DESIGN, ANALYSIS AND OPTIMISATION OF PERFORATED NYLON 6,6 SPUR GEAR.

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Abstract - *Gearing is one of the critical components in a mechanical power transmission system and used in most industrial rotating machinery. The general spur gears are made up of steel or cast iron. However, these gears when operating at high speeds generate a lot of noise and their teeth experience great amount of stress. Also, these materials increase the weight of the machine. These problems can be overcome by using plastic materials (Here the material incorporated for the operation is Nylon 6,6). These materials though lightweight tend to have deformation and stress accumulation when operating at high loads and speeds because of plastic properties. Perforation is one of the methods used to rectify these issues. In this project, perforations in the system are used to decrease the stresses and analyze the stress distribution before and after adding cooling holes.*

The theoretical calculations are done by calculating various parameters required for gear design. In the next stage, modelling and different iterations of analysis are performed. The obtained analysis results are used to find the region of least stress accumulation and hence achieving the required properties. The modelling and analysis software's used in the project are Solid Works for modelling and Ansys for analyzing these models respectively.

Key Words: Spur gear, Nylon 6,6, Static Analysis, Dynamic Analysis, Perforation, Strength to weight ratio.

1. INTRODUCTION

Power transmission is an important section of any running machinery as the efficiency of the machine is highly dependent on the amount of power lost during the process. Gears are one of the elements having high priority in a setup, used for power transmission.

Usage of plastic gears are however restricted by poor mechanical properties and lower temperature limits and heat conduction limitations. To find the solution for these issues, new methods were developed which helped to limit the damage of the plastic gear surface in contact for the plastic gear tooth. The major thought behind this was to minimize the extent of failure of the gear by decreasing the heat which gets accumulated in the tooth during regular operations. Small holes known as Perforation were drilled, with aim to reduce the overall stresses acting on the gear teeth.

Fig 1: Spur gear

This project is limited to the study of spur gears and their analysis. Spur gears are gears which have teeth cut in straight direction, and parallel to the axis were rotation takes place. They are the simplest type of gears in use and their tooth profile is generally having an involute shape.
Although spur gears can be used at speeds (as high as other types of gears) they are kept limited in order to avoid high-frequency vibration and unacceptable noise levels.

They are employed in various industry operations. They transmit power through shafts which are held parallel to each other. They produce no axial loads but lead to the gears producing radial loads on the shaft. Noise produced in this case is more than helical gears because while they operate, the teeth's share only 1 point where they touch each other at any given time. While they keep rolling while meshing with each other, they shift off the point of contact from one tooth and move to engage in contact with simultaneous tooth. It is the major difference that they share with helical gears, which share more than one tooth in simultaneous contact and torque is transmitted by them in a smoother fashion. These are also the most easily imagined gears. Their shape leads them to be classified as one type of cylindrical gears. Due to less difficulty during production, these gears when produced are much more précised.

2. LITERATURE SURVEY

This chapter contains the overview of major writings and resources available regarding the work carried out by researchers in the area of perforated spur gear manufacturing. The chapter consists of brief description, summary and conclusion of each source.

1. **Hayrettin Du¨zcu¨kog˘lu et al** – For analysing this topic in a broader perspective, perforations were made at different points for reducing damage caused by the generation of heat in the point of contact. Their purpose was to reduce the heat by ousting it to atmosphere. By this process, life cycle is increased, and the damage is decreased.

2. **Demagna Koffi et al -** This paper selects the best possible orientation of perforation on the surface by the process of comparison between different available options. Simulations are incorporated to analyse different possible aspects.

3. **Huseyin Imrek -** Due to poor thermal coefficients of plastic gears, they tend to perform very poorly at high operating temperatures. This paper discusses the change in various gear designing parameters and analyses their effect on the gear performance.

4. **S. Mahendran et al** – Studies were made considering the aspects of reducing weight and their distribution in gears made from different materials. Based on the study, the analysis of wear was performed. The analysis portrays that the stress present in composite materials is lower when kept parallel to a gear made of metal.

5. **K. Mao -** concluded that Rate of wear increases after the material reach a certain point during operation. The same rate is slow when the operation is conducted below the certain limiting point. A viable cause for sudden increment in wear rate is due to the gear operating temperature reaching the material melting point under the critical load condition. The rate of material wear is highly dependent on the amount of loading done on the setup. Quick change to higher rates were found as he increased the torque to the given value. It primarily happens due to the variation in the temperature on the objects surface.

3. RESEARCH GAP

Papers existing on the analysis of Spur Gears which are made up of traditional materials and various parameters which effect the material selections are quite lot in number. However, very few papers exist on the usage of cooling holes in non-traditional plastic materials as they are still in the development stage.

In this paper consists of both static and dynamic stress analysis on the gear setup along with comparison before and after putting cooling holes. Also, instead of going with general assumption that stress concentration will be more in the pitch circle region (middle portion); we firstly perform different iterations of perforations by varying both the dimensions and positioning of holes and simultaneously finding out the regions of least stressed-out regions.

4. OBJECTIVES

1. To reduce the weight of gears and increase the strength to weight ratio by using nylon material.
2. Performing different iterations based on various sizes and positions of cooling holes. Also, analysing the pattern of stress accumulation across different positions where the cooling holes are added and finding out the position giving least amount of stress accumulation.

