Design and Analysis of Spur Gear

Submitted in partial fulfillment of the requirements
Of the degree of

MASTER OF TECHNOLOGY
IN
CAD/CAM
By

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CERTIFICATE

This is to certify that the Research work titled **Design And Analysis of Spur gear** that is being submitted by **Ankit Yadav** is in partial fulfillment of the requirements for the award of **Master of Technology**, is a record of bonafide work done under my guidance. The contents of this research work, in full or in parts, have neither been taken from any other source nor have been submitted to any other Institute or University for award of any degree or diploma.

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(Ankit Yadav)

ABSTRACT

Gears are important mechanical machine component used for transmission. The properties of gears are influenced due to loading, speed of rotation, etc. These factors result in gear failures like wear, fatigue, pitting, etc. Various researches have been carried out to find out the behaviour of gears made up of different types of materials. Some research has been done to improve the strength, wear characteristics, and life cycle of the gear. These researches were carried out experimentally with the help of various machines as well as with the help of numerical models. In this research work we are going to do the fatigue analysis of spur gear. We have designed the gear in the CATIA software and then imported the drawing to the ANSYS software where rest of the analysis is done. The results we got are then compared with the mathematical results to check how close the software analysis is to the theoretical results. In this work, we have designed the gear with the help of CATIA and have analysed the fatigue life of gear with the help of ANSYS. We have used various mathematical equations for contact and surface fatigue to check the results.

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List of abbreviations

1. LSM — Laser Surface Melted
2. LSMMP — Laser Surface Melted and Micro-shot Peened
3. FZG — Forschungsstelle für Zahnräder und Getriebebau
4. POM — Polyoxymethlene

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1
Introduction

1.1 Project background

The gears have been used for mechanical transmission for a long time. While transmitting power, the gear generates stresses at the mating points over the teeth, which causes various types of failures in gear. Researches have been done on various materials of gears under different conditions to analyze their behaviour. Previously the gear analysis was generally based upon the experimental approaches and various types of failures were analyzed. Nowadays, the analysis part has become very easy due to computers. Now we can easily analyze the gear or any other mechanical structure with the help of various software. So, in spite of using the experimental approach, we can go for a software approach for analysis. Various researches have been done with the help of experimental setup under various conditions. Some software analysis have been done as well but were based upon the static analysis of gear.

1.2 Objective of study

In this research work we are predicting the fatigue life of gear using ANYS and are calculating the contact stress using ANSYS and on the other hand we are calculating the stresses mathematically with the help of equations. Mathematical and software results are then compared to check how close they are.

2

Literature review

2.1 Introduction

Gears are used to transmit motion with the engagement of teeth. The tooth act like small levers. It is one of the most important parts of the machinery used in industry. They are widely used due to their many advantages like high transmission efficiency, reliable operation, compact structure, etc. Its application can be found from a small clock to complicated aerospace accessories. During transmission, gears are subjected to different amounts of forces due to which they are prone to different types of failure like wear, fatigue, pitting, spalling, etc. Researches have been done to improve the strength and the behaviour of gears depending upon the demand by the industries when they are subjected to different parameters with different effects. Researchers are keen to solve these problems of failure of gears so that their life cycle can be increased. Generally researches were based upon the experimental study of the gear. In the experimental study, the gear analysis is done with the help of some experimental test rig. And under various loading conditions and with varying parameters, the analysis takes place. But the experimental procedure requires more costing as the gear is to be manufactured initially for the desired material so that the analysis to be done. While very few researches have been done on software basis. This method is emerging out as the time and money saving. The material of which gear is to be analysed is designed in the modelling software and is imported to some analysis software for further analysis. In the analysis, we can get whatever type of analysis we want. And based on that, we can change the properties and other things in the gear while manufacturing to increase its potential.

2.2 Reviews

Sudhaghar calculated the service life of various fiber weight percentages, i.e., 5%,10%,15%, and 20% of fibers reinforced polyester gears i.e. of Madar and bauhinia reinforced polyester. These are the natural fibers that are found from the branches of the tree. Glass fiber reinforced Nylon and the acetal gears are manufactured, and some relation was made between wear and surface temperature. Based on the researches, it was concluded that to improve the failure of gears generally due to root and pitch fracture, and these natural fibers were used. The composite gears were rolled and slide against nylon gear with the help of testing rig, and the results of the test were checked with unreinforced polyester gear. For testing, different amount of torque was provided, i.e., 4Nm, 8Nm, 12Nm, and 16Nm and results were observed. After the test, it was concluded that the wear rate for composite gears while running with nylon gears was less as compared to unreinforced polyester gears. With an increase in torque, the plastic deformation increases. The service life of composite gear is increased by 16.67%. [1]

Bharti Sarita used Polyamide 66 and steel gear to check the effect on their surface due to lubricant. The gear was manufactured with injection molding. Power absorption test rig was used to carry out the testing. The effect of lubricant and without lubricant on the gear was checked. The lubricant used was Standard automotive gear oil (SAE 75W85). The gears were provided with various amount of loads, ranging from 1.8-4.5 Nm with different parameters taken into consideration like the rotational speed of 800rev/min. The results indicated that the gear tooth temperature estimated at 4.5Nm load was 32 per cent greater than that at 1.8 Nm load. Under these conditions, i.e., 1.8-4.5 Nm, lubricated gears life increased by 66-200%. The lubricant also helped in dissipating the heat at the tooth surface of the gear. [2]

Praveen Silori estimated fatigue life along with understanding the stresses which were concentrated on the root of AISI4041 Alloy Steel and Ti6242s. The modeling of gear was done on Solid Edge V19 with the parameters selected accordingly. For analyzing the stresses on the tooth radius of gear, different gears of varying radiuses like 0.8mm, 1mm, 1.2mm, 1.4mm were modeled. The analysis and the meshing of gears were done on ANSYS 14.0 for different materials. FEA was done for different tooth radiuses for different materials. The gear assembly was meshed by a solid 186 element. Certain parameters, assumptions, and boundary conditions were given to the system. After analysis, they found that for the AISI4041 root radius 0.8mm and 1.4mm were best for design and manufacturing. And for Ti6242S alloy, a 1mm tooth radius was best suited. The fatigue life and safety factor for Ti6242S alloy were best. These gears can be used for high-temperature applications. [3]

Paras Kumar used the AGMA approach to investigate pitting and bending fatigue failure along with the tooth profile of Case Hardened and through-hardened gears. AGMA approach results were compared with FEA results. The tests revealed that Case hardened gears delayed pitting, but it increased the probability of bending fatigue failure. The probability of pitting/bending fatigue was highest at tooth contact region, and the chances for failure by Contact fatigue was more as compared to bending fatigue. Through hardened gear showed better bending fatigue life while the case hardened gear showed better contact fatigue life. [4]

Wei Li investigated the effect of Shot Peening on the fatigue characteristics of 20CrMnMo carburized gear and hardened gears using up-down test method and grouping test method. The fatigue stress values under different reliability were obtained. The results concluded that fatigue strengths of gears increased by 14.56%. The values of gears changed after shot peening processes to 1810MPa from 1580MPa. [5]

