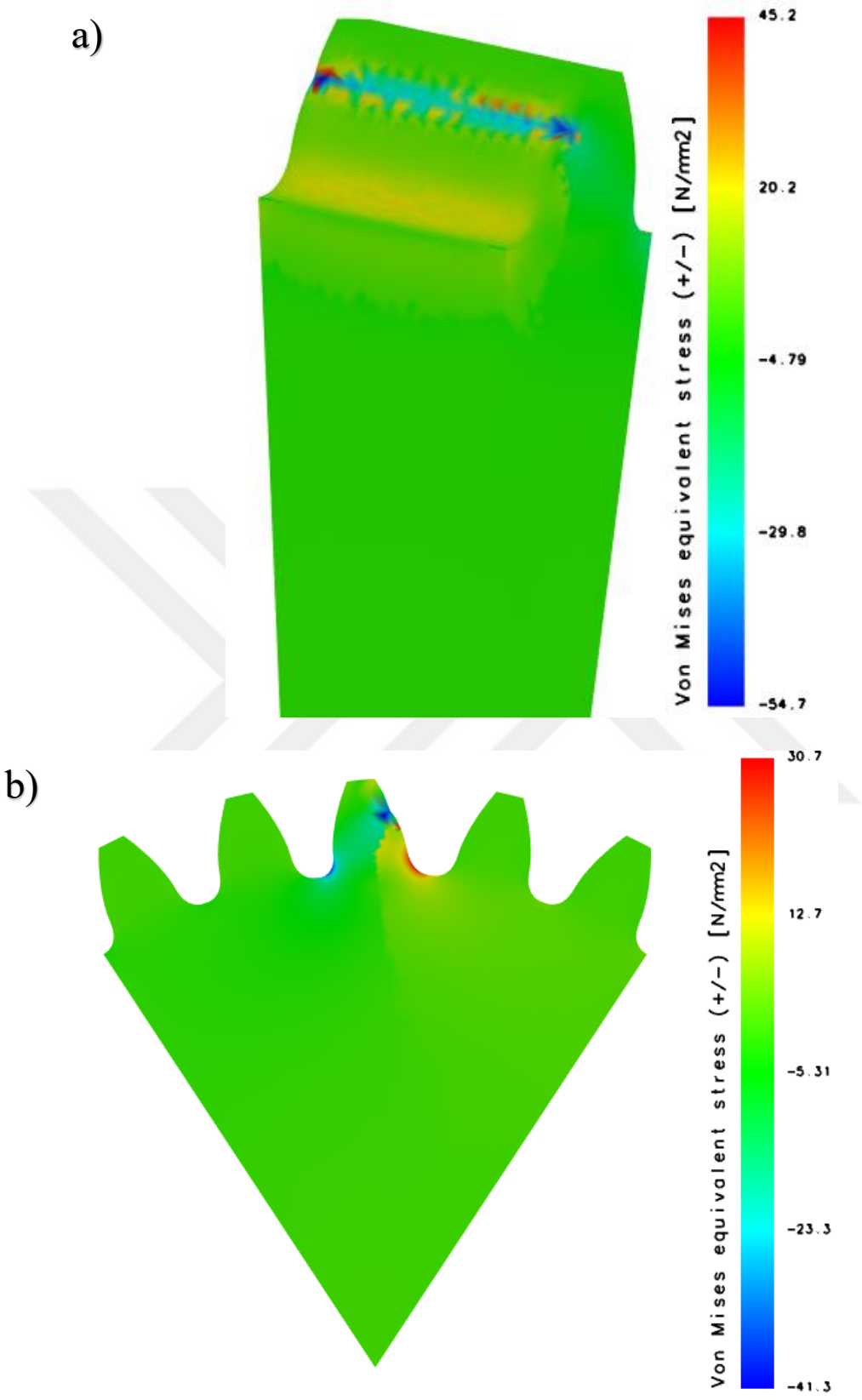


Figure 5.14 3D Tooth Root Stress Distribution of POM-POM Gear Mesh a) Total Stresses b) Root Bending Stress

Three different analyzes are conducted to evaluate the stresses values for gear meshes. Different temperature conditions are examined. Because of POM elastic modulus values depends on temperature means that elastic modulus of POM at 100°C is lower than 50 °C. Due to this state, when gear surface temperature increases because of transmitted loads, then gear safeties will decrease. Because of 120°C temperature is not suitable for POM material, the analysis is not conducted. There is real data in the material database.

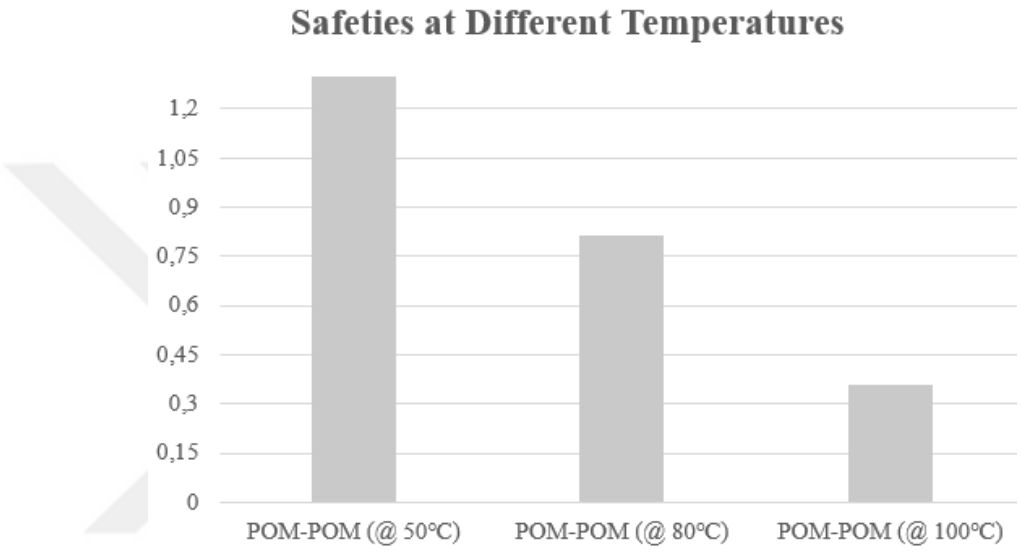


Figure 5.15 Comparison of Safeties of POM at Different Temperatures

5.2.2 PA66 - PA66 Gear Pair Numerical Analysis

In this part, gear pair will be examined which is made of PA 66 material. Gear design parameters and load conditions are set which are same values with Chapter 5.2.1. Temperature conditions are set as 50 °C.

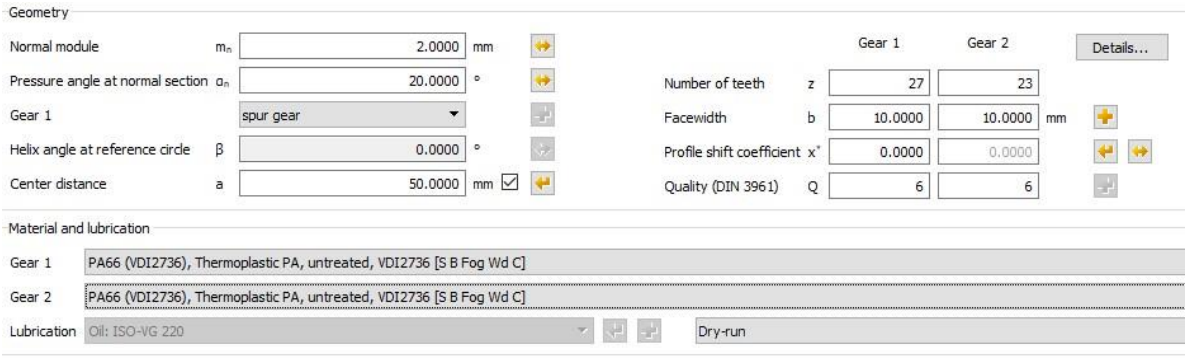


Figure 5.16 Design Parameters of PA 66-PA 66 Gear Mesh

Obtained results from VDI 2736:2013 calculation method is listed in Table 5.3. Also, due to the elasticity of plastic materials depends on temperature, analysis needs to examine with the different temperature conditions. Table 5.4 is given to show stress values of gear at 100 ℃.

Table 5.3 PA66-PA66 Gear Pair Analysis Results (@ 50 ℃)

Results	Pinion	Gear
Radial Force	77.2 N	77.2 N
Normal Force	225.8 N	225.8 N
Root Stress	39.25 MPa	39.60 MPa
Nominal Contact Stress	25.33 MPa	25.33 MPa
Root Stress Safety	1.447	1.431
Tooth Deformation	397.887 µm	397.887 µm

Table 5.4 PA66-PA66 Gear Pair Analysis Results (@ 100 ℃)

Results	Pinion	Gear
Radial Force	77.2 N	77.2 N
Normal Force	225.8 N	225.8 N
Root Stress	39.25 MPa	39.60 MPa
Nominal Contact Stress	25.33 MPa	25.33 MPa
Root Stress Safety	0.397	0.394
Tooth Deformation	795.775 µm	795.775 µm

For the other parameters which are heat development and sliding velocity parameters can be compared. Because of sliding velocity depends on only gear geometry and gear

design parameters, obtained values from PA66-PA66 gear mesh shows same behavior of POM-POM gear mesh.

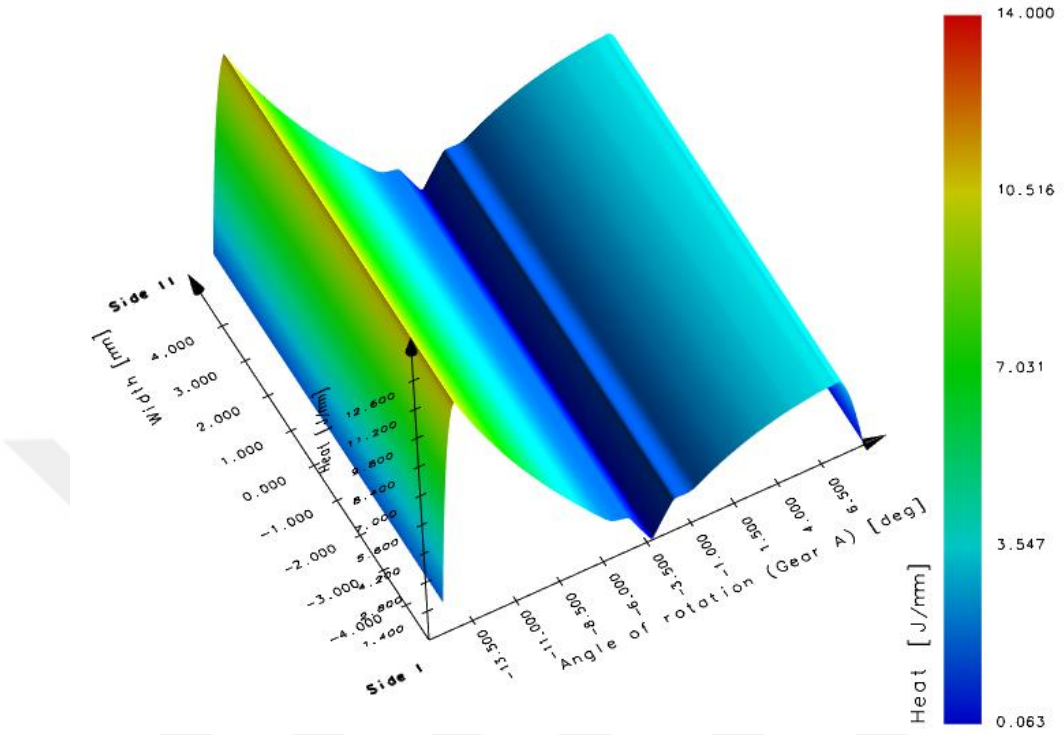


Figure 5.17 3D Heat Development Curve of PA66-PA66 Gear Mesh

Tooth root stress is provided in KissSoft FEM module as given. 3D tooth root stress distribution is illustrated.