5. METHODOLOGY

- Material selection is done after going through the requirements of a spur gear. The various properties effecting gear parameters have been explained in the next chapter.
- The various parameters required for gear designing are calculated theoretically
- The gear is modelled on SolidWorks using the parameters calculated theoretically.
- Gear analysis is performed on Ansys to analyse the static and dynamic stressed regions.
- Cooling holes are added on the spur gear to relieve the stresses from the gear.
- Different iterations are performed in terms of change in position as well as dimensions. This helps in figuring out the trend followed by stress accumulation as well as deformation in the spur gear set.
- The results from analysis are tabulated and comparisons are made.

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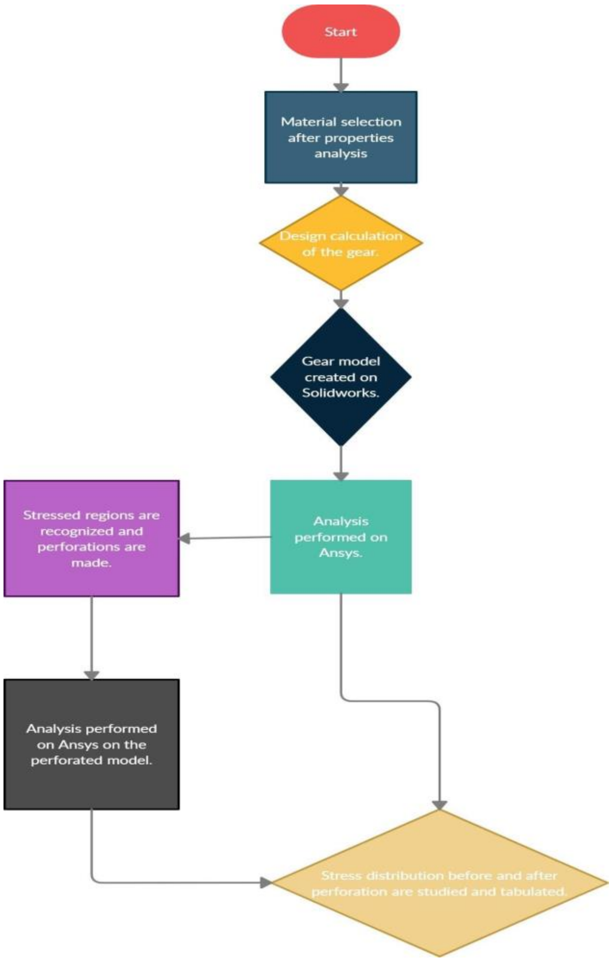


Fig 2: Methodology Flowchart

6. DESIGN CALCULATIONS

Specifications

Vehicle	TATA Super Ace
Engine Capacity	1405cc
Engine Torque	13.8 Kg-m at 2500 rpm

Assuming

Factor of safety (FOS)	3
Pressure Angle (α)	20°
Number of teeth (z)	18
Module (m)	12

Abbreviations

Symbols	Descriptions and Unit
α	Pressure Angle (degrees)
σ_{all}	Allowable stress (N/mm2)
σ_b	Bending stress (N/mm2)
σ_{sf}	Contact stress (N/mm2)
υ_p	Poisson's ratio (smaller gear)
υ_g	Poisson's ratio (larger gear)
b	Face width (mm)
d	Diameter of pitch (mm)
d_b	Diameter of base circle (mm)
d_g	Gear diameter (mm)
d_p	Pinion diameter (mm)
E_g	Young's Modulus of gear material (N/m2)
E_p	Young's Modulus of pinion material (N/m2)
F_r	Force in radial direction (N)
F_t	Force in tangential direction (N)
h_a	Outer circle (mm)
h_f	Inner circle (mm)
m	Module (mm)
N	Speed of gear (rpm)
p_c	Circular Pitch (mm)
P	Power output of engine (kW)
r	Fillet Radius (mm)

6.1 GEAR PARAMETERS AND CALUCALTIONS

1) Circular Pitch (p_c):
$P_c = \pi \times m$
P_c = **37. 699 mm**

2) Base Circle Diameter (db):
$d_b = d \cos \alpha$
$d_b = 216 \times \cos 20$
d_b = **202. 97 mm**

3) Pitch Diameter (d):
$d = m \times z$
$d = 12 \times 18$
d = 216 mm

4) Face Width (b):
$b = 9 \times m$
$b = 9 \times 12$
b = 108mm

5) Addendum (h_a):
$h_a = 1 \times m$
$h_a = 1 \times 12$
ha = 12mm

6) Dedendum (h_f):
$h_f = 1.25 \times m$
$h_f = 1.25 \times 12$
h_f = 15mm

7) Clearance:
Clearance = $0.25 \times m$
Clearance = 0.25×12
clearance = 3mm

8) Fillet Radius (r):
r = 0.4m
$r = 0.4 \times 12$

r = 4. 8mm

9) Backlash
Backlash= 0mm in 20° involute gear system.