You Lv investigated the effect of micro shot peening on the fatigue behaviour of Laser melted W6Mo5Cr4V2 steel gear. Both the gears were used for study by Laser surface melted (LSM) and Laser surface melted and micro shot-peened (LSMMP). The materials used for shot particles were high carbon steels and ceramics. Different materials were used for shot accordingly with different peening time, ranging from the 20s to 100s. The fatigue behaviour was tested using the Forschungsstelle für Zahnräder und Getriebebau (FZG) back-to-back spur gear test machine. The results concluded that the fatigue strength of micro shot-peened gears was more as compared to laser surface melted gears. The properties of LSMMP did not change with an increase in peening time. [6]

Mao investigated effect of temperature and load on the polymer composite gear. The gears used for testing were polymer composite gears running against each other. Acetal and composite (55% nylon, 30% glass fibers, and 15% PTFE as an internal lubricant) were used. The experimental setup was provided to get the wear and endurance limit of the gear. The experiment was carried out with different temperatures, different load capacity, and different geometries of gear. A critical torque value was calculated with the help of its surface temperature. The results concluded that the gear performance was totally dependent on the load capacity. With increase in torque, wear rate increased. The wear rate of gear was low when it was operated under the critical value. [7]

Senthilvelan investigated the effect on the unreinforced and glass fiber reinforced Nylon 6 spur gears due to the rotational speed. A power absorption-type gear test setup was used to check the gears at various speeds i.e. 600,800,1000,1200rpm, different torque levels i.e. 1.5,2,2.5,3Nm and different stress levels i.e. 8,15,20,25,30MPa respectively. The results concluded that with an increase in rotational speeds, the stress increases, and surface temperature of the gear increases, which causes weakening in gear due to which the gear life decreases. The stress rates were small at low speeds and the gear and gear root cracking were considered to be the only failure modes. [8]

Masaya Kurokawa investigated the behaviour of various types of carbon fiber reinforced polyamide gears. The materials used in this paper were Polyamide 12(PA12), Polyamide 6(PA6), Polyamide 66(PA66), Polyamide 46(PA46) with carbon fiber reinforced Polyamide, and in the presence of grease. The test specimens, i.e., the pellets, were dehumidified at 80^oC for 24h and then injection molded into a gear. The gear testing apparatus was used to investigate the behaviour. The apparatus absorbs the power with the help of a powder brake. About 0.5 gm of grease was applied on the teeth surfaces before starting the test. Many properties of gears were investigated in this paper, like load-bearing capability, wear properties, sound properties. The results concluded that PA 12/CF gears were found superior to other gears. Such gears had the highest load-bearing potential and the wear property was found excellent in these gears, the sound properties were also considered to be the strongest as they were the most quiet during the engagement. [9]

Anand Mohan investigated the bending fatigue of symmetric gears $20^o/20^o$ and asymmetric gears $20^o/34^o$ of unreinforced and 20% glass fiber reinforced polypropylene materials with the help of servo-hydraulic fatigue testing machine. The teeth were provided with constant loading and deflection. The results concluded that the bending load capacity increased due to the addition of glass fibers, and the load-carrying capacity of $20^o/34^o$ asymmetric gears was better than $20^o/20^o$ symmetric gears, due to more gear tooth width at the root region. [10]

Dhanashekaran investigated the wear characteristics of sintered spur gear under unlubricated conditions. A MATLAB program was developed to check the changes occurring on the contact point of gear per mesh cycle. Wear prediction was made with the help of Archard’s wear formula. The gears used for investigation were Fe-C-Cu, Fe-C-Cu-3%MoS_2, and Fe-C-Cu-5%MoS_2. The test was conducted on the power absorption-type gear test machine. The gears were subjected to different torques ranging from 1 to 2.5 Nm with the rotational speed of 800rpm until tooth breakage or $20x10^4$ cycles. Experimental results were validated with the predicted methodology. The results concluded that the wear was maximum at the dedendum of the tooth. The addition of MoS_2 improved the wear characteristics of the gear like improvement of part density, hardness, strength, and wear resistivity. And the results were quite similar to the predicted results. [11]

Ken Mao investigated the wear performance of non-reinforced Polyoxymethylene (POM), and 28% glass fiber reinforced POM (GFR POM). The gears are manufactured using injection molding. A Polymer composite gear test rig was specially designed for the test. The difference in the performance of both types of gears was observed. Both the gears were running against each other, i.e., Polymer against polymer. The test was carried out with POM gear pairs that were provided with 3Nm initial torque, and this torque keeps on increasing with 0.5Nm with every 20000 cycles, and based on that, results were taken into consideration. For GFR POM, initial torque was taken as 6 Nm, and increased by 1 Nm for every 20000 cycles, and results were observed. The results concluded that 28% of GFR POM showed better performance compared to non-reinforced POM, with about 50% better load capacity. The wear rate of non-reinforced gears increased at high speed above transition load because gear operating temperature reached to the material melting point. [12]

3

Problem description

3.1 Problem description

Gears are the most important component for transmission of power and torque. They are used for many applications from clocks to aerospace industry. From the reviews, it is found that researchers were keen to find the best material for gears for various applications. Various researches have been done to predict the stress levels in the gear under various loading conditions. They had used various methods to improve the strength of gear. They used lubricants and other methods as well. Various researches have been done on metal gears, plastic gears and many other kind of gears. Basically the research was based upon the experimental setup. Very few researches have been done with the help of software analysis. Stress analysis, thermal analysis and other analysis was generally done for the gear. Now-a-days software analysis is coming as the better way to analyse stresses and other factors of any part when it is provided with any loading or other parameter. In this research, we have used the software based approach to analyse the fatigue life of Nylon gear. We have also used the mathematical approach to find out the contact and bending fatigue strength of gear. Nylon gear is proved to be the best material for the low loading conditions without getting wear out. We are calculating the stresses which are generated on the nylon gear with the help of mathematical calculations and the same stresses we are going to find out using the ANSYS software.

4

Research Work

4.1 Mathematical calculation

We are going to model a nylon gear for around 100W power with 1200 rpm.

Therefore, Angular velocity (ω) is given by $2\pi n/60$.

$\omega = 2\pi x1200/60=40\pi$ rad/sec

Torque is given by Power/Angular velocity.

$T= P/ \omega= 100/40 \pi= 0.795$Nm= 0.8(approx.)

From KHK gear catalogue, the module series is 1, 1.5, 2, 2.5, 3.

We are assuming the module m=1.5

From the KHK gear catalogue, for 1.5 m, the number of teeth series is 15, 16, 18, 20, 22, 24, 25, 26, 28, 30…

We are assuming teeth=22

For 22 teeth gear of nylon the allowable torque should be in the range 0-4.99Nm.

And according to our calculated torque it was 0.8Nm (approx).

So, our torque ranges in between 0-4.99, hence our selection is safe.

The equations have been calculated with the help of various factors.