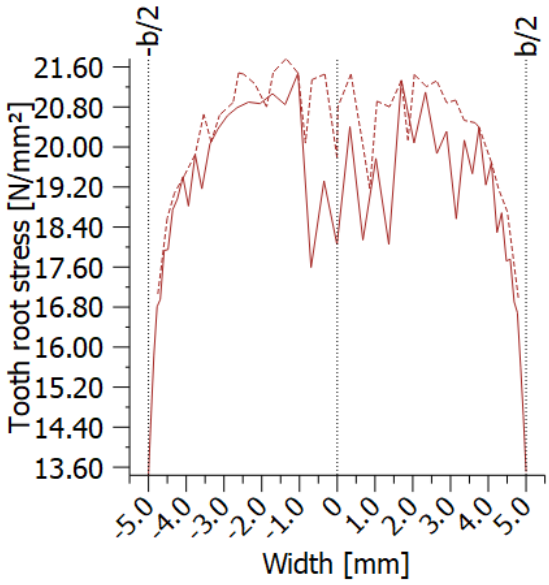


Figure 5.18 3D Tooth Root Stress Curve of PA66-PA66 Gear Mesh

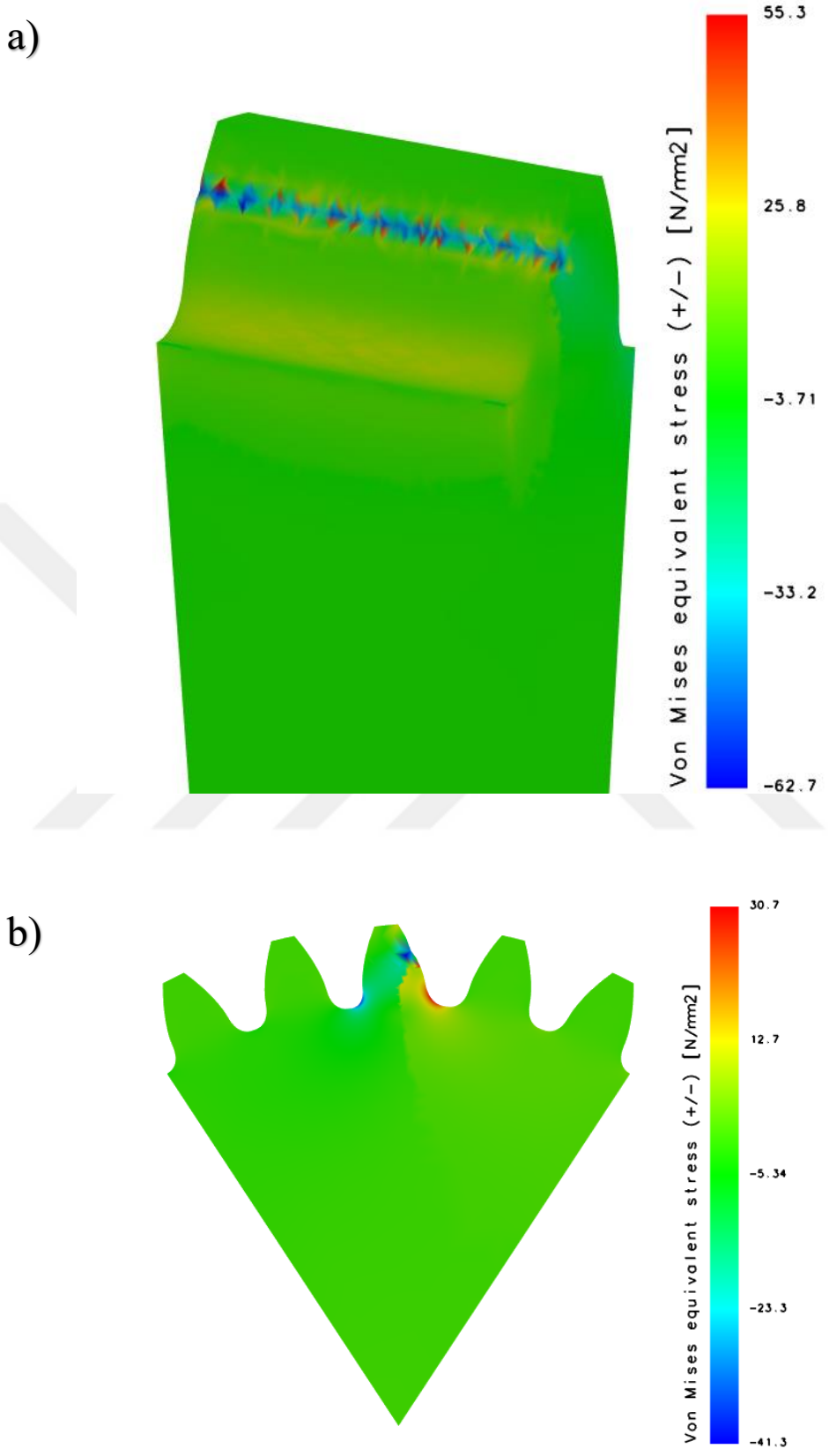


Figure 5.19 3D Tooth Root Stress Distribution of PA66-PA66 Gear Mesh a) Total Stresses b) Root Bending Stress

Three different analyzes are conducted to evaluate the stresses values for gear meshes. Different temperature conditions are examined. Because of PA 66 elastic modulus values depends on temperature means that elastic modulus of PA 66 at 100°C is lower than 50 °C. Due to this state, when gear surface temperature increase because of transmitted loads, then gear safeties will decrease.

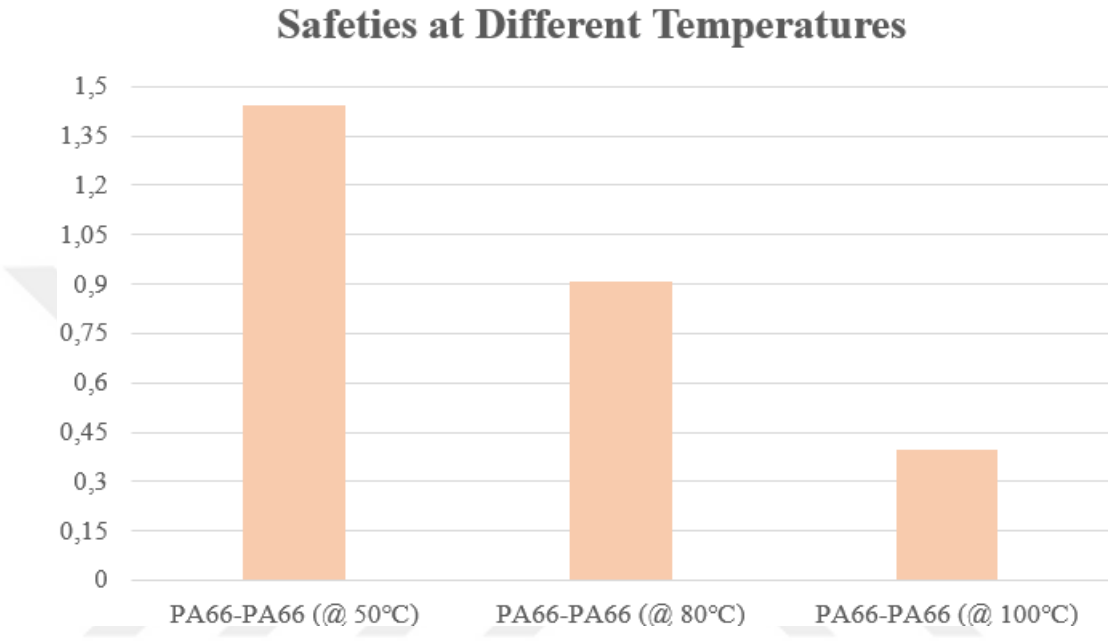
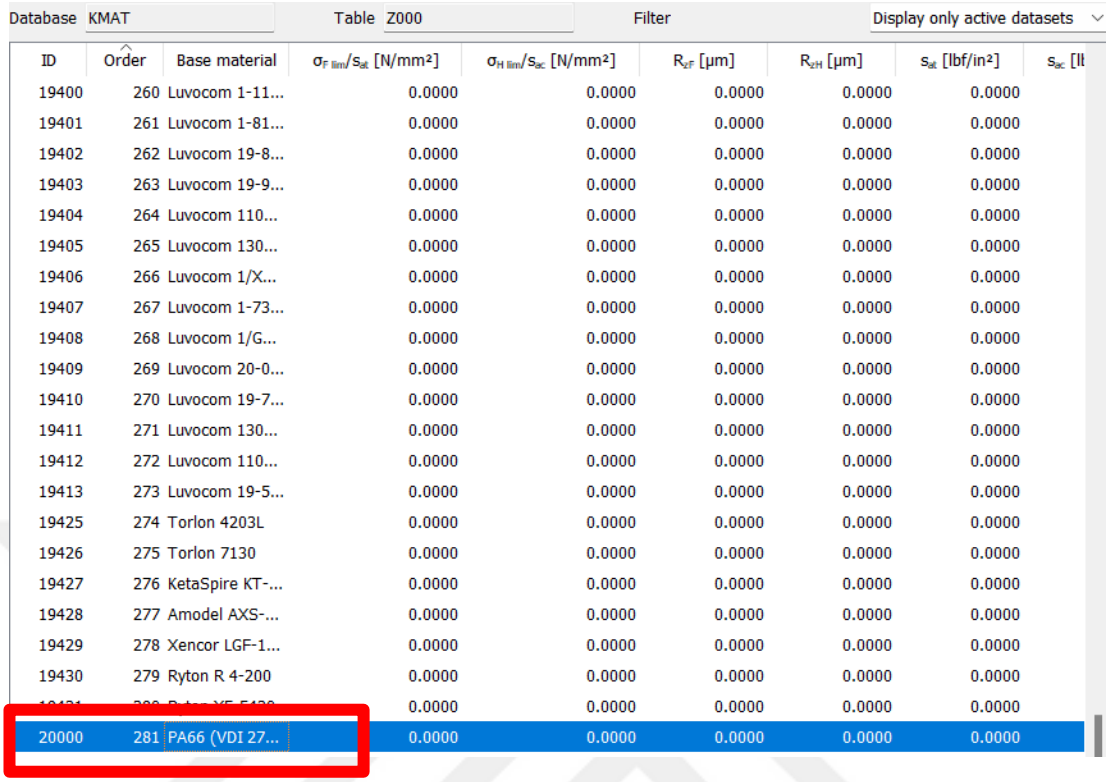


Figure 5.20 Comparison of Safeties of PA 66 at Different Temperatures

5.2.3 PA66-CF-60 % Material KissSoft Studies

The same studies which are mentioned in Chapter 5.2.1 are conducted to analyze fiber reinforced PA 66 material. New material is added in KissSoft material library as given in figure below. Material properties are set from previous studies and literature reviews. By using Digimat MF, different mechanical values have been obtained because of fiber size and angles. Three different analyses are conducted to obtain ideal mechanical properties within the scope. Tooth flank strength, tensile strength, elastic modulus, tooth root strength values with respect to temperature and load cycles are calculated and assumed.



Database KMAT | Table Z000 | Filter | Display only active datasets

ID	Order	Base material	$\sigma_{F\,lim}/S_{at}$ [N/mm²]	$\sigma_{H\,lim}/S_{ac}$ [N/mm²]	R_{zF} [µm]	R_{zH} [µm]	s_{at} [lbf/in²]	s_{ac} [lb
19400	260	Luvocom 1-11...	0.0000	0.0000	0.0000	0.0000	0.0000	
19401	261	Luvocom 1-81...	0.0000	0.0000	0.0000	0.0000	0.0000	
19402	262	Luvocom 19-8...	0.0000	0.0000	0.0000	0.0000	0.0000	
19403	263	Luvocom 19-9...	0.0000	0.0000	0.0000	0.0000	0.0000	
19404	264	Luvocom 110...	0.0000	0.0000	0.0000	0.0000	0.0000	
19405	265	Luvocom 130...	0.0000	0.0000	0.0000	0.0000	0.0000	
19406	266	Luvocom 1/X...	0.0000	0.0000	0.0000	0.0000	0.0000	
19407	267	Luvocom 1-73...	0.0000	0.0000	0.0000	0.0000	0.0000	
19408	268	Luvocom 1/G...	0.0000	0.0000	0.0000	0.0000	0.0000	
19409	269	Luvocom 20-0...	0.0000	0.0000	0.0000	0.0000	0.0000	
19410	270	Luvocom 19-7...	0.0000	0.0000	0.0000	0.0000	0.0000	
19411	271	Luvocom 130...	0.0000	0.0000	0.0000	0.0000	0.0000	
19412	272	Luvocom 110...	0.0000	0.0000	0.0000	0.0000	0.0000	
19413	273	Luvocom 19-5...	0.0000	0.0000	0.0000	0.0000	0.0000	
19425	274	Torlon 4203L	0.0000	0.0000	0.0000	0.0000	0.0000	
19426	275	Torlon 7130	0.0000	0.0000	0.0000	0.0000	0.0000	
19427	276	KetaSpire KT-...	0.0000	0.0000	0.0000	0.0000	0.0000	
19428	277	Amodel AXS-...	0.0000	0.0000	0.0000	0.0000	0.0000	
19429	278	Xencor LGF-1...	0.0000	0.0000	0.0000	0.0000	0.0000	
19430	279	Ryton R 4-200	0.0000	0.0000	0.0000	0.0000	0.0000	
			0.0000	0.0000	0.0000	0.0000	0.0000	
20000	281	PA66 (VDI 27...	0.0000	0.0000	0.0000	0.0000	0.0000	

Figure 5.21 KissSoft Library (POM-CF-60 %)

Added material is selected as gear material to analyze gear parameters as given.