10) Tooth thickness:
tooth thickness =pc/2
tooth thickness =37.6992/2
tooth thickness = 18. 849mm

11) Torque (t):
T = (13.8 × 10)Nm
T = 138 Nm
T = 138000 Nmm

12) Power (P):
P =(2 π NT)/60
P =(2 × π × 2500 × 138)/60
P = 36. 128 KW

13) Tangential Force (F_t):
Ft =T/r
Ft =138000/108
Ft = 1277. 778 N

14) Radial force (F_r):
$F_r = F_t \times \tan \alpha$
$F_r = 1277.77 \times \tan 20$
$\mathbf{F_r = 465.07\ N}$

15) Lewis equation (σ_b):

$$\sigma b = \frac{Ft}{b * pc * y}$$

$$\sigma b = \frac{1277.77}{108 * 37.699 * .309}$$

$\boldsymbol{\sigma_b}$ **= 1.0156 MPa**

16) Allowable Stress (σ_{all}):
$\sigma_{all} = \sigma_b \times$ FOS (FOS is factor of safety=3)
$\sigma_{all} = 1.0156 \times 3$
$\boldsymbol{\sigma_{all}}$ **= 3. 0468 MPa**

17) Contact Stresses (σ_{sf}):

$$\sigma_{sf} = \sqrt{\frac{Ft * (\frac{2}{dp\ \sin\alpha} + \frac{2}{dg\ \sin\alpha})}{\pi b \cos\alpha \left(\frac{1-\vartheta^2 p}{Ep} + \frac{1-\vartheta^2 g}{Eg}\right)}}$$

$$\sigma_{sf} = \sqrt{\frac{1277.77 * (\frac{2}{216 * \sin 20} + \frac{2}{216 * \sin 20})}{\pi * 108 * \cos 20 \left(\frac{1-0.402^2}{0.850 * 10^3} + \frac{1-0.402^2}{0.850 * 10^3}\right)}}$$

$\boldsymbol{\sigma_{sf}}$ **= 10. 489MPa**

7. MATERIAL SELECTION AND PROPERTIES

Material properties of Nylon 6,6 have been tabulated below.

Table 1: Material properties

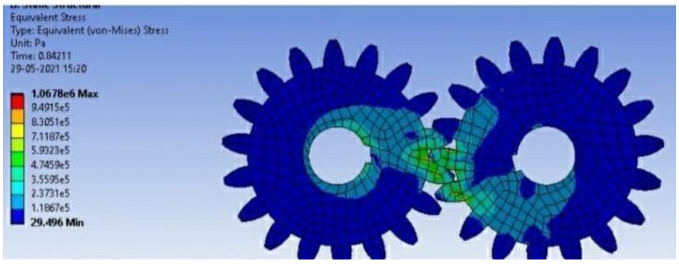
Density	1.17 g/cm^3
Coefficient of Thermal Expansion	0.00013 C^-1
Melting Temperature	280 C
Young's Modulus	0.85 GPa
Poisson's Ratio	0.402
Shear Modulus	0.303 GPa
Tensile Yield Strength	72.4 MPa
Compressive Yield Strength	63.6 MPa
Tensile Ultimate Strength	83.5 MPa

8. Analytical Results without Cooling Holes

In order to confirm about the theoretical calculations made and the approach used in the analytical process, we have performed the analysis on Ansys. The result has been mentioned below.

Fig 3: Stress in normal spur gear set without perforations

From theoretical calculations above,
Using Lewis Equation, Bending Stress obtained is
σb = 1. 0156 MPa



Similarly, Allowable Stress obtained is
σall = 3 × 1. 0156 MPa
σall = 3. 046 MPa

Ultimate Tensile Strength for Nylon 6,6=80 MPa.
So, Allowable Stress for Nylon 6,6 =80/3 MPa
=26.66 MPa

In this case,

26.66 MPa > 3.046 MPa
Hence the design is safe.

Table 2: Stresses in normal gear set

Type of Stress	Ansys Model (MPa)	Theoretical Calculation (MPa)
Bending Stress	1.0678	1.0156

The value of stress calculation made using the FEA model and analysis is accurate in the range of 95.12%. Thus, it can be concluded that element is suitable for static analysis.

9. RESULTS AND DISCUSSION

9.1 Grid Convergence Test

Grid Convergence test are usually performed to minimize the effect that the number of grids/grid size have on the end computational results. Grid convergence is used to mention the improvement in results with successive smaller cell sizes for calculations. In this case, Grid size is fixed at a position when the end results come the least. Grid convergence study is shown in the table below.

Table 3: Grid convergence table

Grid size (mm)	Max Stress (Mpa)
10	1.2207
9	1.1577
8	1.0678
7.5	1.1192
7.4	1.1451

9.2 Simulation Results for Static Analysis

The aim of simulation was to find out the effect of position change of perforations and the change in dimensions of perforations at those positions. We have selected 3-hole dimensions which are 5 mm, 7 mm and 8 mm respectively. At the same time, the different positions of perforations have been identified by keeping an offset of 5 mm between different positions. The results have been tabulated in below sections.

9.3 Stress and Deformation of 5 mm holes

Perforation (ϕ5mm hole)	Distance from Dedendum Circle (mm)	Max Stress (Mpa)	Deformation (mm)
Position 1	0	1.3285	0.10533
Position 2	5	1.3873	0.10478
Position 3	10	1.199	0.10359
Position 4	15	1.1963	0.10353
Position 5	20	1.3063	0.1047