- Allowable tangential force (F) at the pitch point can be obtained from lewis formula, i.e. :-

  $F= myb\sigma_b f$ (Kgf)

  Where, m is the module(mm)

  y is tooth profile factor

  b is face width(mm)

  σ_b is the Allowable bending stress(Kgf/mm^2)

  f is the speed factor

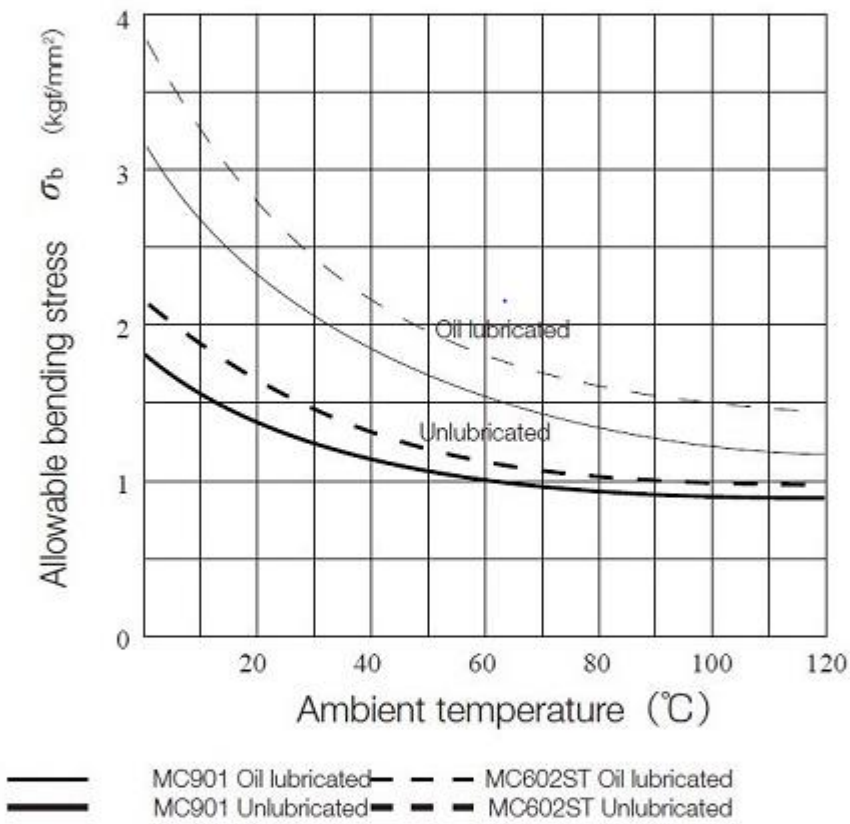


Fig.1: Allowable bending stress

In fig.1, we will be finding the value of allowable bending stress based on the ambient temperature under lubricated or unlubricated condition.

As we are taking MC602ST unlubricated from the KHK gear catalogue and under 20^0C the value will be somewhat close to 1.6 Kgf/mm$^{2.}$

Table.1: Tooth profile factor(y)

No. of teeth	Tooth profile factor		
	14.5°	20°Full depth tooth	20°Stub tooth
12	0.355	0.415	0.496
14	0.399	0.468	0.540
16	0.430	0.503	0.578
18	0.458	0.522	0.603
20	0.480	0.544	0.628
22	0.496	0.559	0.648
24	0.509	0.572	0.664
26	0.522	0.588	0.678
28	0.535	0.597	0.688
30	0.540	0.606	0.698
34	0.553	0.628	0.714
38	0.556	0.651	0.729
40	0.569	0.657	0.733
50	0.588	0.694	0.757
60	0.604	0.722	0.774
75	0.613	0.735	0.792
100	0.622	0.757	0.808
150	0.635	0.779	0.830
300	0.650	0.801	0.855
Rack	0.660	0.823	0.881

In table.1, we are finding the value of tooth profile factor based on the number of teeth in gear and the type of angle.

In our case we are going for 22 teeth with 20^0 full depth tooth so the value for that is 0.559.

Table.2: Speed Factor (f)

Lubrication	Tangential speed m/s	Factor
Oil lubricated	Below 12 More than 12	1.0 0.85
Unlubricated	Below 5 More than 5	1.0 0.7

In table.2, we are getting the value of speed factor based on various conditions.

Tangential speed is given by, v= Rxω

Where R is the radius of pitch circle

We now that pitch circle diameter is given by, PCD=module x number of teeth

We have taken m=1.5 and number of teeth=22

Therefore, PCD= 1.5x22=33mm

Put the value of R as 33/2 mm=16.5mm in the tangential speed formula.

v= Rxω= 16.5x40 πx10^{-3}=2.07m/s.

Our tangential speed is below 5m/s for unlubricated condition, so we ill take speed factor as 1.

Therefore, we can find the value of allowable tangential force with the help of given tables.

F= mybσ$_b$f

m=1.5; b=15mm (assumed from standard catalogue)

F= 1.5x0.559x15x1.6x1= 20.124Kgf= 197.4N (max. permissible bending)

Actual bending force is given by f= Torque/Radius

Therefore, f= 0.8/16.5x10^{-3}= 48.485N

As actual is less than the maximum permissible so it is safe.

Allowable bending stress can be calculated with the help of,

$\sigma_B = \sigma_b.\ K_V.\ K_T.K_L.K_M/(C_S)$

where σ_b= maximum allowable bending stress under standard condition(Kgf/mm^2)

C_S= working factor

K_v= Speed factor

K_T= temperature factor

K_L is Lubrication factor

K_M is Material factor

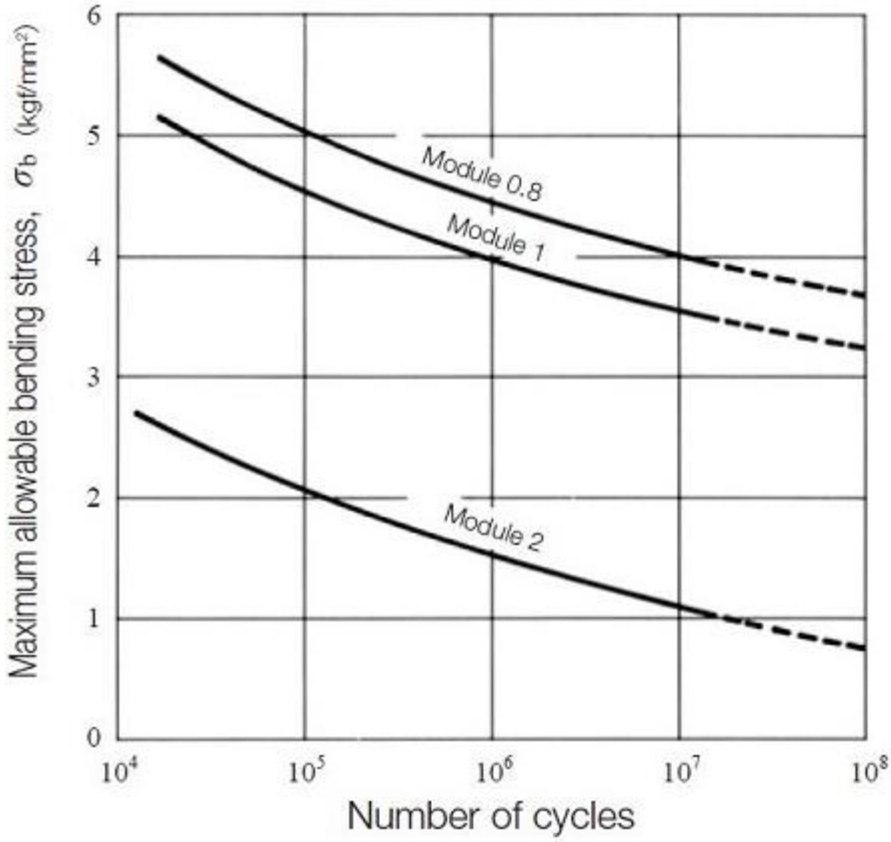


Fig.2: Maximum allowable bending stress under standard conditions

In fig.2, we are going to find the value of maximum allowable bending stress based on the number of cycles for different modules of gears.