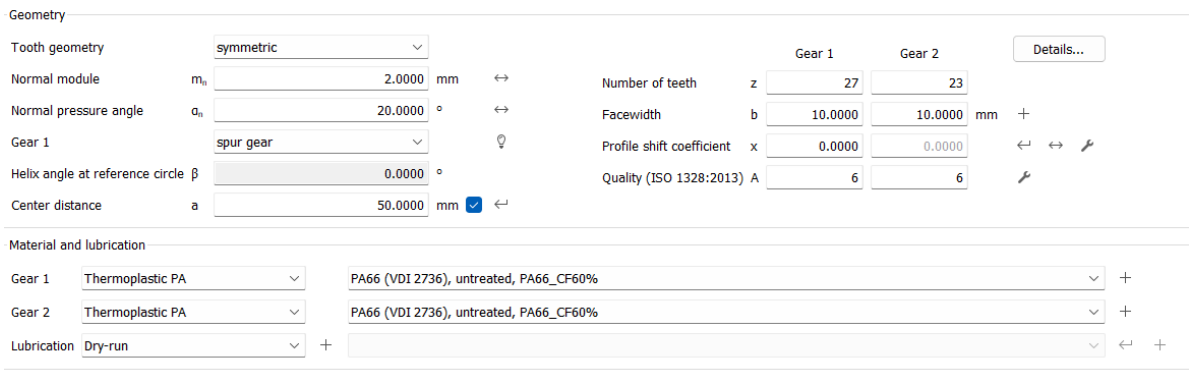


Figure 5.22 Gear Design Parameters for PA66-CF-60 % Gear Mesh

Obtained values are listed following tables.

Table 5.5 PA66-CF-60 % and PA66-CF-60 % Gear Pair Analysis Results (@ 50 ℃)

Results	Pinion	Gear
Radial Force	77.2 N	77.2 N
Normal Force	225.8 N	225.8 N
Root Stress	39.25 MPa	39.60 MPa
Root Stress Safety	5.98	5.87
Tooth Deformation	15.435 µm	15.435 µm

Table 5.6 PA66-CF-60 % and PA66-CF-60 % Gear Pair Analysis Results (@ 100 ℃)

Results	Pinion	Gear
Radial Force	77.2 N	77.2 N
Normal Force	225.8 N	225.8 N
Root Stress	39.25 MPa	39.60 MPa
Root Stress Safety	4.60	4.54
Tooth Deformation	19.89 µm	19.89 µm

Specific sliding which can be used to reduce flash temperature and contact temperature is shown. Because of specific values depend on gear design parameters not material, there is no difference.

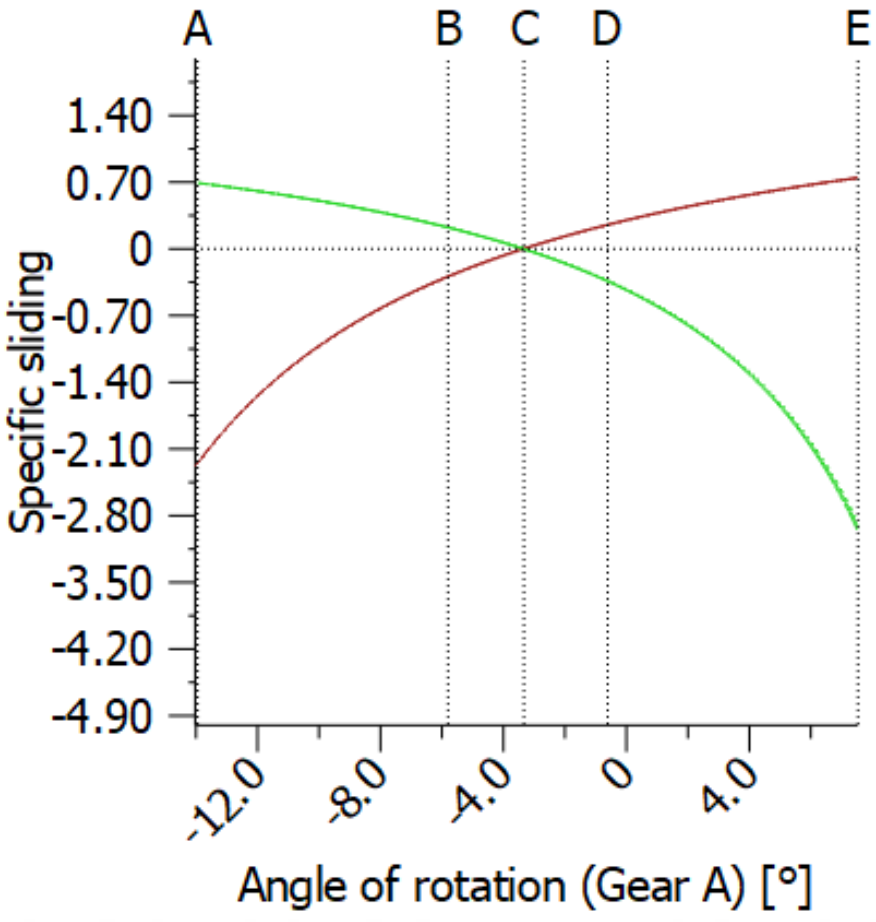


Figure 5.23 Specific Sliding of PA66-CF-60 % and PA66-CF-60 % Gear Mesh

Total stress distribution and root bending stress are illustrated in following figure. Due to the added reinforced fiber material in plastic material, material mechanical properties are improved. Root bending stress is decreased but because of using same material for pinion and gear, contact stress is increased.

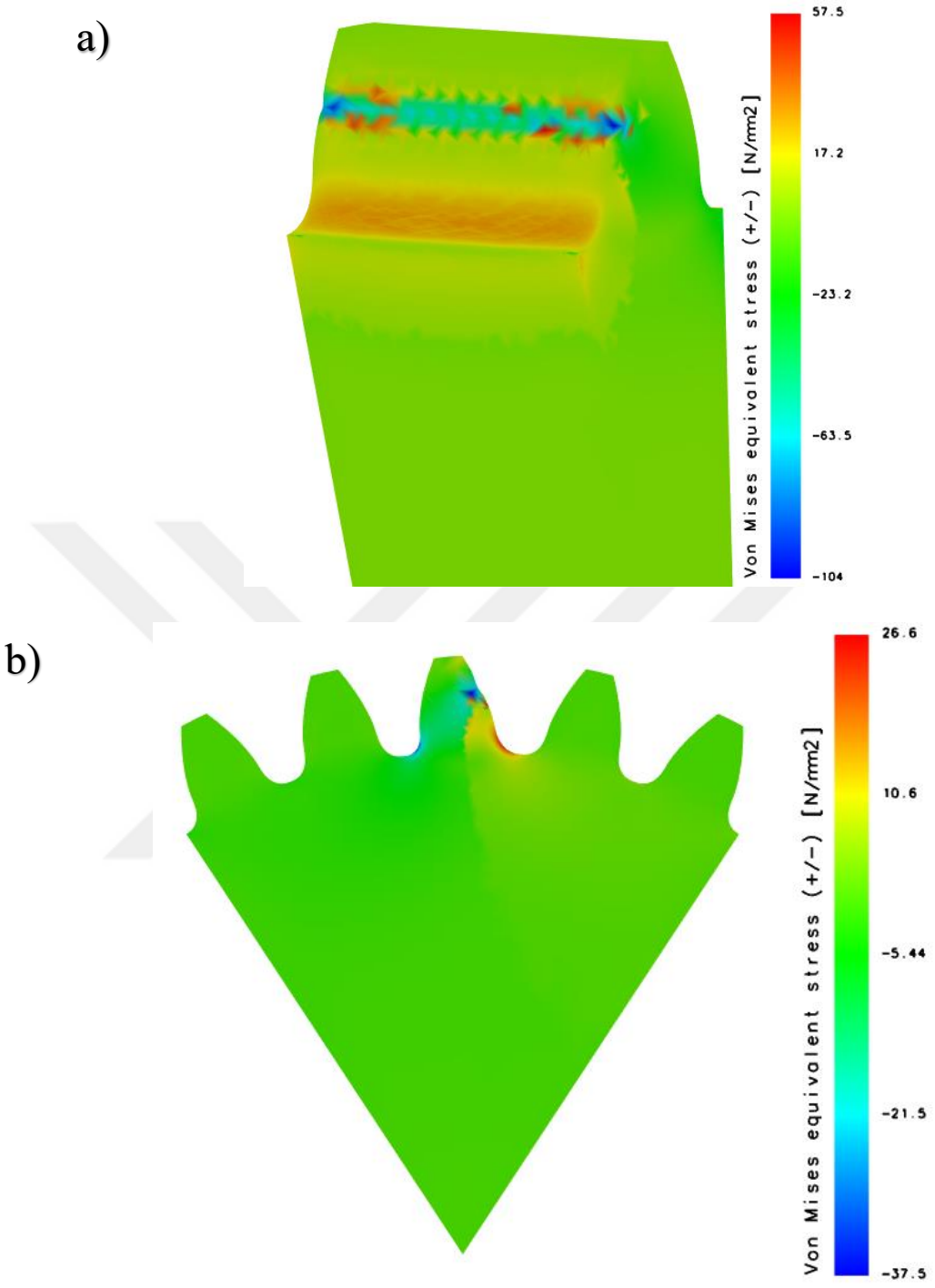


Figure 5.24 3D Tooth Root Stress Distribution of PA66-CF-60 % and PA66-CF-60 % Gear Mesh a) Total Stresses b) Root Bending Stress

When gears are rotated and transmitted power, temperature values are provided as 90 °C. These values can be determined as suitable for analysis which is mentioned in Chapter 0.

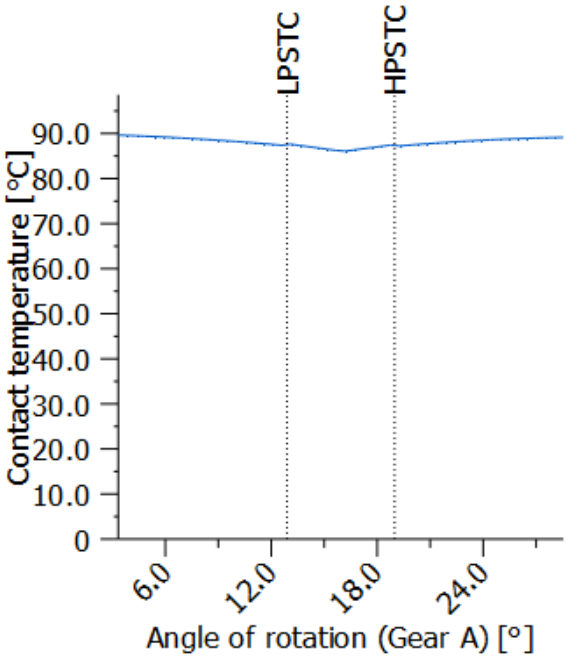


Figure 5.25 Temperature in Contact

Heat Development Curve in 3D is provided as following figure:

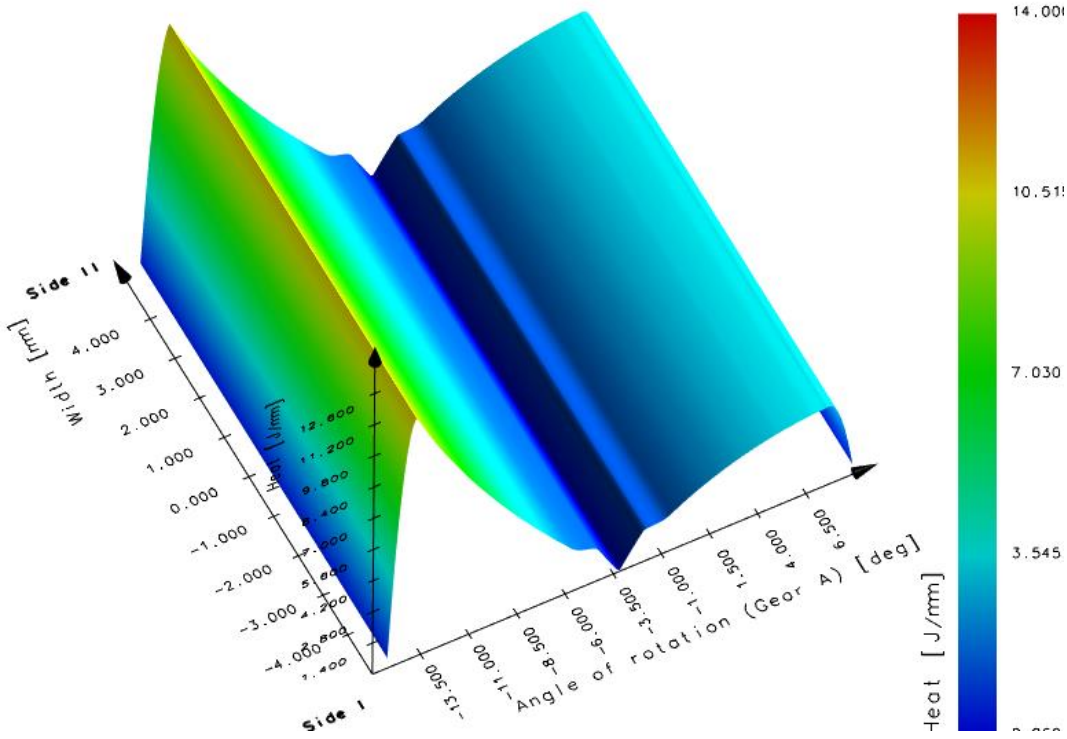


Figure 5.26 3D Heat Development Curve

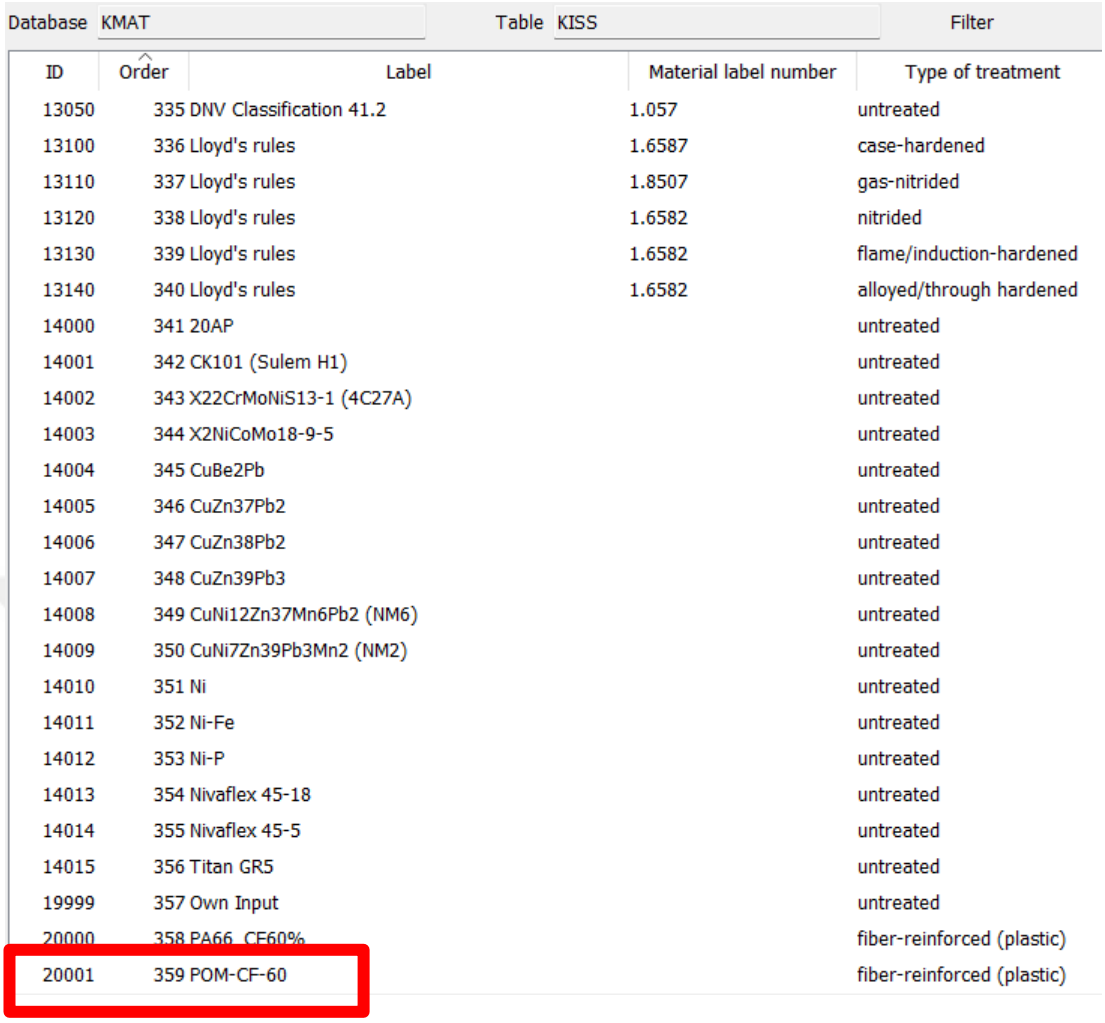
PA66-CF-60 % material can resist given initial conditions which are 3 kW power, 5000 rpm rotational speed in 8000 hours. Safety margin is obtained as higher than 6.7. To compare this value with PA66-PA66 gear pair safety, which is calculated in Section 5.2.2, safety of carbon fiber reinforced polymer shows high mechanical property. According to analysis, which is shown below, PA66-CF-60 % can resist higher loads as 13.5 kW at 5000 rpm.

Table 5.7 PA66-CF-60 % and PA66-CF-60 % Gear Pair Analysis Results (13.5 kW Power and 5000 rpm)

Results	Pinion	Gear
Radial Force	373.3 N	373.3 N
Normal Force	1091.5 N	1091.5 N
Root Stress	189.71 MPa	189.71 MPa
Root Stress Safety	1.41	1.38
Tooth Deformation	85.472 µm	1.472 µm

5.2.4 POM-CF-60 % Material KissSoft Studies

The same studies which are mentioned in Chapter 5.2.1 are conducted to analyze fiber reinforced POM material. New material is added in KissSoft material library as given in figure below. Material properties are set from previous studies and literature reviews. Tooth flank strength, tensile strength, elastic modulus, tooth root strength values with respect to temperature and load cycles are calculated and assumed.



Database KMAT Table KISS Filter

ID	Order	Label	Material label number	Type of treatment
13050	335	DNV Classification 41.2	1.057	untreated
13100	336	Lloyd's rules	1.6587	case-hardened
13110	337	Lloyd's rules	1.8507	gas-nitrided
13120	338	Lloyd's rules	1.6582	nitrided
13130	339	Lloyd's rules	1.6582	flame/induction-hardened
13140	340	Lloyd's rules	1.6582	alloyed/through hardened
14000	341	20AP		untreated
14001	342	CK101 (Sulem H1)		untreated
14002	343	X22CrMoNiS13-1 (4C27A)		untreated
14003	344	X2NiCoMo18-9-5		untreated
14004	345	CuBe2Pb		untreated
14005	346	CuZn37Pb2		untreated
14006	347	CuZn38Pb2		untreated
14007	348	CuZn39Pb3		untreated
14008	349	CuNi12Zn37Mn6Pb2 (NM6)		untreated
14009	350	CuNi7Zn39Pb3Mn2 (NM2)		untreated
14010	351	Ni		untreated
14011	352	Ni-Fe		untreated
14012	353	Ni-P		untreated
14013	354	Nivaflex 45-18		untreated
14014	355	Nivaflex 45-5		untreated
14015	356	Titan GR5		untreated
19999	357	Own Input		untreated
20000	358	PA66, CF60%		fiber-reinforced (plastic)
20001	359	POM-CF-60		fiber-reinforced (plastic)

Figure 5.27 KissSoft Library (POM-CF-60 %)

Added material is selected as gear material to analyze gear parameters as given in Figure 5.28.

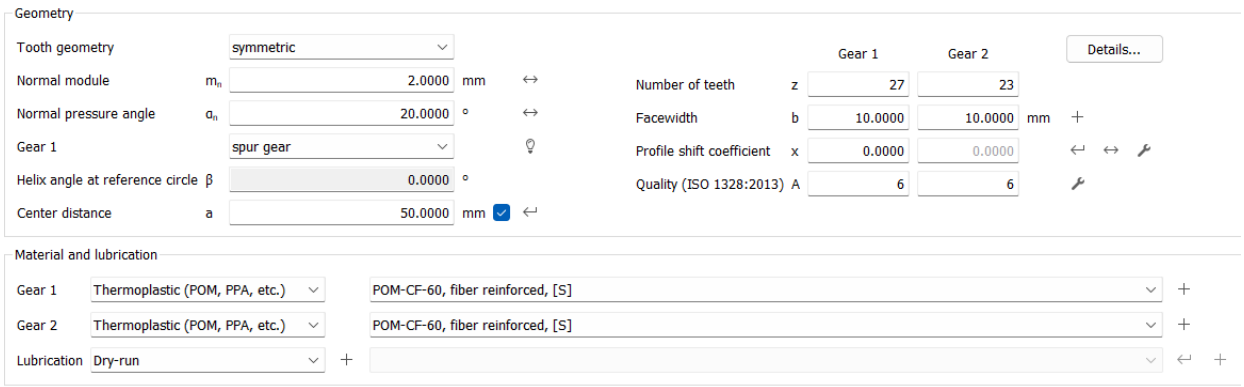


Figure 5.28 Gear Design Parameters for POM-CF-60 % Gear Mesh

Obtained values are listed following table.

Table 5.8 POM-CF-60 % and POM-CF-60 % Gear Pair Analysis Results (@ 50 ℃)

Results	Pinion	Gear
Radial Force	77.2 N	77.2 N
Normal Force	225.8 N	225.8 N
Root Stress	39.25 MPa	39.60 MPa
Root Stress Safety	5.85	5.80
Tooth Deformation	15.45 µm	15.45 µm

Table 5.9 POM-CF-60 % and POM-CF-60 % Gear Pair Analysis Results (@ 100 ℃)

Results	Pinion	Gear
Radial Force	77.2 N	77.2 N
Normal Force	225.8 N	225.8 N
Root Stress	39.25 MPa	39.60 MPa
Root Stress Safety	4.60	4.52
Tooth Deformation	17.98 µm	17.98 µm

Specific sliding which can be used to reduce flash temperature and contact temperature is shown in Figure 5.22. Because of specific values depend on gear design parameters not material, there is no difference between Figure 5.22 and Figure 5.10.

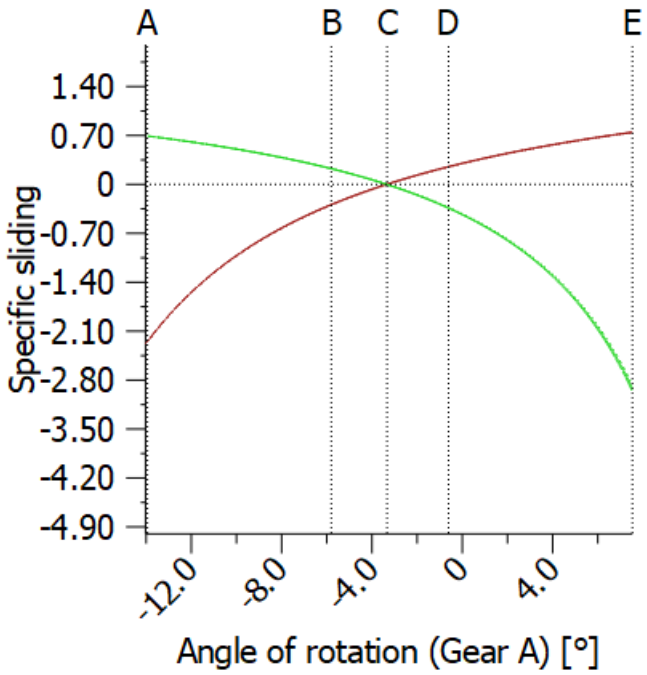


Figure 5.29 Specific Sliding of POM-CF-60 % and POM-CF-60 % Gear Mesh

Root bending stresses are illustrated in following figure. Due to the added reinforced fiber material in plastic material, material mechanical properties are improved. Root bending stress is decreased but due to the same material usage for both pinion and gear, contact stress is increased.

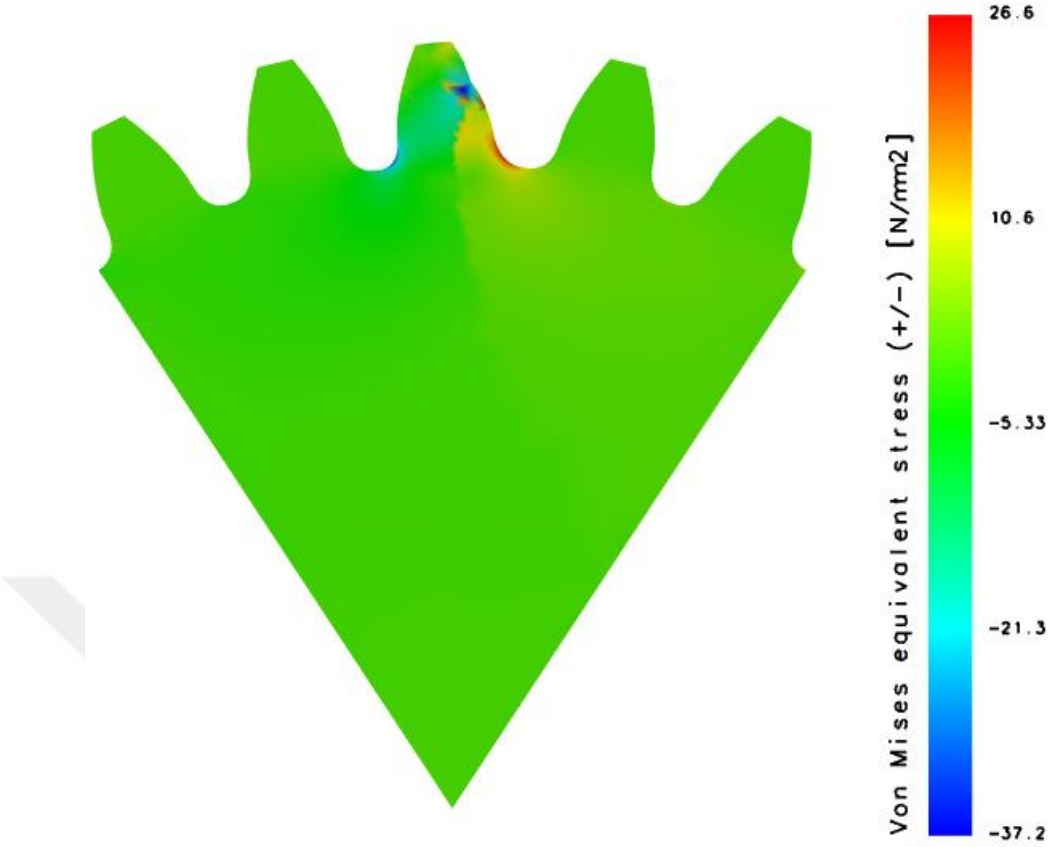


Figure 5.30 3D Tooth Root Stress Distribution of POM-CF-60 % and POM-CF-60

When gears are rotated and transmitted power, temperature values are provided as 90 °C.