Table 4: Results from hole diameter of 5 mm

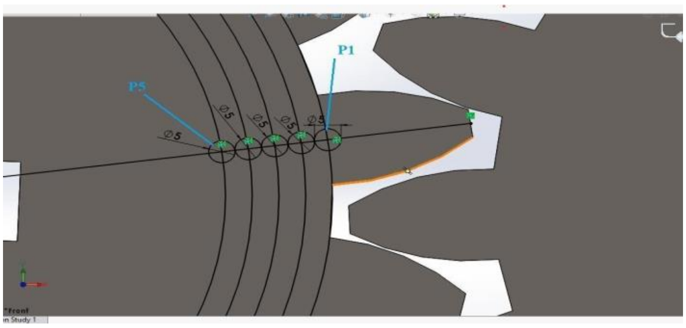


Fig 4: Positions of perforations

9.3.1 Perforations at 0 mm offset

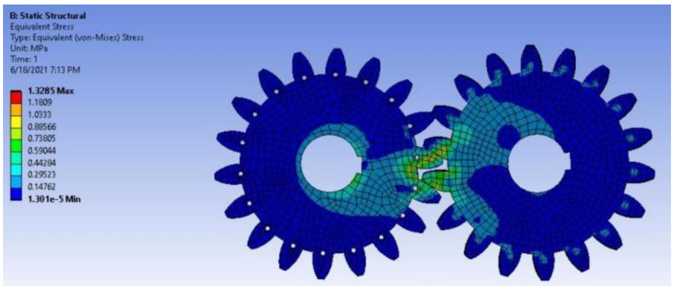


Fig 5: Stress at 0 mm offset for 5 mm hole

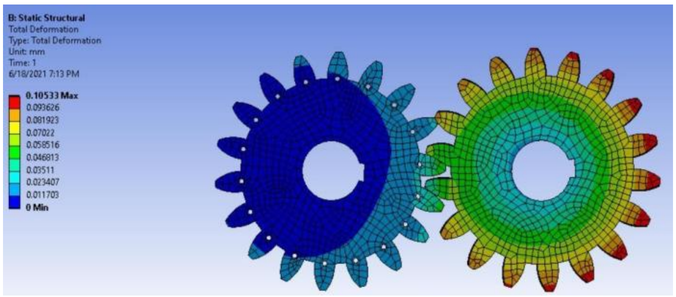


Fig 6: Deformation at 0 mm offset for 5 mm hole

9.3.2 Perforations at 5 mm offset

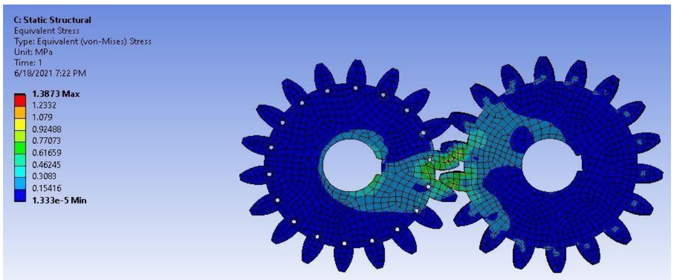


Fig 7: Stress at 5 mm offset for 5 mm hole

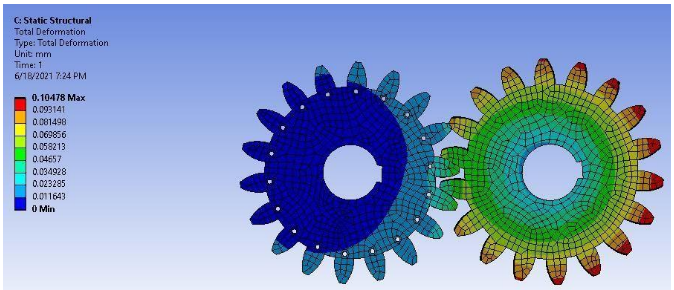


Fig 8: Deformation at 5 mm offset for 5 mm hole

9.3.3 Perforations at 10 mm offset

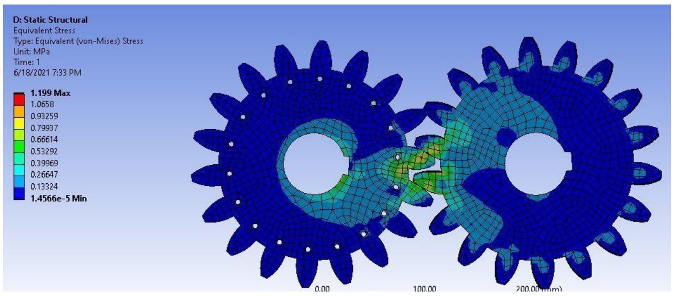


Fig 9: Stress at 10 mm offset for 5 mm hole

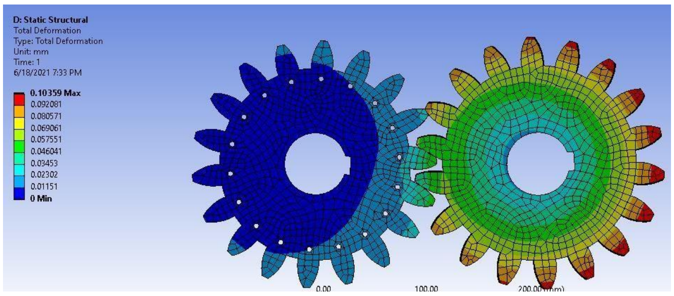


Fig 10: Deformation at 10 mm offset for 5 mm hole

9.3.4 Perforations at 15 mm offset

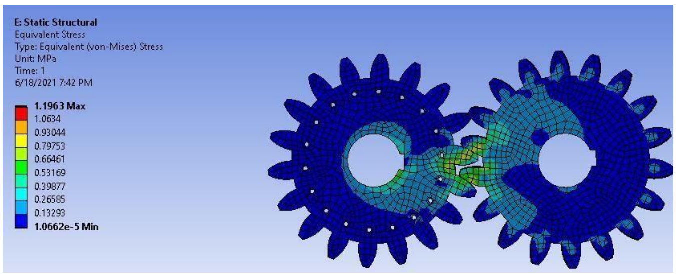


Fig 11: Stress at 15 mm offset for 5 mm hole

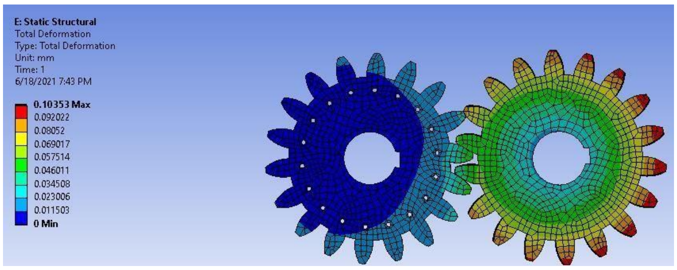


Fig 12: Deformation at 15 mm offset for 5 mm hole

9.3.5 Perforations at 20 mm offset

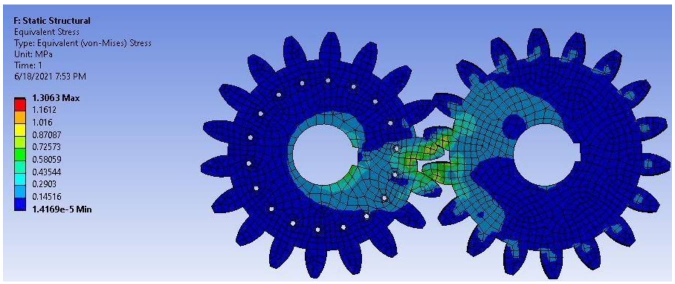


Fig 13: Stress at 20 mm offset for 5 mm hole

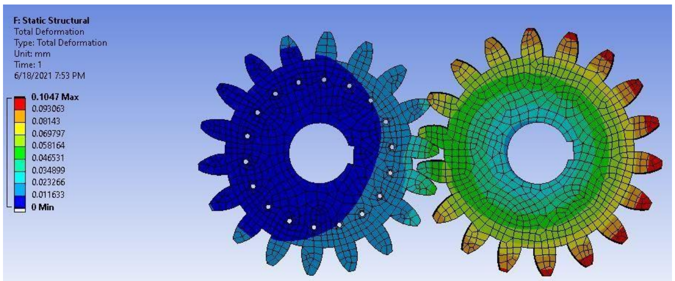


Fig 14: Deformation at 20 mm offset for 5 mm hole

The values of stresses are obtained for 5mm diameter holes at positions which are at dedendum circle, 5mm offset, 10 mm offset, 15 mm offset and 20 mm offset respectively.