As our module is 1.5 and we are modeling the gear for 10^5 cycles, so we will take:

σ_b= 3.3Kgf/mm^2 from fig.2.

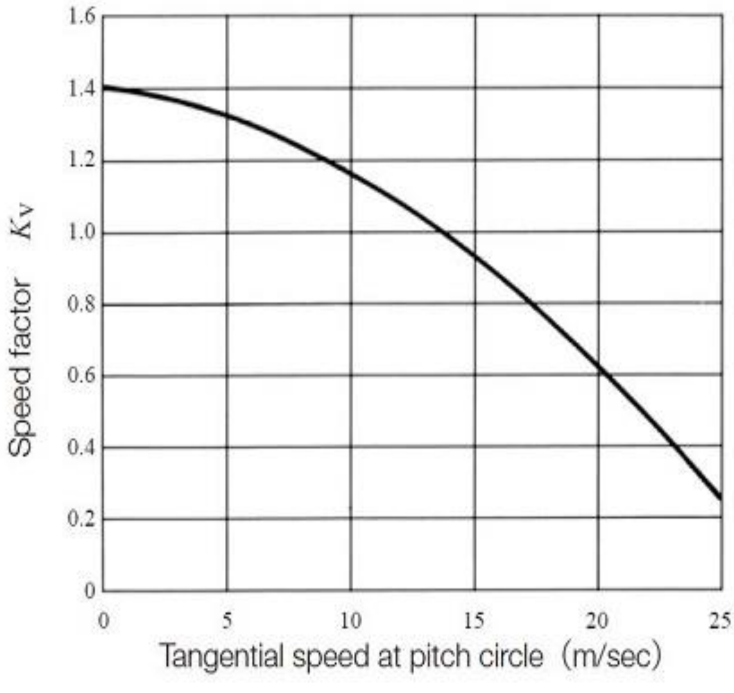


Fig.3: Speed Factor (K_v)

In fig.3, we are going to find the value of speed factor based on the tangential speed at pitch circle of gear.

Tangential speed as calculated earlier was 2.07m/s.

Therefore, K_v= 1.4 for 2.07m/s tangential speed from fig.3.

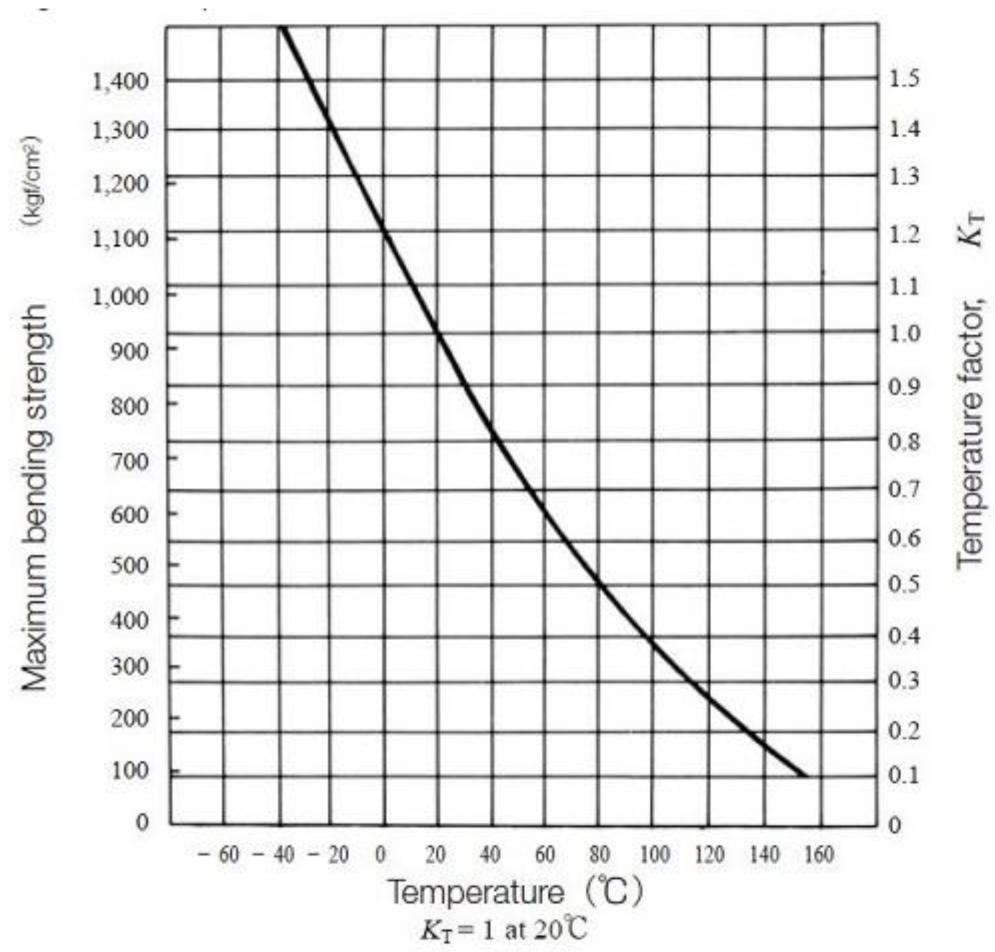


Fig.4: Temperature Factor (K_T)

In fig.4, we are finding the value of temperature factor based on the temperature at which the working is to take place.

We are calculating at 20^0C, hence the value of K_T is 1.

Table.3: Working Factor (C_s)

Types of load	Daily operating hours			
	24 hrs./day	8~10hrs/day	3 hrs./day	0.5 hr./day
Uniform load	1.25	1.00	0.80	0.50
Light impact	1.50	1.25	1.00	0.80
Medium impact	1.75	1.50	1.25	1.00
Heavy impact	2.00	1.75	1.50	1.25

In table.3, we are going to find the working factor under various types of loading and operating conditions.

We are taking the uniform loading condition for 8hrs/day.

So, C_s=1 from table.3.

Table.4: Lubrication Factor (K_L)

Lubrication	K_L
Initial grease lubrication	1
Continuous oil lubrication	1.5 − 3.0

In table.4, we are going to get the value of lubrication factor based on these conditions.

We are taking initial gear lubrication only.

Therefore K_L=1

Table.5 Material Factor

Material combination	K_M
Duracon with metal	1
Duracon with duracon	0.75

In table.5, we are getting the value of material factor based on the type of engagement of material of pinion and gear.

Therefore K_M=1

With the help of given data from the tables we can find out the value of allowable bending strength, i.e.

$\sigma_B = \sigma_b.\ K_V.\ K_T.K_L.K_M/(C_S)$

$= 3.3x\ 1.4x1x1x0.75/(1)$

=3.465Kgf/mm^2

=3.465x9.8 N/mm^2

=33.99N/mm^2 = 34 MPa(approx.)