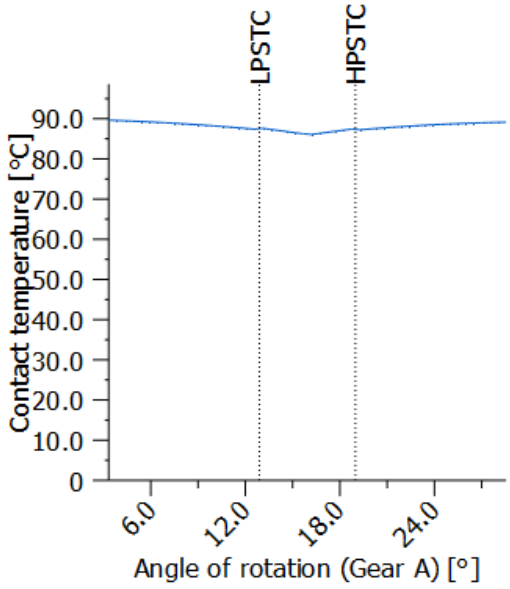


Figure 5.31 Temperature in Contact

Heat Development Curve in 3D is provided as following figure:

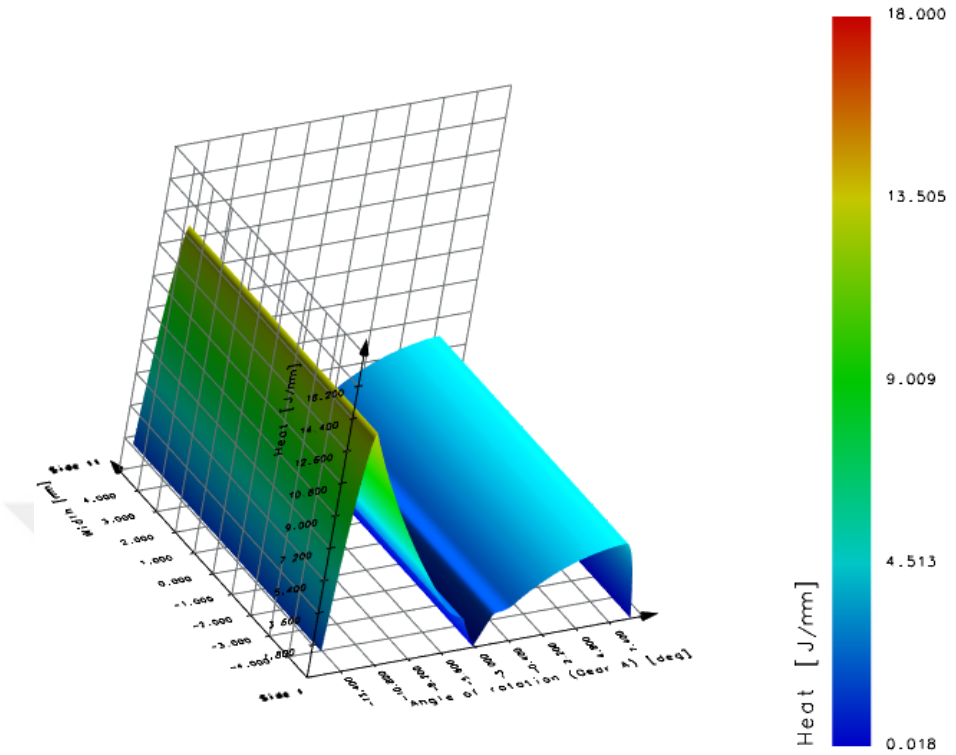


Figure 5.32 3D Heat Development Curve

POM-CF-60 % material can resist given initial conditions which are 3 kW power, 5000 rpm rotational speed in 8000 hours. Safety margin is obtained as higher than 6. To compare this value with POM-POM gear pair safety, which is calculated in Section 5.2.1, safety of carbon fiber reinforced polymer shows high mechanical property. According to analysis, which is shown below, POM-CF-60 % can resist higher loads as 13 kW at 5000 rpm.

Table 5.10 POM-CF-60 % and POM-CF-60 % Gear Pair Analysis Results (13 kW Power and 5000 rpm)

Results	Pinion	Gear
Radial Force	334.7 N	334.7 N
Normal Force	978.6 N	978.6 N
Root Stress	170.09 MPa	171.60 MPa
Root Stress Safety	1.51	1.43
Tooth Deformation	76.63 µm	76.63 µm

By using POM-POM gear materials, we can examine the analysis which is given above. However, we need the bigger size gear to resist 13 kW power at 5000 rpm. When the gear is 4.5 instead of 2, the gear mesh which is manufactured from POM plastic can resist related loads. Size comparison is provided in the figure below:

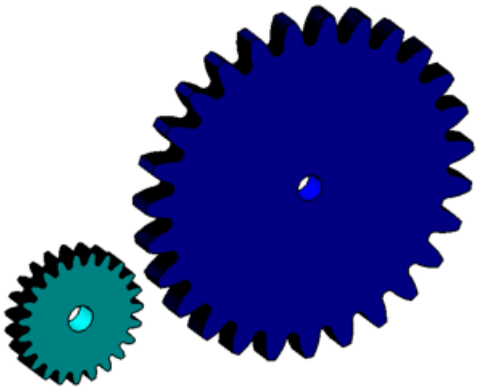


Figure 5.33 Size Comparison of POM Gear and POM-CF-60 % Gear (Pinion)

In other case, thickness of gear can increase to 50 mm instead of 10 mm. Both gears which are POM with 50 mm thickness and POM-CF-60 % with 10 mm thickness have same load capacity.

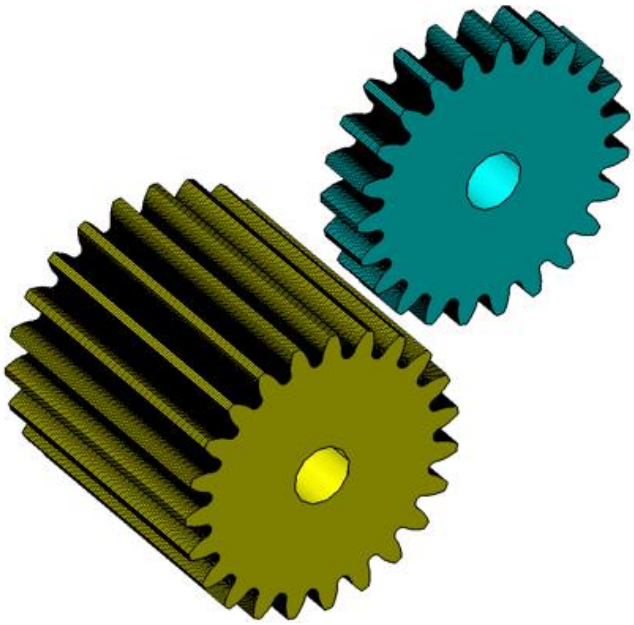


Figure 5.34 Comparison of Thickness of POM Gear and POM-CF-60 %

Table to provide from previous studies which can be used to show weight comparison of gears.

Volume of the gear can be calculated by using the following formula:

$$\mathbf{V_{tooth} = \pi \times \left(\frac{thickness\ of\ tooth}{2}\right)^2 \times Face\ Width} \qquad \textbf{(Eq 5-8)}$$

$$V_{gear} = V_{tooth} \times \text{Number of Teeth}$$

By using formulas given above, masses of the gears are calculated in Excel.

Table 5.11 Calculation of Mass of Gears

Material	Thickness (mm)	z	module	Density (kg/m^3)	Diameter (mm)	**MASS (kg)**
POM-CF	10	27	2	1614	54	0.063
POM 1	50	27	2	1410	54	0.275
POM 2	10	27	4,5	1410	121,5	0.278

Results are provided in Table 5.12 below.

Table 5.12 Weight Comparison of POM and POM-CF-60 %

GEAR	**WEIGHT (g)**
POM (t=10 mm, m=4.5)	≅ 278 g
POM (t=50 mm m=2)	≅ 275 g
POM-CF-60 (t=10 mm, m=2)	≅ 63 g

Same weight analysis can be conducted for PA66-CF-60 material. Similar results can be obtained. As seen, fiber reinforcement can be used to reduce overall system weight. For gear pair used in this study, weight can be reduced for both gears.

CHAPTER 6

FINITE ELEMENTS METHOD

In this section, gear analysis is to be conducted by using ABAQUS Standard Finite Element Software. Gear and pinion are modeled by using KissSoft and Catia V5 R21.

PA 66 – Carbon Fiber (Volume fraction: 60) and POM – Carbon fiber (Volume Fraction: 60) will be analyzed separately. As mentioned before, gear mesh which is made from the same material shows high mechanical properties. Furthermore, two different analyses will be conducted.

By using Digimat MF software, material properties are calculated. ABAQUS and Digimat have interfaces (ABAQUS has Digimat plug-in). Material properties are set according to the following figures.

Carbon fiber is set an isotropic elastic material, then polyamide is set an elastoplastic material.

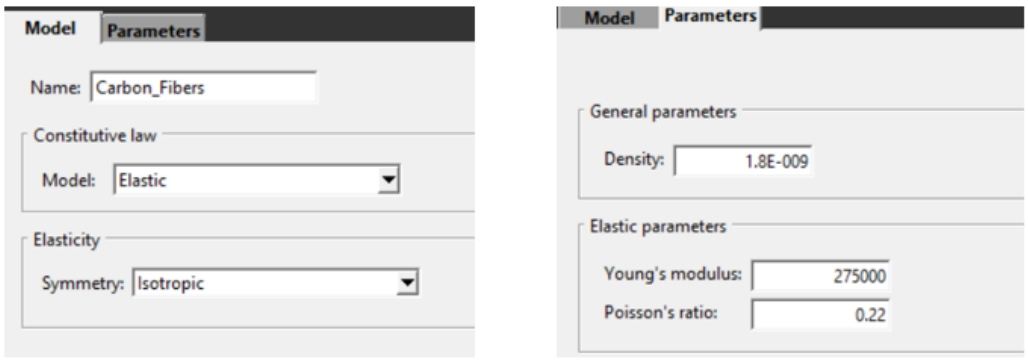


Figure 6.1 Fiber Material Properties

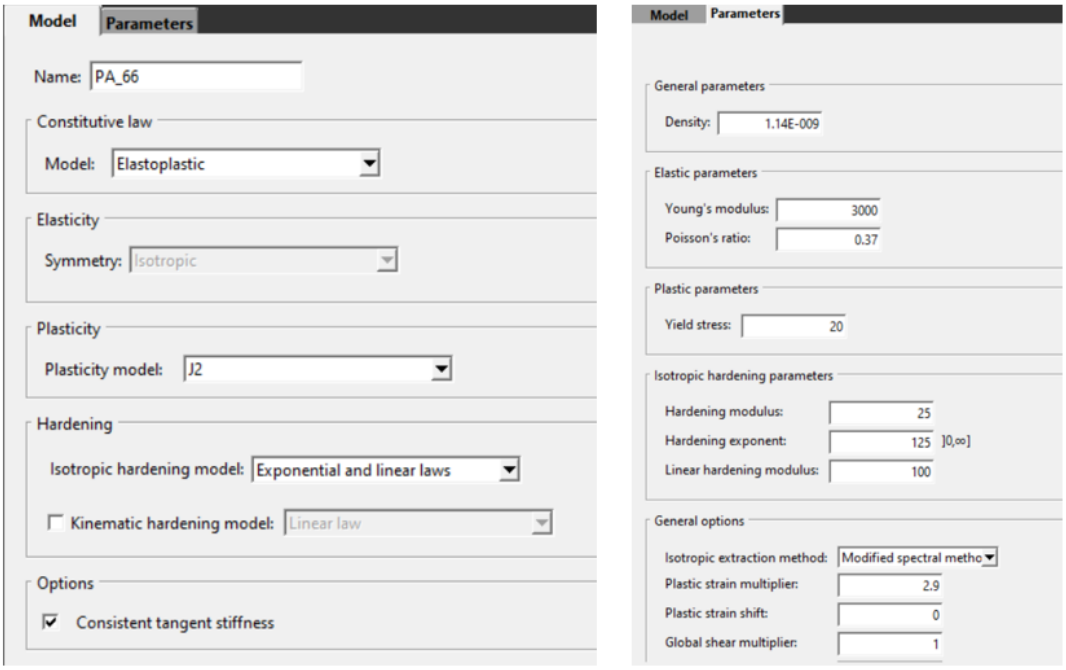


Figure 6.2 Matrix Material Properties

Tsai-Hill 3D failure mode is used in this analysis which formulated as follows:

$$F_A(\sigma) = \frac{{\sigma_{11}}^2}{X^2} - \frac{\sigma_{11}(\sigma_{22}+\sigma_{33})}{X^2} + \frac{{\sigma_{22}}^2+{\sigma_{33}}^2}{Y^2} + \left(\frac{1}{X^2} - \frac{2}{Y^2}\right)\sigma_{22}\sigma_{33} + \frac{{\sigma_{12}}^2+{\sigma_{13}}^2}{S^2} + \left(\frac{4}{Y^2} - \frac{1}{X^2}\right){\sigma_{22}}^2 \qquad \textbf{(Eq 6-1)}$$