The graph of the respective stress values at different positions has been plotted in a graph as shown below and it shows us the orientation of stresses as we move in different regions of the gear set.

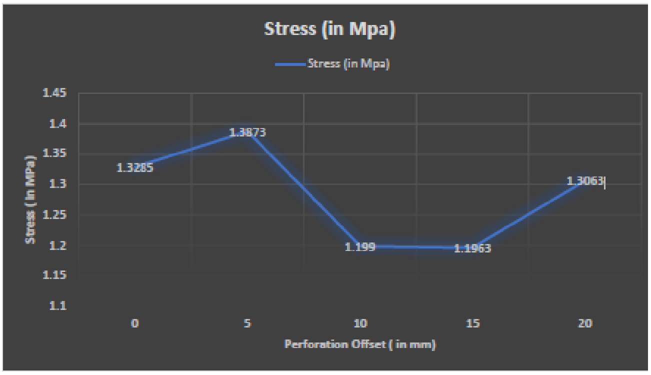


Fig 15: Stress Accumulation Graph

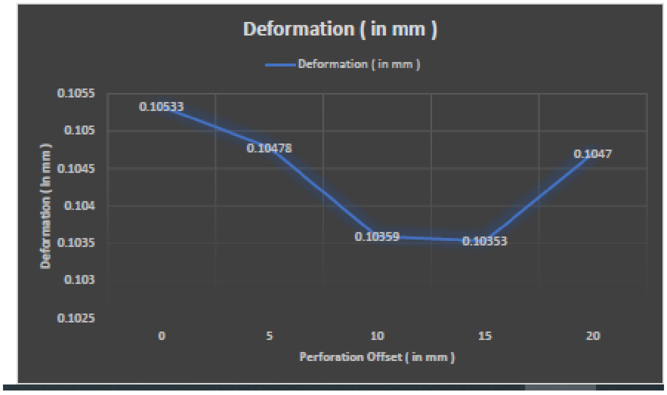


Fig 16: Deformation Graph

9.4 Stress and Deformation of 7 mm holes

Table 5: Results from hole diameter of 7 mm

Perforation (ϕ7 mm hole)	Distance from Dedendum Circle (mm)	Max Stress (MPa)	Deformation (mm)
Position 1	0	1.3265	0.10317
Position 2	5	1.2842	0.10412
Position 3	10	1.2202	0.10414
Position 4	15	1.3028	0.11357
Position 5	20	1.2224	0.10342

From above table 5 the values of stresses and deformation are obtained for 7 mm diameter holes at positions which are at dedendum circle, 5 mm offset, 10 mm offset, 15 mm offset and 20 mm offset respectively.

9.5 Stress and Deformation of 8 mm holes

Perforation (ϕ8 mm hole)	Distance from Dedendum Circle (mm)	Max Stress (MPa)	Deformation (mm)
Position 1	0	1.783	0.10608
Position 2	5	1.2684	0.10452
Position 3	10	1.2713	0.10466
Position 4	15	1.2036	0.10416
Position 5	20	1.1799	0.10381

Table 6: Results from hole diameter of 8 mm

From table 6 the values of stresses and deformation are obtained for 8 mm diameter holes at positions which are at dedendum circle, 5 mm offset, 10 mm offset, 15 mm offset and 20 mm offset respectively.