Bending Stress for the given gear pair is 34MPa.

Now, we will have to calculate the contact stress for the given gear.

The contact stress can be calculated using the following equation.

$$S_C = \sqrt{\frac{F}{bd_{01}} \frac{i+1}{i}} \sqrt{\frac{1.4}{(1/E_1 + 1/E_2)\sin 2\alpha}}$$

Where, S_c is the contact stress generated

F is the tangential force in Kgf

b is the face width in mm

d01 is the pitch diameter of pinion

i is the gear ratio i.e. i= Z2/Z1

E is the modulus of elasticity of nylon in Kgf/mm^2.

α is the pressure angle.

we have the following values:

b=15mm

d_{01}=33mm

i=1

α=20^0

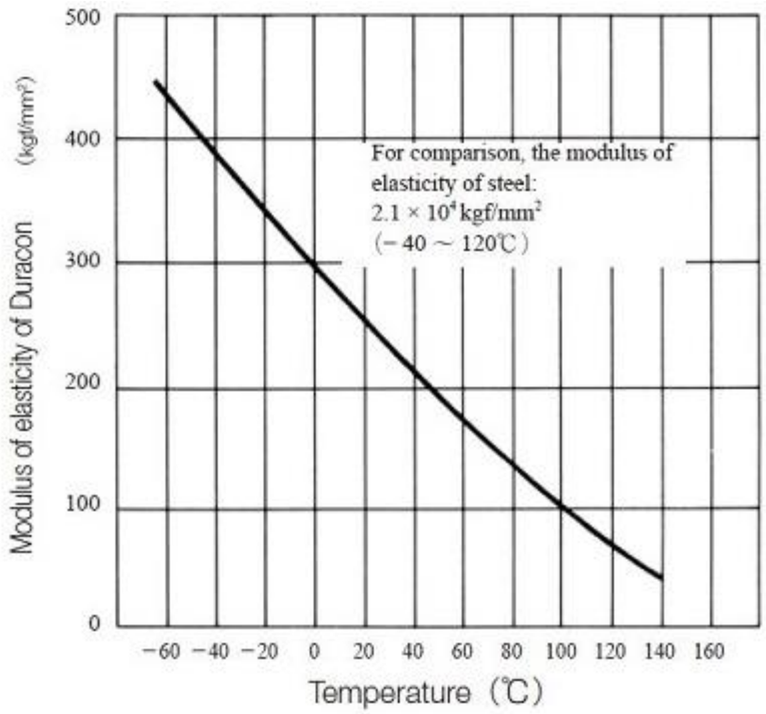


Fig.5: Modulus of elasticity (E)

From fig.5, we can calculate the value of Modulus of elasticity under ambient temperature.

We have taken 20^0C and for that the value of modulus of elasticity is $240Kgf/mm^2$.

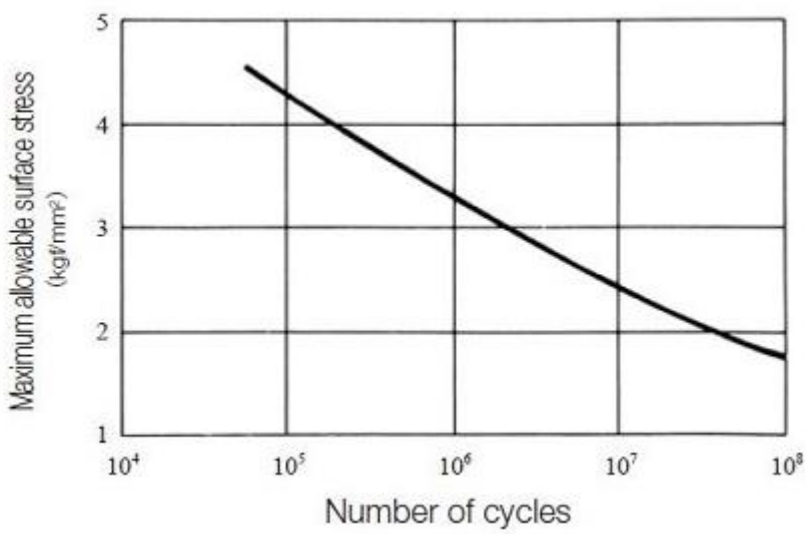
Actual F=50N=50/9.81 (Kgf)=5.1 Kgf

After putting all these values in the equation of contact stress we get,

$S_c=1.64Kgf\backslash mm^2$

$=1.64x9.8N/mm^2$

$=16.1N/mm^2=16.1MPa$



. Fig.6: Maximum allowable surface stress

From fig.6, we can calculate the maximum allowable bending stress for 10^5 cycles which is $4.2Kgf/mm^2 = 41.2N/mm^{2.}$

$= 41.2MPa$

We got surface stress from calculation is $1.64Kgf/mm^2$, which is less than maximum allowable bending stress ($1.64<4.21Kgf/mm^2$), hence it is considered as safe.

4.2 Modeling of Gear

The gear has been modeled with the help of CATIA software, which is generally a designing software with the given gear specifications.

The gear dimensions have been taken from the catalogue of KHK gears accordingly for the power transmission of 100W at 1200 rpm with torque of 0.8Nm. And based on that the dimensions have been selected.

Gear Specifications:-

- Pitch circle diameter= 33mm
- Addendum= 36.6mm
- Dedendum=29.4mm
- Teeth=22
- Module=1.5
- Face width=15mm
- Shaft diameter=15mm

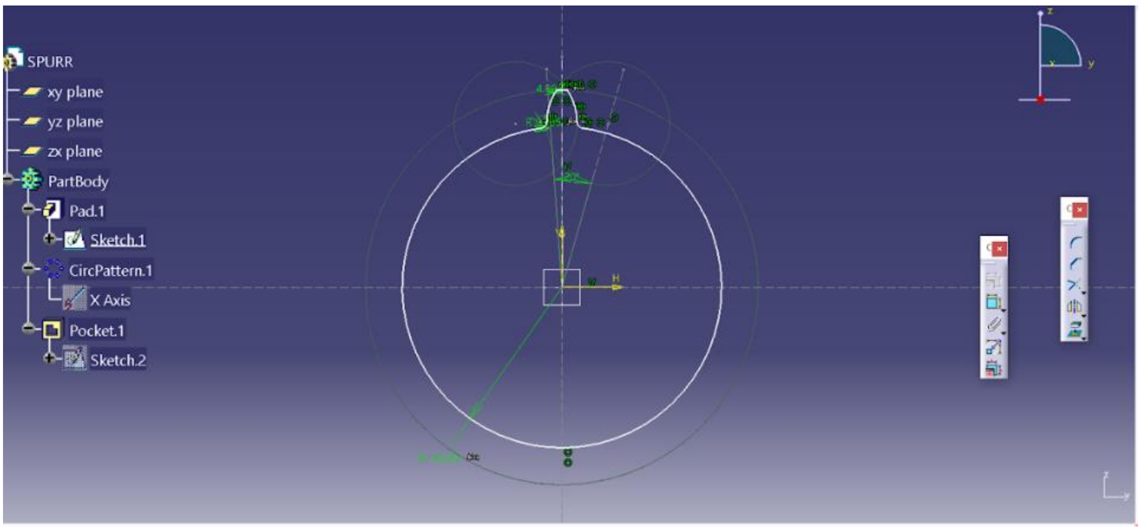


Fig.7: Sketch of gear with 1 tooth profile

This is the general sketch for one of the tooth profile which is created using the given dimensions. After preparing one of the tooth profile, opt for circular pattern in the

CATIA menu after extruding the teeth, we will get the gear with desired number of teeths.