After analysis is completed, stress-strain diagram is obtained as following figure:

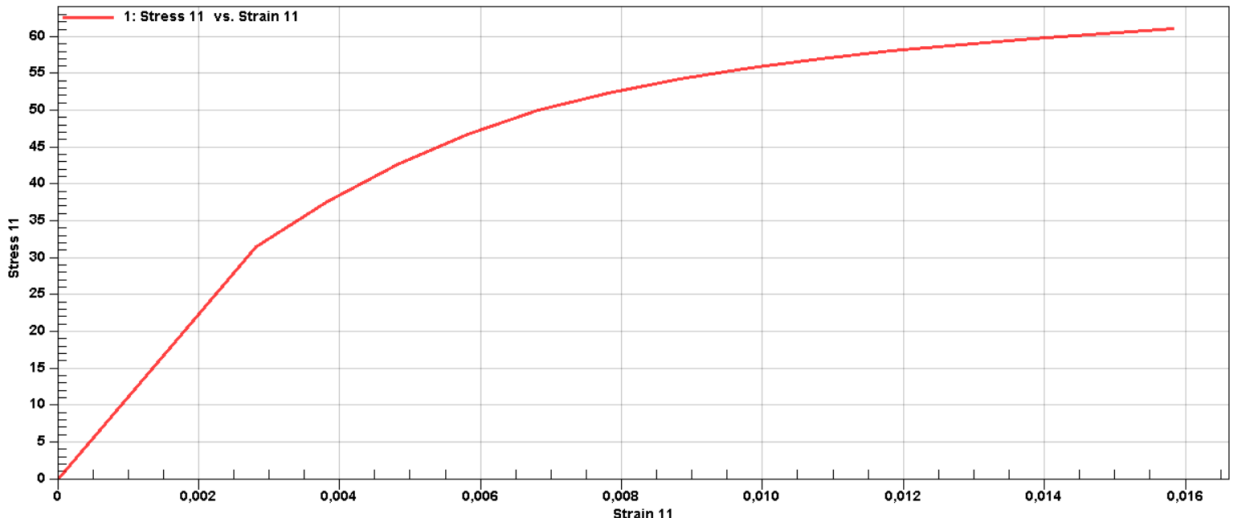


Figure 6.3 Stress-Strain Curve of PA66-CF-60

Two gears (a pinion and a gear) have meshed each other. Center distance is arranged when profile shift coefficient is considered as 0. Hence center distance is calculated as follows [25]:

$$CD = \frac{m_g \times z_g + m_p \times z_p}{2} \qquad \textbf{(Eq 6-2)}$$

Where;

m_g is module of gear,

m_p is module of pinion,

z_g is number of teeth of gear,

z_p is the number of teeth of pinion.

According to calculation results, Center Distance of two gears are 50 mm.

For detail 3D analysis, mesh is set as default which is illustrated in Figure 6.4.

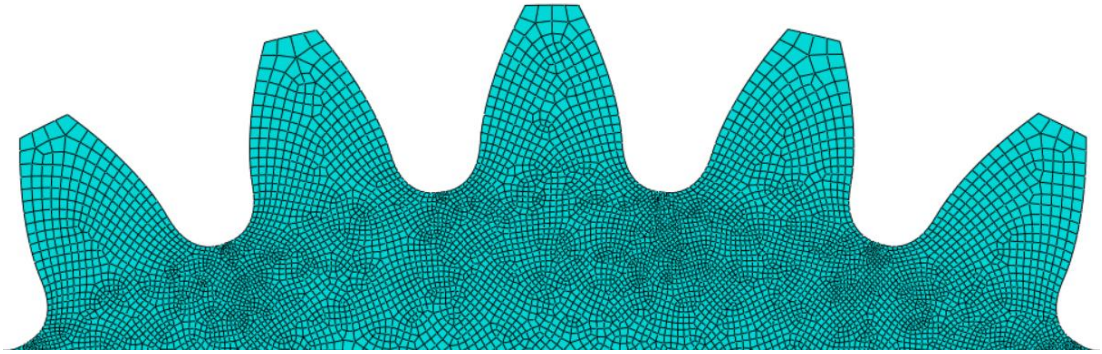


Figure 6.4 Gear Mesh Detail

For boundary conditions, center of gear are considered as couplings. Statical analysis is ran by ABAQUS Software. Results are illustrated as follows in Figures.

One tooth of gear which to be subjected of load from other gear (pinion) is analyzed. The illustration of force is given below:

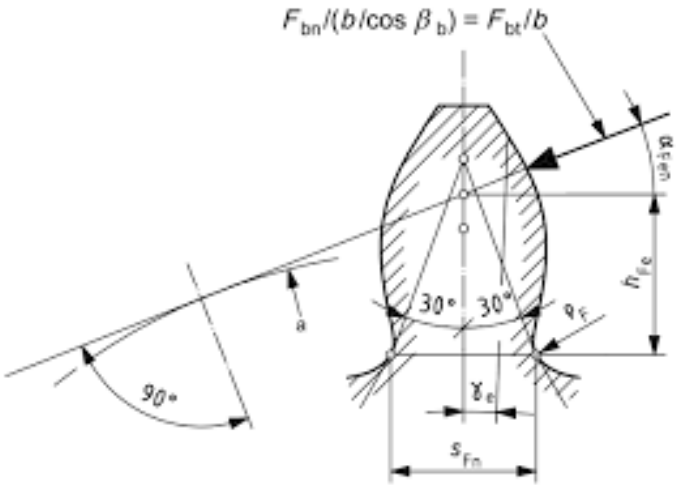


Figure 6.5 Applying Load on Pitch of Gear

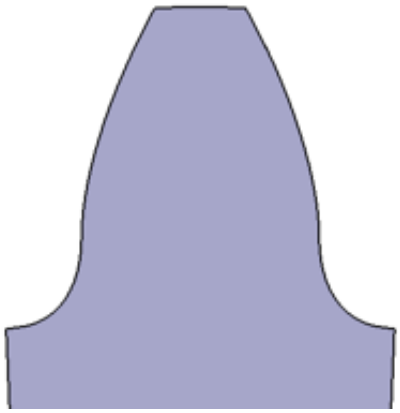


Figure 6.6 Gear Tooth Front View

Force is applied on pitch diameter (HPSTC). Only rotational movement in x-direction is applicable as shown in Figure 6.7.

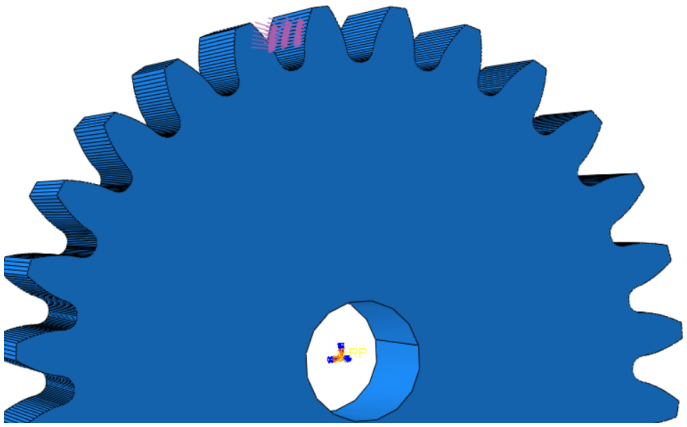
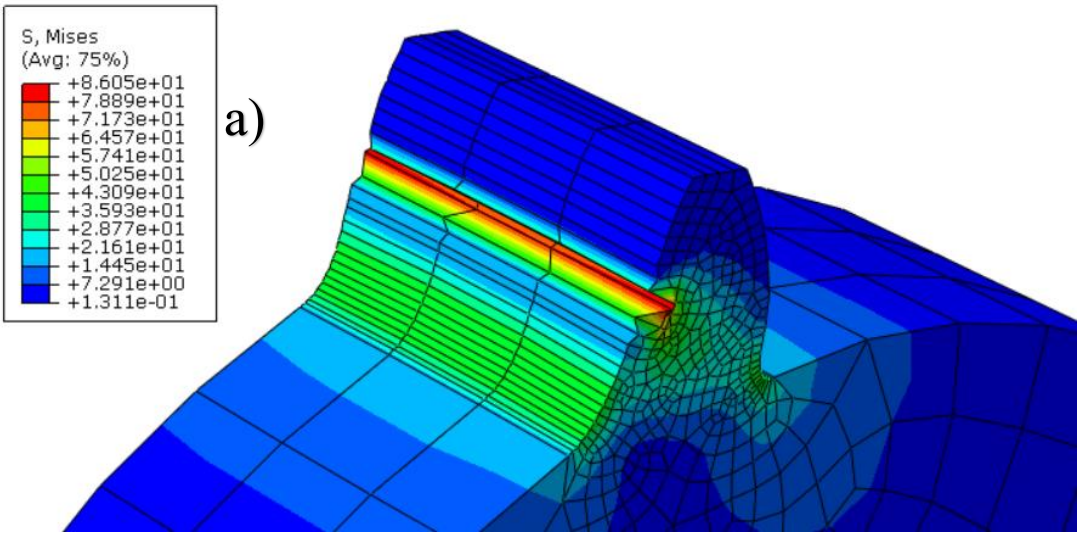


Figure 6.7 Gear Boundary Conditions

Results are illustrated below.



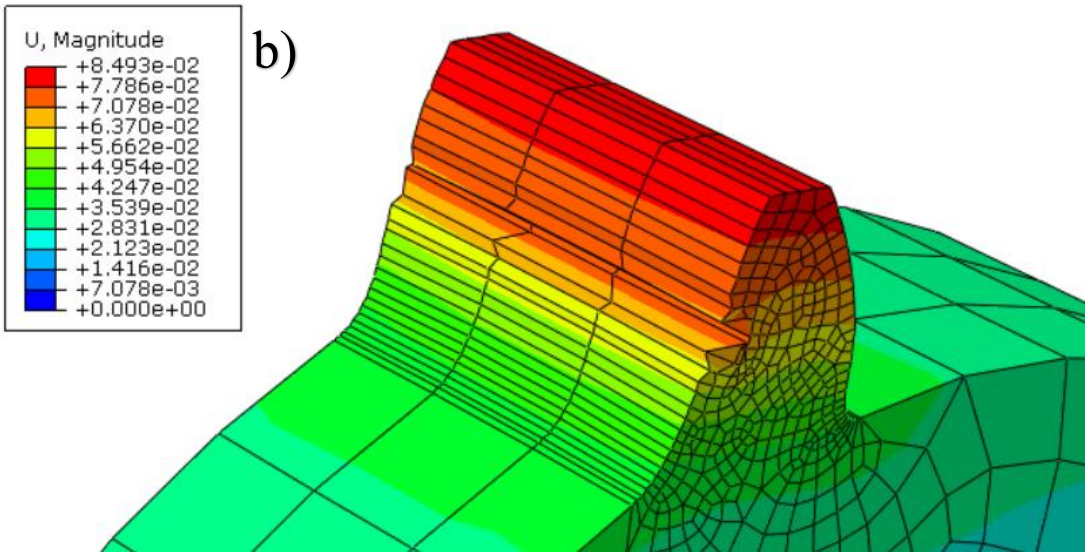


Figure 6.8 One Teeth Analysis Result of PA66-CF-60 % a) Von Mises Stress Values b) Displacement Values

According to analysis results, when force is applied on pitch diameter of teeth because of load transmitted, stress values of teeth root is illustrated.

Same steps are applied when PA66 structural static analysis is conducted. Obtained stress and deformation values are illustrated in Figure 6.9.