Graphs combining all the different dimensions have been plotted below to get a clearer picture in terms of their effect on the gear set.

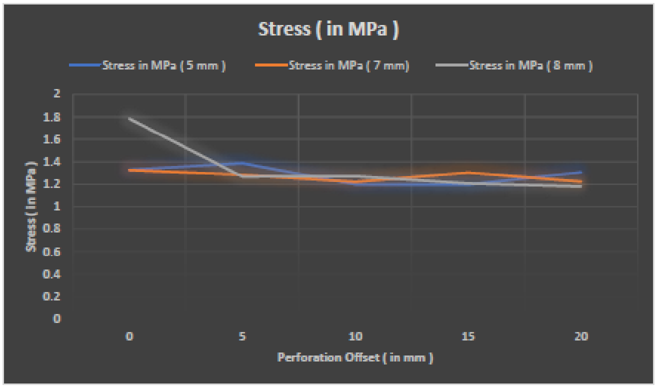


Fig 17: Combined graph for Stress Accumulation

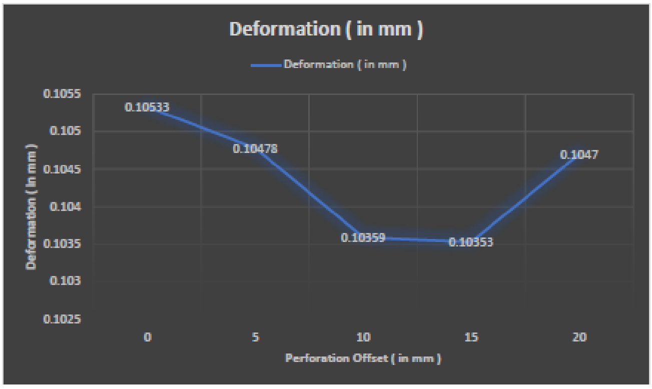


Fig 18: Combined graph for Deformation

From the combined graphs plotted above for stresses and deformations, it's observable that both of these properties have the least values in the range of 10 mm to 15 mm. Also, among all the dimensions of holes used, 5 mm diameter hole has given us the least stress value.

To find the exact location of the least stresses in the range of 10 mm to 15 mm, further iterations are performed with an offset of 0.5 mm, starting from 10.5 mm and going to 14.5 mm.