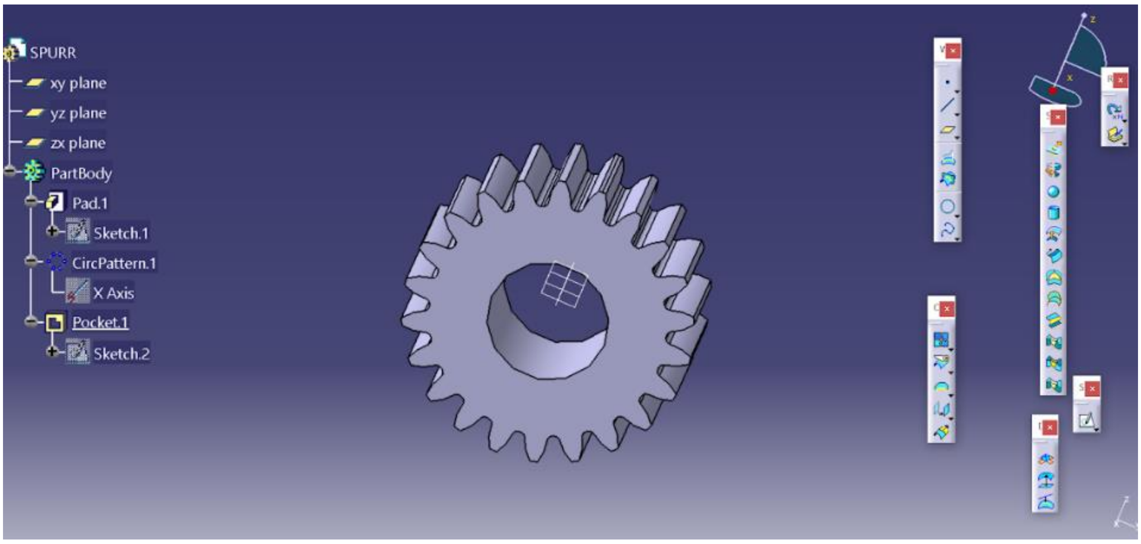


Fig.8: 3D view of the designed gear

This is the 3D view of the 22 teeth gear which is designed using the gear specifications given earlier. This profile is generated after giving the circular pattern to the 1 tooth profile which was generated initially.

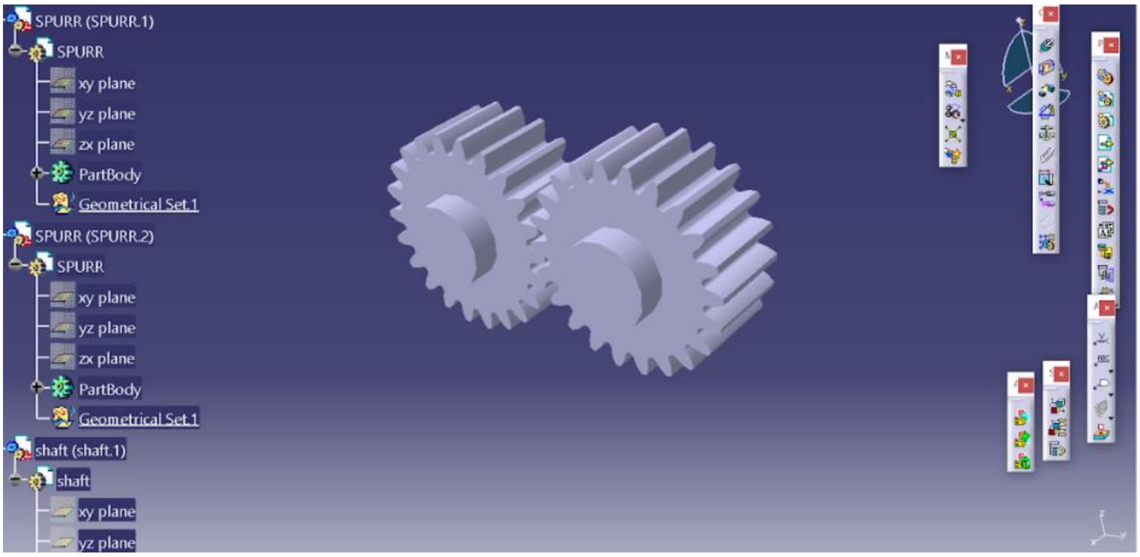


Fig.9: Assembly of gear

This is the assembly of gear pair, which is to be imported to the ANSYS software for further analysis. The gear and pinion are made of same material and same size for the analysis.

4.3 Analysis using ANSYS

The assembled gear is imported to the ANSYS software for further analysis to take place. ANSYS software can carry out various types of analysis which are required for any part or body under different circumstances.

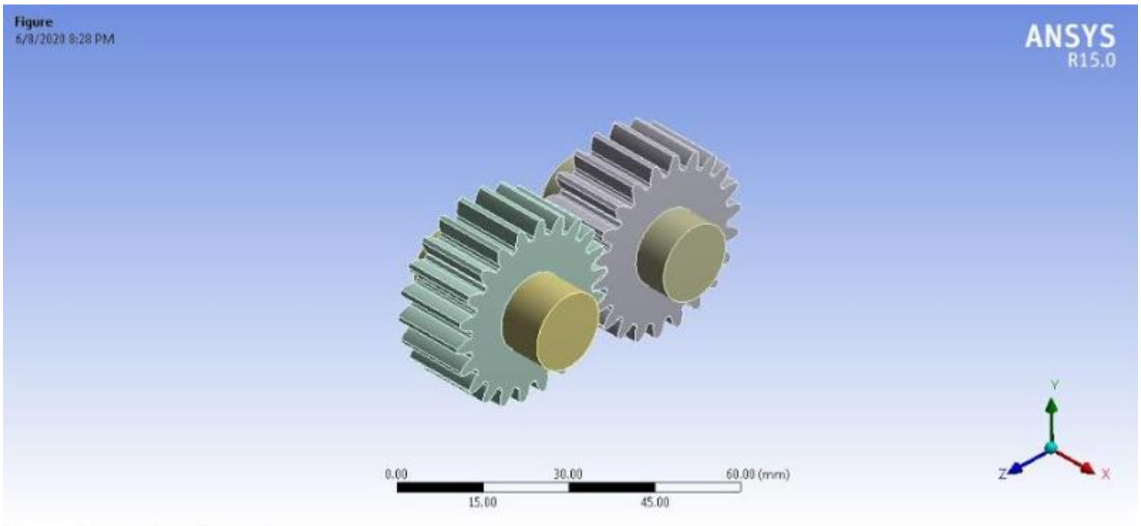


Fig.10: Model view of assembly of gear

Fig.10 is the model view which is generated after the assembly of gear is imported in the ANSYS using the .igs file.