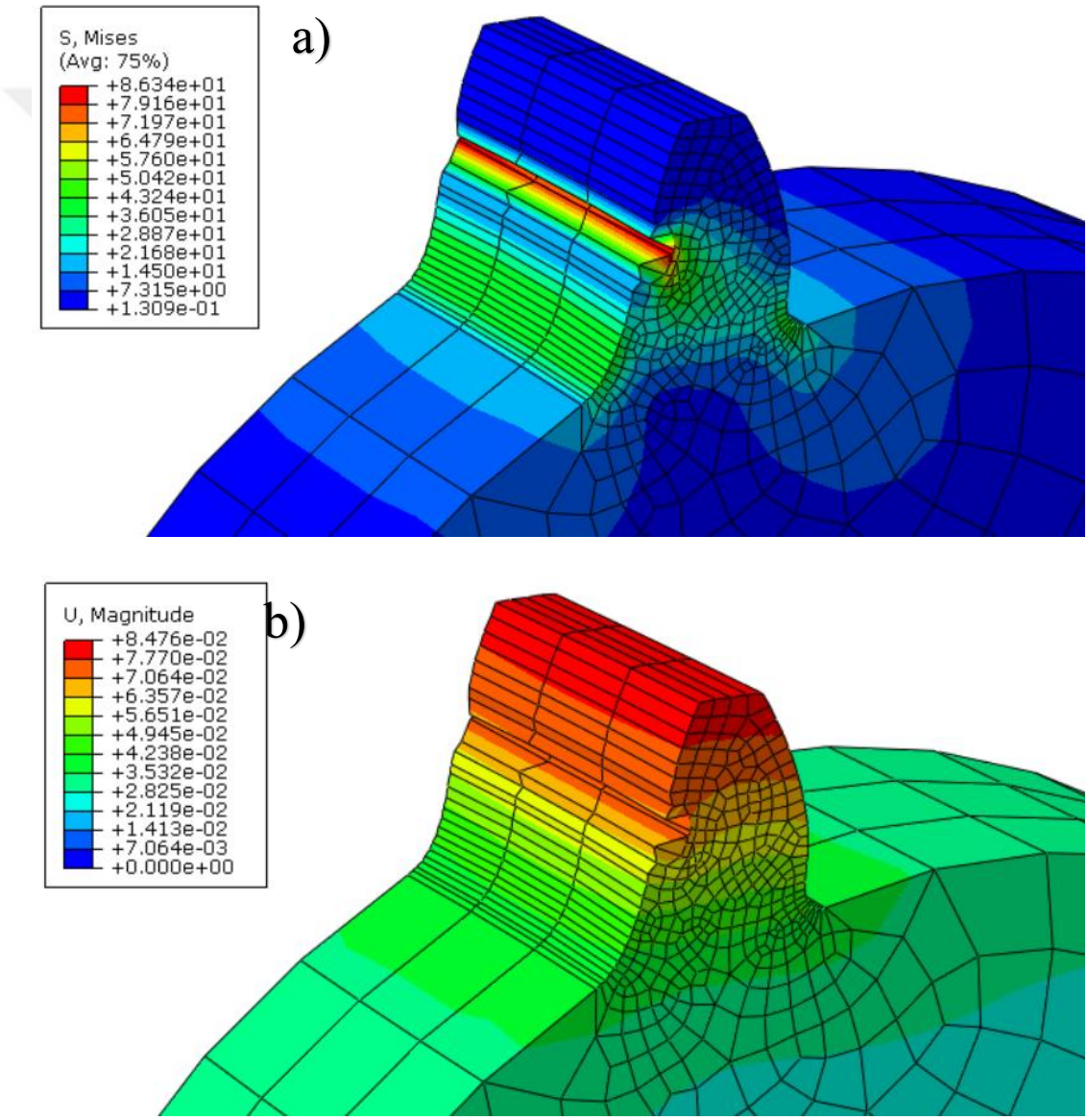


Figure 6.9 One Teeth Analysis Result of POM-CF-60 a) Stress Values b) Displacement Values

Obtained deformation values of PA66-CF-60 % in root of the teeth is smaller than PA 66 material.

Furthermore, by using KissSoft Software, same analysis was conducted. According to results which are given in Table 5.3, root bending stress was obtained as 39.25 N/mm^2.

The graph is shown below in Figure 6.10 to illustrate differences between analysis results.

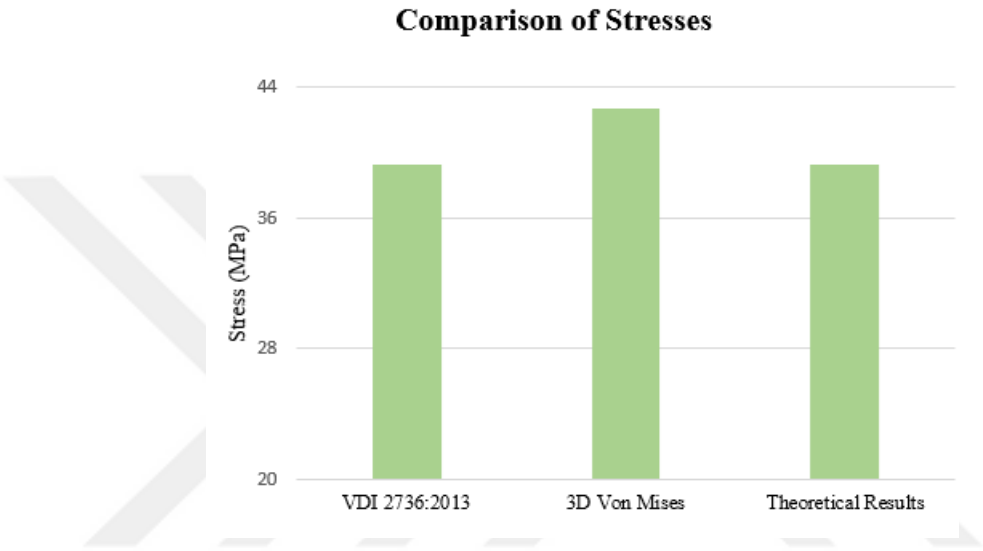


Figure 6.10 Stress Results of PA 66-CF-60

The reason of the differences between results is that ABAQUS is running to analyze the system in 3 dimensions. KissSoft is studied only 1D which is load applied direction. By using ABAQUS software, stress values which are $\sigma_{11}, \sigma_{22}, \sigma_{33}$ normal stresses and $\sigma_{12}, \sigma_{13}, \sigma_{23}$ shear stresses can be found.

Same analysis can be applied to gear which is manufactured from PA66-CF-60. This study shows that VDI 2736 standard is compatible with FEM, also using theoretical studies can be suitable for analysis.

CHAPTER 7

GEAR OPTIMIZATION

After material and stress calculations of gears, gear optimization is studied in this chapter. Gear stresses depend on material mechanical properties and thermal conditions as examined previous sections.

Furthermore, root bending stress can be decreased by using gear design parameters. Micro and macro-optimizations can be applied on gear surfaces and root then tangential and normal forces can be reduced. In this section, gear optimization is considered, and stress changes will be shown. Firstly, effect of profile shift coefficient on gear root stress is studied. Tip relief and root fillet enhancement which are other most important micro-modifications for gears will be also examined.

7.1 Profile Shift Coefficient

Profile shift coefficient is also known as “the addendum modification coefficient” It is used in gear design to improve gear parameters such as reducing contact stress, root bending stress, reduce specific sliding and so on. Profile shift coefficient describes the intentional alteration of the standard gear tooth profile to achieve specific objectives, such as adjusting center distance, improving load distribution which helps to reduce gear surfaces stresses [30, 31].

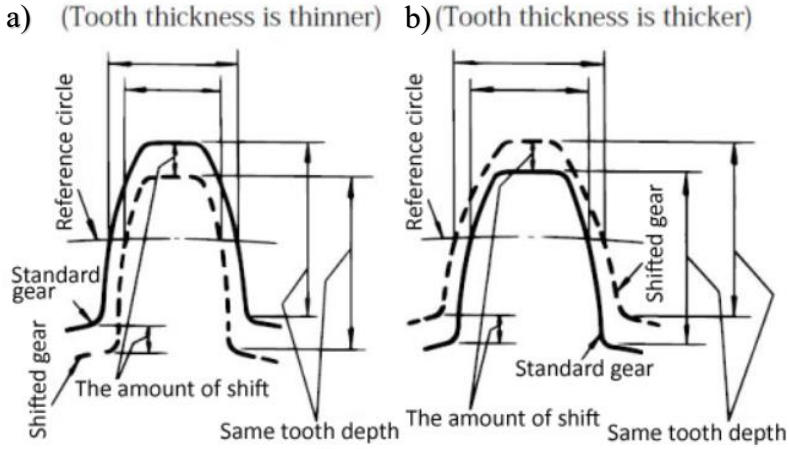


Figure 7.1 Profile Shift Coefficient a) Negative Shifting b) Positive Shifting [30]

As shown in Figure 7.1, in positive shifting, the tooth thickness increases then the tooth is more resistant to bending.

Profile shift coefficient is examined as "x" and general formula is given below.

$$x = \frac{\Delta a}{m} \qquad \textbf{(Eq 7-1)}$$

For the analysis, which is mentioned in Section 5.3.1, reanalysis is conducted by using profile shift coefficients.

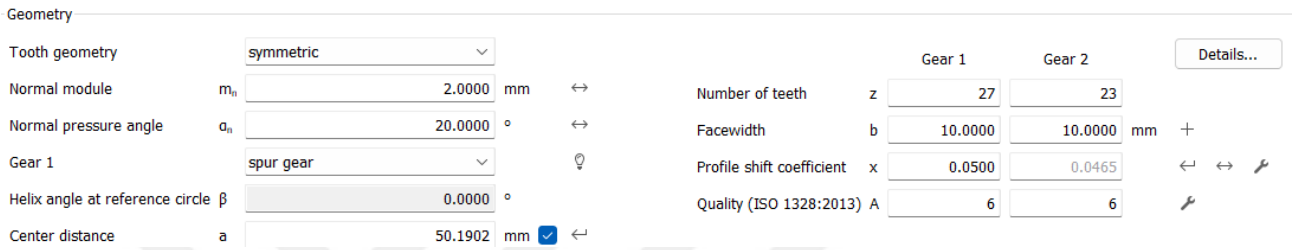


Figure 7.2 Gear Design Parameters with Profile Shift Coefficient

When analysis is conducted with same steps which are listed below:

- 5000 rpm, 3 kW
- VDI 2736:2013-modified (YF Method B)

The following results are obtained

Table 7.1 POM-POM with Profile Shift Gear Pair Analysis Results (@ 50 °C)

Results	Pinion	Gear
Radial Force	77.2 N	77.2 N
Normal Force	225.8 N	225.8 N
Root Stress	31.55 MPa	31.55 MPa
Nominal Contact Stress	45.22 MPa	45.22 MPa
Root Stress Safety	1.617	1.606

Then, we can compare obtained results and data as follows:

- Comparison of Root Stresses
    - Root stresses of gears with profile shift is higher than without profile shift.

- The other obtained values are given in Table

Table 7.2 Comparison of Gear Parameters

Parameters	Gears with Profile Shift	Gears without Profile Shift
Operating Pitch Diameter (mm)	54.205	54.200
Nominal Contact Stress (MPa)	45.22	45.66
Radial Backlash (mm)	0.279/0.153	0.285/0.160
Mass (g)	32.60	32.33
Wear Service Life	58.73	56.34

Parameters which are listed in Table 7.2 may not affect root bending stress directly. However, gear designers also consider these parameters to improve gear service life. All the values are improved except mass. Because, to design lightweight equipment is important in this study.

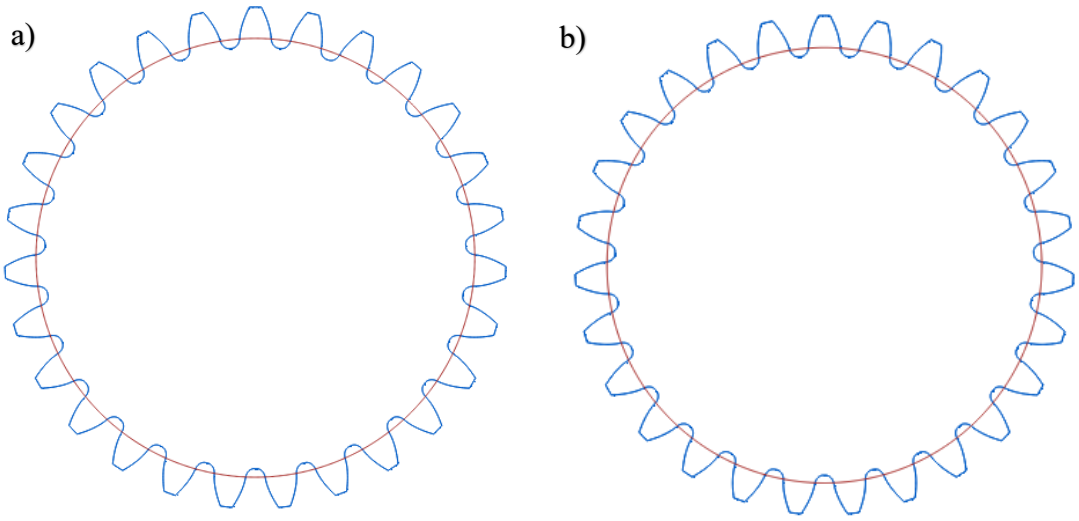


Figure 7.3 Tooth forms of Gears a) Without Profile Shift b) With Profile Shift

7.2 Tip Relief

Tip relief is another modification type of gears which is applied to the tip of gear teeth. In this process, a small amount of material is removed from the tip. This adjustment provides a smoother transition at the beginning and end of the teeth' contact area, reducing sudden load changes and noise [8]. The purpose of this process is to improve load distribution, to reduce vibration and noise, to minimize wear, and to lower impact loads [24].

There are two types of relief which are linear and parabolic relief. In linear relief, material is removed linearly from the tip of the gear tooth. In parabolic relief, material is removed parabolic fashion, and it offers a more complex relief profile [31].