9.6 Stress and Deformation for 10.5 mm to 14.5 mm offset

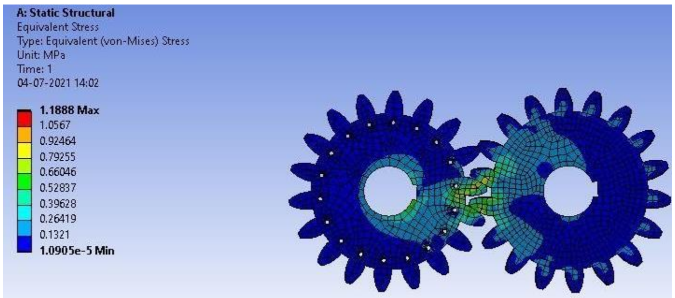


Fig 19: Stress at 10.5 mm offset for 5 mm hole

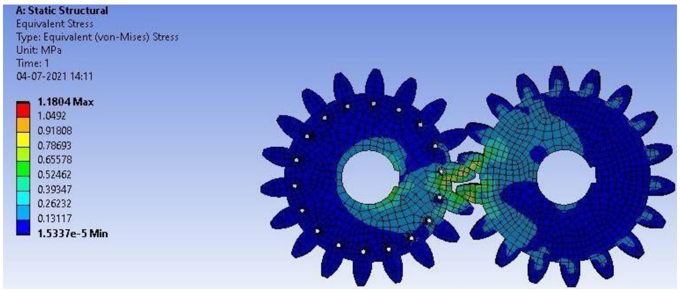


Fig 20: Stress at 11 mm offset for 5 mm hole

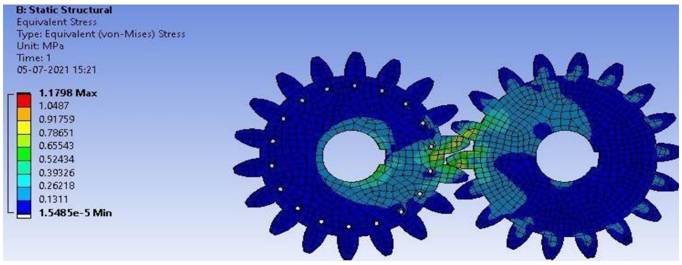


Fig 21: Stress at 11.5 mm offset for 5 mm hole

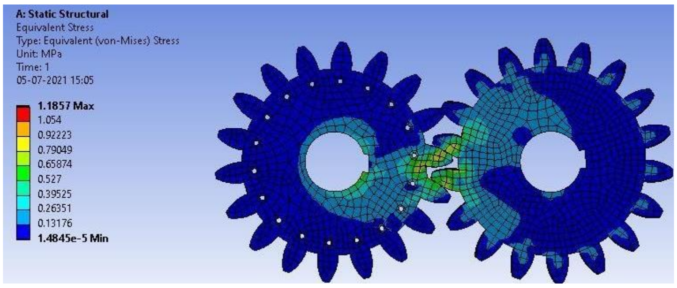


Fig 22: Stress at 12 mm offset for 5 mm hole

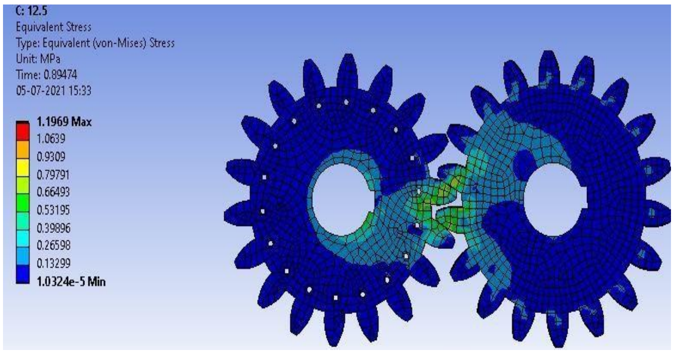


Fig 23: Stress at 12.5 mm offset for 5 mm hole

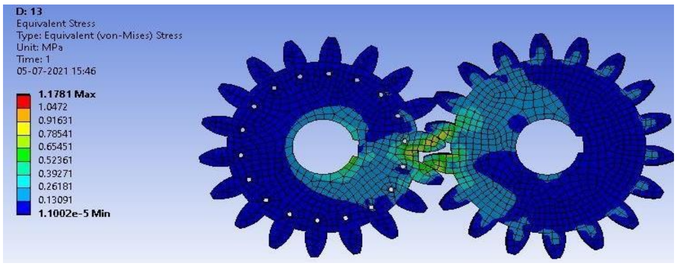


Fig 24: Stress at 13 mm offset for 5 mm hole

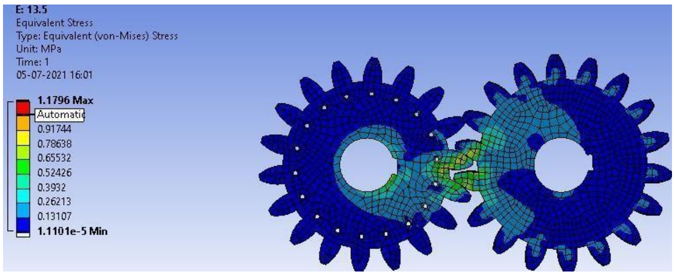


Fig 25: Stress at 13.5 mm offset for 5 mm hole

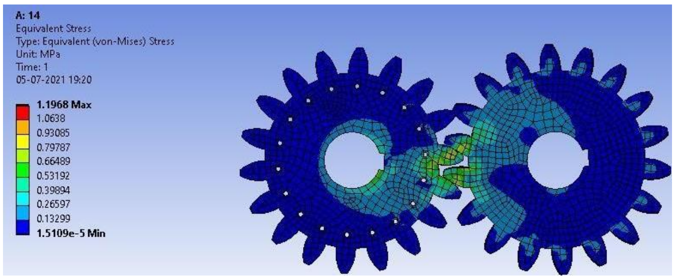


Fig 26: Stress at 14 mm offset for 5 mm hole

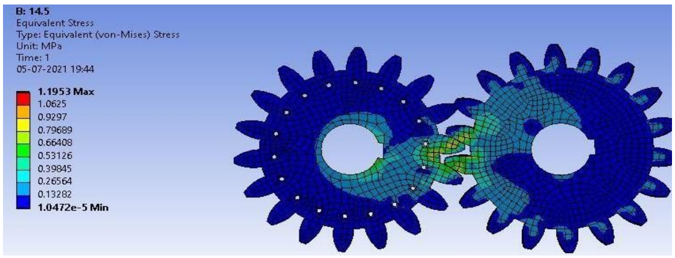


Fig 27: Stress at 14.5 mm offset for 5 mm hole

Table 7: Stress and Deformation for 10.5 mm – 14.5 mm offset with 5 mm hole

Offset	Stresses (in MPa)	Deformation (in mm)
10.5	1.1888 MPa	0.10351
11	1.1804 MPa	0.10339
11.5	1.1798 MPa	0.10326
12	1.1857 MPa	0.10326
12.5	1.1969 MPa	0.10330
13	1.1799 MPa	0.10342
13.5	1.1796 MPa	0.10327
14	1.1968 MPa	0.10298
14.5	1.1953 MPa	0.10353

The least value of stress obtained is 1.1798 MPa for 5 mm diameter hole at an offset of 11.5 mm.

9.7 Simulation Results for Dynamic Analysis

Loads which vary in terms of magnitude, direction or point of application with respect to change in time are known as Dynamic Loads. Such loads in general lead to generation of fluctuating stresses in the body. When we talk about Spur Gears in particular, the load acting on gear tooth is constant in terms of both magnitude as well as direction but varies with respect to the point at which the load is acting. Hence, the load acting on these keeps on fluctuating and leads to gear failure.

This situation makes the Dynamic Analysis very important for the development of gear set.