After importing the file, we have to provide the boundary conditions to the assembly, i.e. which part is taken as support, which part will move or rotate and what are the loading parameter. We have taken shaft as the support and a load of 50N is applied to the teeth of gear as shown in fig.10.

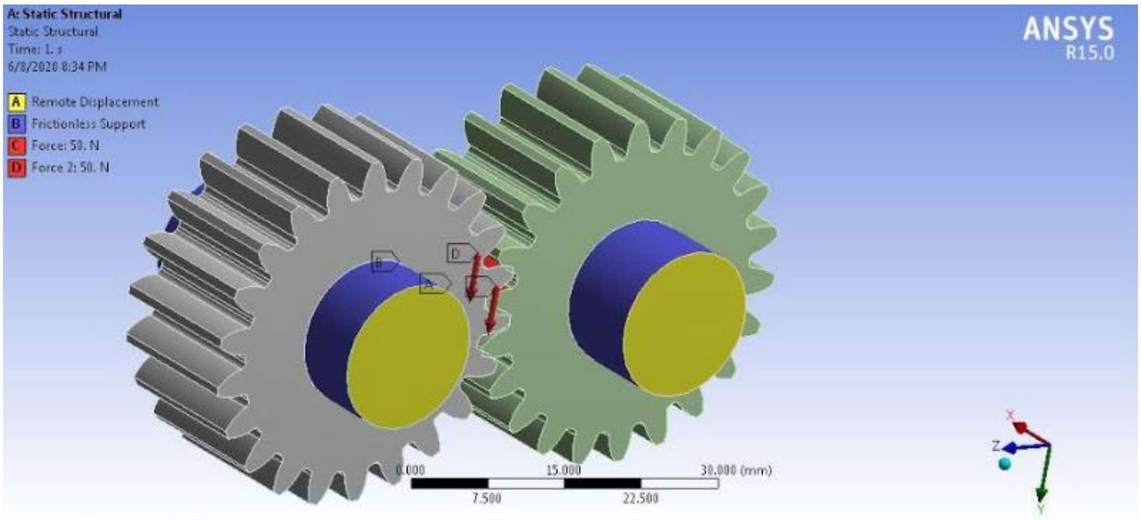


Fig.11: Boundary conditions

After providing the boundary conditions we will have generated the mesh of the gear to be fine, so that the results are supposed to be more accurate.

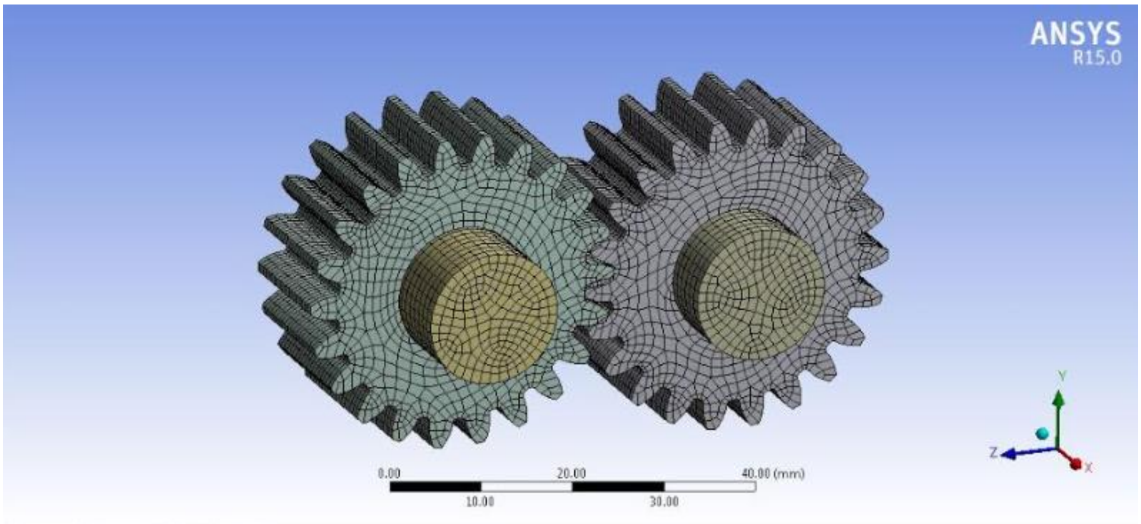


Fig.12: Mesh model of gear

Fig.12 is the meshed model of assembled gear which is generated after providing the fine mesh option in the refinement menu.

After meshing, we have done various analysis to calculate the life, factor of safety, deformation and stresses that are generated on the tooth contact region.

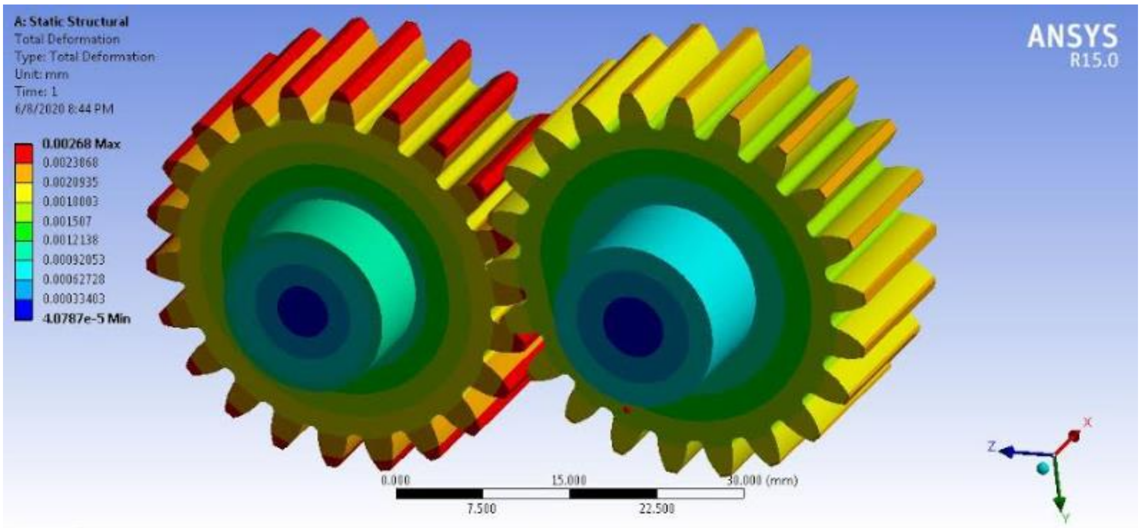


Fig.13: Total deformation

Fig.13 shows the total deformation which is generated on the gear tooth under working conditions. The maximum deformation that can be generated on the gear is 0.00268mm.