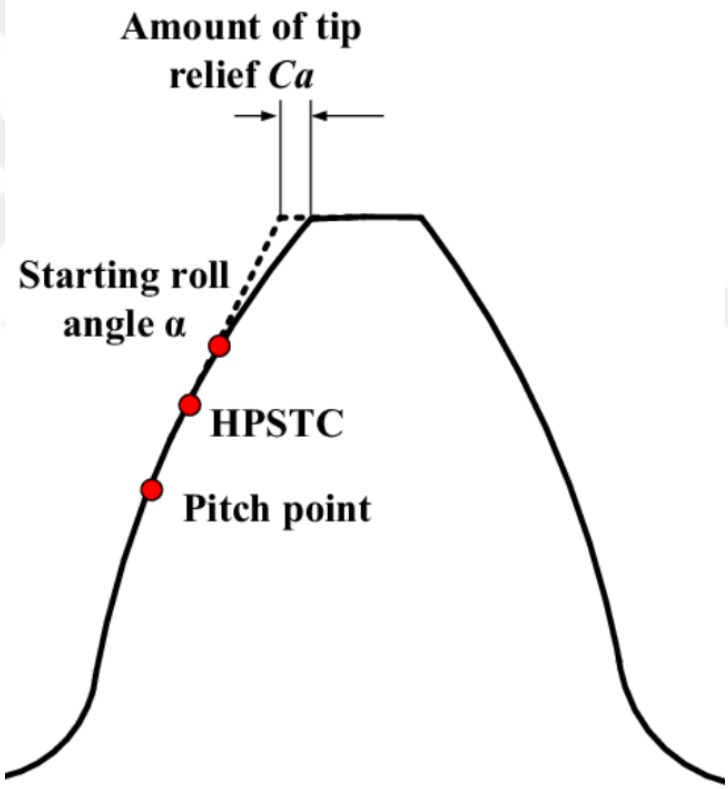


Figure 7.4 Tip Relief

CHAPTER 8

RESULTS and CONCLUSIONS

In this chapter, the results obtained from previous studies are mentioned and discussed. Steps and used programs are summarized. Thesis aims, assumptions, future works are mentioned.

KissSoft is used VDI-2736 standard for root bending stress calculations. According to formula within the standard, which is mentioned in Chapter 5, bending stress does not depend on strength of material. However, in Finite Element Method, ABAQUS solves the analysis by using stiffness values of elements. Strain values depend on mechanical properties and the ratio of stress and strain is equal to elastic modulus of material. Elasticity of SFRP depends on the elastic modulus of fiber and matrix material. According to literature reviews and reference books which are mentioned in Chapter 1 and 2, POM and PA66 materials are selected as matrix materials because of their advantages. Furthermore, carbon fiber and glass fiber materials are selected as fiber reinforcement materials.

This thesis is prepared to investigate small fiber reinforced plastic gears behaviors. The demand for strength ratings for plastic gears has been consistently increasing. Today, studies on plastic gears develop. Engineers and scientists try to improve plastic gears. Many of them studied on gear parameters and gear optimizations, material improvement, manufacturing, improving production etc. Tests are conducted to examine plastic gears behavior. Obtained values are examined and published to show plastic gear behaviors. Mechanical properties of plastic gears are tried to improve. However, there are difficulties in this area. Plastic gears can have poor mechanical and thermal properties. Manufacturing tolerance range cannot be arranged easily. Also, there are no international standards for plastic gears. The only widely accepted strength rating method in Western countries had been the German guideline VDI 2545, which was withdrawn in 1996. ISO 6336 is used to develop VDI standard. Due to the poor mechanical properties of polymers, fiber materials can be used to their mechanical properties.

Fiber reinforcement is used to improve mechanical properties of plastics. Engineers and scientists try to develop new methods to improve plastic gears mechanical and thermal behaviors. In this study, new materials are tried to develop and used in analysis. Carbon and glass fibers are used to develop mechanical properties of plastics. Most used plastics are chosen such as POM and PA 66. All the plastics are suitable to develop their mechanical properties with fibers. Three types of mixing ratios (30 %, 45 %, 60 %) are examined. When fiber ratio increases, then mechanical properties also increase.

In this thesis, by using fiber reinforcement materials, mechanical and thermal behavior of plastic gears are examined. Especially, tooth root stress which is known as root bending stress or bending stress is examined at high temperatures. Carbon fiber and glass fiber are mixed in PA 66 and POM materials separately. Mechanical and thermal behavior of new materials are provided in Digimat Software. Gear parameters and force values are obtained by KissSoft. Finite element analysis is executed in ABAQUS and KissSoft Software separately. Firstly, by using fiber and matrix materials properties, new materials are identified with thermal and mechanical properties. Varied materials and mixing ratio are considered within the scope of thesis. As a result, PA66-CF-60 % and POM-CF-60 % materials are chosen. By using Digimat FE Software, CFRP composites are obtained, and their mechanical properties are calculated.

In this study, three types of results which are theoretical results, finite elements method results and KissSoft results are provided and examined. KissSoft and theoretical results are same because VDI-2736 is used for both methods. Furthermore, KissSoft is not sufficient to calculate and determine plastic gears failures. Especially wear is the most known failure type. It depends on the contact temperature, friction ratio, mechanical properties at elevated temperatures [32].

For the gear bending calculations, KissSoft software is used. Firstly, POM-POM gear mesh and PA66-PA66 gear mesh are analyzed separately by using VDI-2736 to calculate bending stress of gears. Then new materials are analyzed, and differences are compared. Contact stress is obtained as high values because of using same materials for pinion and gear.

Plastic gears and short fiber reinforced plastic gears were compared according to weight of the gears, life, manufacturing status, and so on. According to the studies which are mentioned in 0 and 0, effects of carbon fiber can be shown. By using fiber reinforced plastic gears instead of plastic gears, gear pair can resist higher loads at high temperatures and weight is reduced. Root bending stress of PA66-CF-60 is lower than PA66 gear pair. It means that, we can use lightweight gear to transmit same loads or higher loads. Also, when gears are meshed and started to transmit loads each other, heat is developed in gear tooth because of contact stress. Furthermore, if we compare Figure 5.11 and Figure 5.32, we can say that by using fiber reinforcement with thermoplastics, heat development can be reduced.

In Section 5, all gears which are POM-POM, PA66-PA66, PA66-CF-60-PA66-CF-60, POM-CF-60-POM-CF-60 gear pairs are analyzed separately. Carbon fiber reinforced materials are shown better mechanical properties, lower tooth root bending stress and lower heat development. PA66-CF-60 gear pair can resist higher loads. Overall, the study demonstrates the effectiveness of fiber reinforcement in improving mechanical properties and reducing root bending stress in plastic gears, with PA66-CF-60% gear pairs showing higher load resistance.

In sixth section which is mentioned as Finite Element Method in above, by using ABAQUS CAE Software, 3D stress analysis is conducted. Applied force is calculated both theoretical formulas and KissSoft analysis. Boundary conditions are identified then analysis is executed. The Finite Element Method analysis confirms the findings obtained from KissSoft and theoretical calculations, validating the use of VDI 2736 for fiber-reinforced plastics.

Overall steps and results are illustrated below as workflow block diagram in Figure 8.1. Mechanical properties are obtained and examined for plastic gears. Firstly, material is improved by fiber reinforcement. Analyzes are conducted separately. Then, optimizations are applied on gear which are manufactured by fiber reinforced material. Comparison is shown between results. By using fiber reinforced material, gear can resist higher loads.

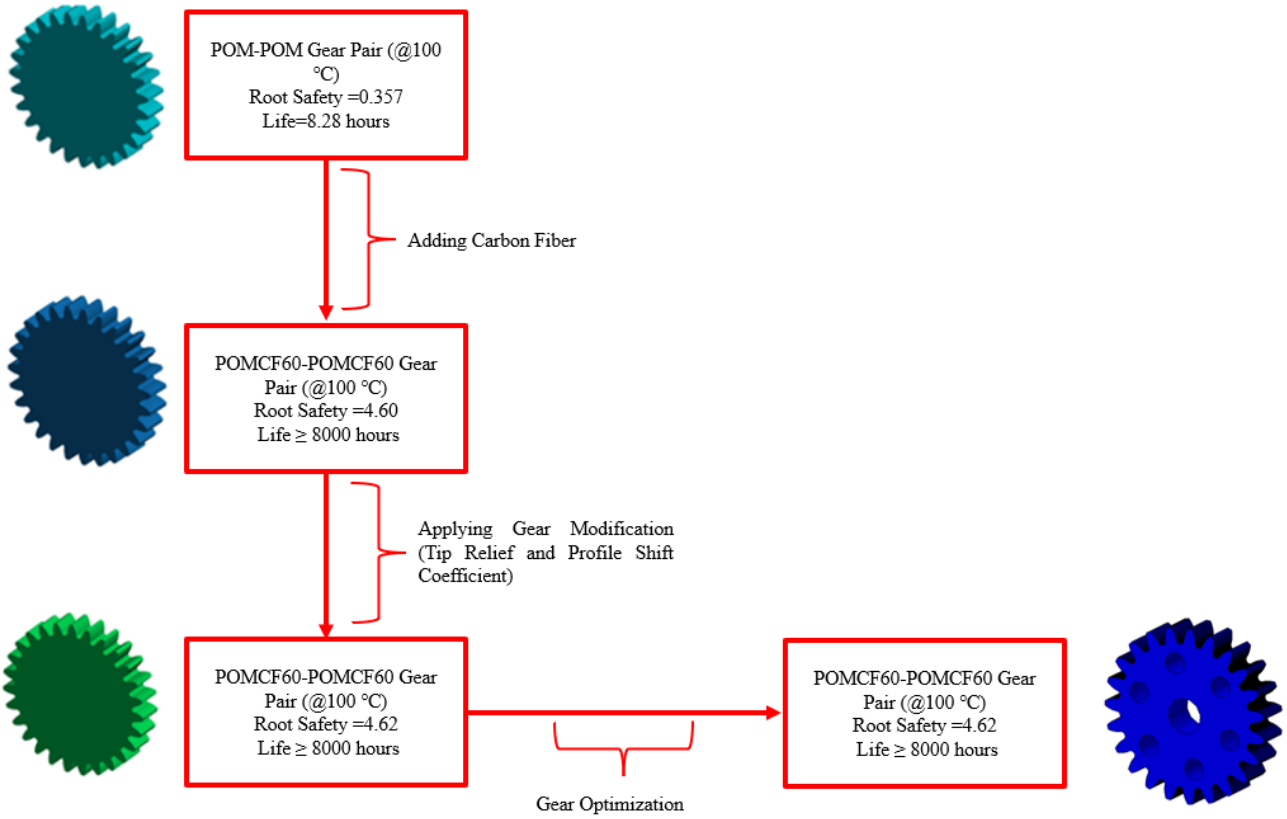


Figure 8.1 Study Workflow Block Diagram

In this thesis, following results are obtained:

- RVE size and mesh frequency can affect mechanical properties of CFRP, but this effect can be negligible.
- PEEK, PA66, and POM can be used as gear material for light power transmission. Furthermore, these materials can be enhanced by fibers.
- CFRP materials exhibit better mechanical properties.
- Carbon fiber and glass fiber materials can be combined with polymer materials for gears.
- VDI 2736 can be utilized within the scope of root bending stress. Comparable results are obtained and can be compared to 3D FEM analysis results.
- PA66-CF-60 has demonstrated better mechanical properties than the PA66 gear pair. PA66-CF-60 material can be chosen instead of PA66 material considering crucial factors such as bending stress, life, weight, and so on.
- A gear made of POM-CF-60 material can withstand four times the load under the same parameters than POM-POM gear mesh.
- To bear the load carried by POM-CF-60, a gear made of POM needs to have its thickness increased by at least five times or its module increased by 2.5 times.
- Due to the mechanical properties of PA66-CF-60 exhibits better mechanical properties than POM-CF-60 at elevated temperatures, PA66-CF-60 can be chosen for heavy applications.
- Tooth root service life can be increased by using fiber reinforcement. Life of PA66-CF-60 gear pair can be obtained as eight thousand hours.
- Gear optimization can be applied on gear teeth surfaces. Profile shift coefficient and tip relief can be considered to improve gear mechanical properties.
- For weight reduction, material which do not affect the mechanical properties can be removed from gear surfaces.

This study is conducted theoretically, assuming CFRP materials are perfectly manufactured. In real life, the injection molding process is applied to obtain fiber-

reinforced plastics. In this process, to obtain foreign objects such as air in a mixture, a vacuum process is also performed. In any case, air can be still mixed in the material.

Additionally, experimental validation is necessary since VDI 2736, used in KissSoft for root bending stress, may not suffice as an internationally accepted standard.

Also, this study needs to be assessed. Because VDI 2736 which is used in KissSoft for root bending stress is not sufficient and an internationally accepted standard.

The tensile strength of PA66-CF-60 and POM-CF-60 is assumed. Studies in literature utilize test results, and these values should be validated through testing in a controlled environment. Gears can be manufactured using injection molding methods and assessed accordingly.

This thesis is aimed to root bending stress because of the VDI 2736 interested area. Wear is considered an important failure type of plastic gears. Due to the contact stress of gear flanks, temperature increases in the flank of gears. It causes the wear flank of gears. Gear pair which is manufactured with the same materials have shown high wear. PEEK shows better mechanical properties than POM and PA66 which resist wear, but this material is expensive. Thermal degradation can be also considered and studied. It is caused by contact stress too.

For future work, a test rig can be developed to verify and improve analyses. Plastic materials cannot be evaluated exactly as metallic materials in some methods due to temperature dependency. Therefore, a test rig with various temperature conditions can be employed. Endurance limit root and flank values are assumed in this thesis based on test results.

Introducing these values indicates the need for further testing. However, due to the theoretical nature of this study, endurance limits are assumed using academic and test studies.

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