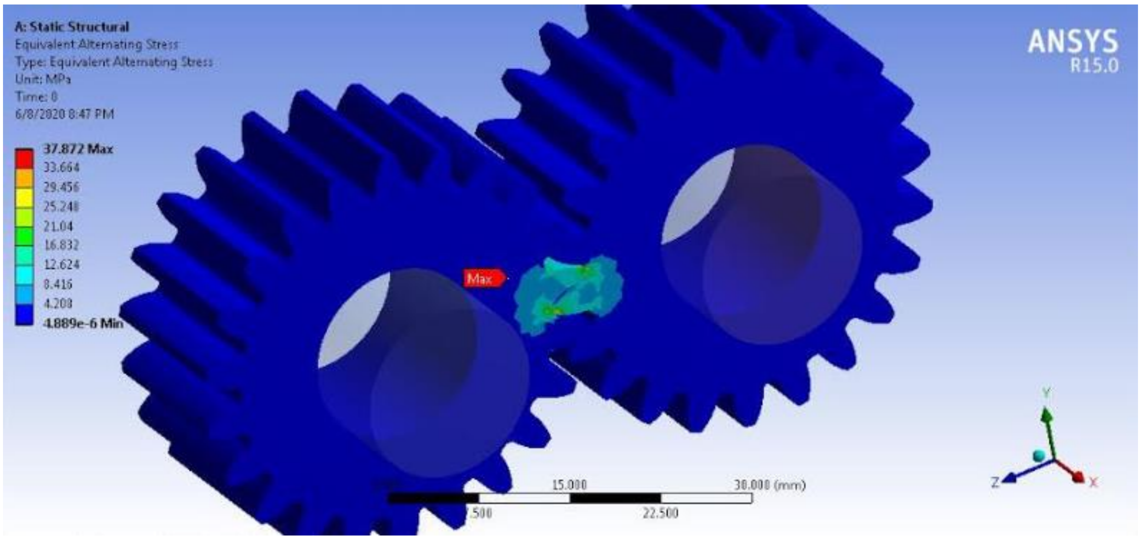


Fig.14: Contact stress

Fig.14 shows the stress generated on the contact region while mating of two gears under working conditions. The maximum allowable contact stress generated on the contact is 37.872MPa.

Now after calculating the stress, we have found the factor of safety and the life.

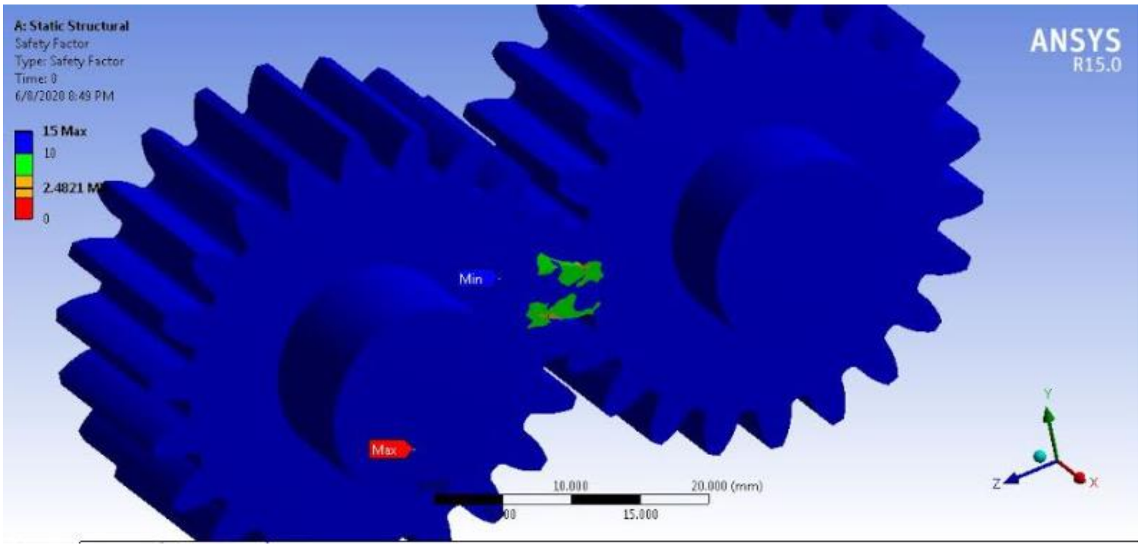


Fig.15: Factor of safety

Fig.15 shows the factor of safety of gear under working condition. The factor of safety for the designed nylon gear is 2.4821.

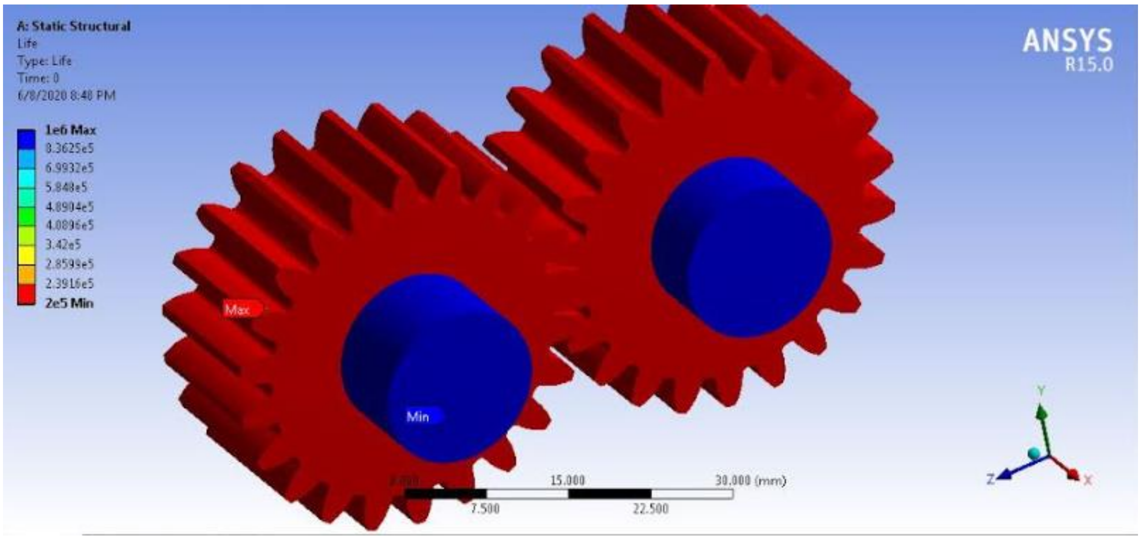


Fig.16: Life of gear

Fig.16 shows the life of gear under cyclic load of 50 N. The life of the designed nylon gear is $2x10^5$ cycles.

5

Result and Discussion

We have calculated the dimensions of the nylon gear with the help of standard gear catalogue and have modeled the gear with the help of those specifications in the CATIA software and have calculated the bending stress and contact stresses of the nylon gear, manually with the help of the tables and data provided in the KHK gear standard gear catalogue. After applying the given formulae and data we have got the value of maximum allowable bending and contact stress as 34MPa and 41.2MPa

On the other hand we have calculated the stress in the designed Nylon gear with the help of ANSYS software. The result for the total deformation was found to be 0.00268mm while the contact stress was found to 37.72MPa. The factor of safety and life was also calculated for the gear. The factor of safety was found as 2.48 while the life is calculated as $2x10^5$ cycles.

From the results it is clear that the calculated contact stress and the software generated stress of the gear under similar conditions was found to be really close to one another.

6

Conclusion and Future Scope

Based on the results we can conclude that we can also calculate the bending and contact strength of gears with the help of manual calculations using various factors. And can use lubricant and other things to improve their properties.

Future scope of work in this is, we can take some other gear material as well to calculate their fatigue strength. Different materials can be considered for analysis. Composite material, alloys can also be considered.

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