9.7.1 Dynamic Analysis without perforations

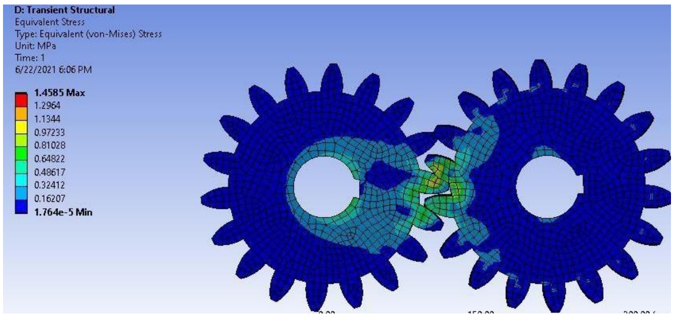


Fig 28: Dynamic Analysis stress without perforation

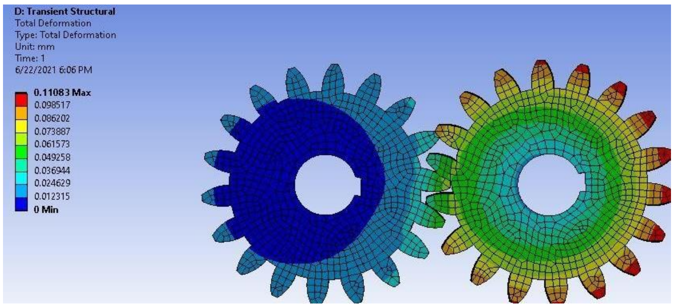


Fig 29: Dynamic Analysis deformation without perforation

9.7.2 Dynamic Analysis with perforations

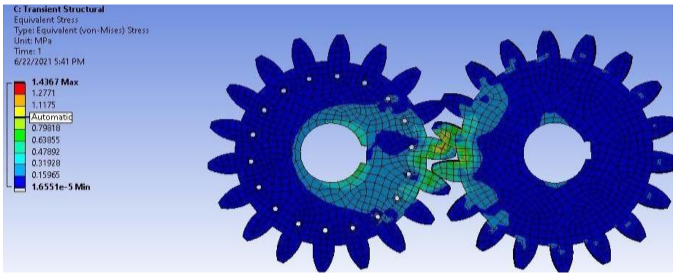


Fig 30: Dynamic Analysis stress with perforation

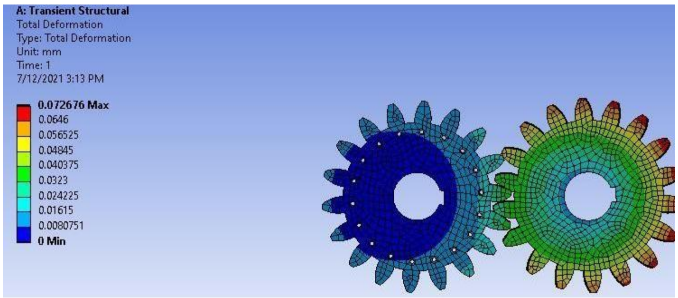


Fig 31: Dynamic Analysis deformation with perforation

Table 8: Dynamic Analysis Results

Type of gear structure	Stress (in Mpa)	Deformation (in mm)
Without Perforation	1.4585	0.11083
With Perforation	1.4367	0.07267

From the dynamic analysis performed, it's evident that both the deformation and stress have reduced in the gear combination after addition of perforations.

9.8 Weight difference between Normal gear and perforated gear

Weight is one of the critical parameters when it comes to the performance of the part. Materials like Nylon 6,6 offer lesser weighing parts compared to parts made up of metals. A comparison has been performed to find out the difference of weight between a normal gear and a perforated gear respectively.

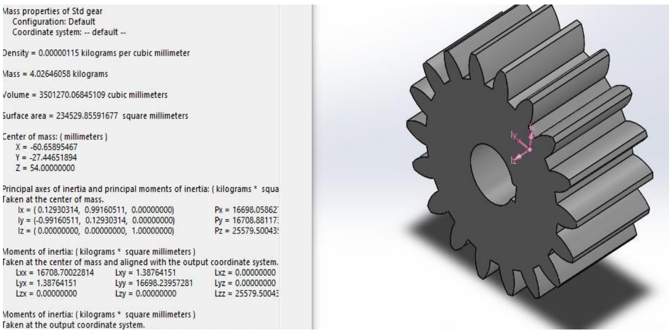


Fig 32: Weight of a normal gear

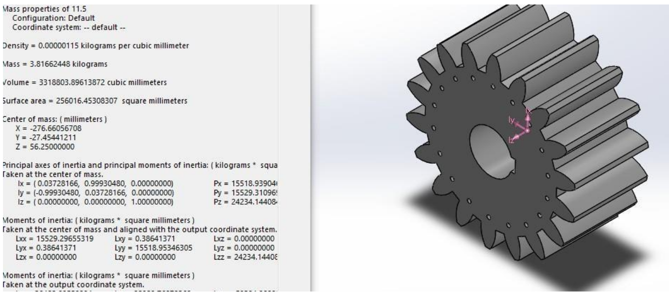


Fig 33: Weight of a perforated gear

The weight of normal gear is 4.026 Kilograms whereas that of perforated gear is 3.816 Kilograms. Hence, the perforated gear offers less weight by 0.21 Kilograms (210 grams).

10 Conclusions

The project was aimed to co-relate the bending stresses and displacements of a spur gear tooth, obtained both analytically as well as by FEM approach. With the help of static and dynamic analysis of spur gear tooth, factors like maximum displacement, maximum induced stress and effect of stress variation with respect to time are determined.

The conclusions which can be drawn from the results obtained are:

- By performing different iterations in terms of hole dimensions as well as hole positions, we have come to a conclusion that an offset of 11.5 mm with a hole diameter of 5 mm has given the least value of stress i.e 1.1798 MPa.
- The maximum stresses for gear teeth occur in the root region of gear tooth with a value of 1.9613 MPa.
- The project was started with carrying out analysis to find stresses at traditional locations. After performing those, it was observed that least stresses were present in the region below dedendum circle. Hence, the different iterations have been performed in the region.
- The weight of normal gear is 4.026 Kilograms whereas that of perforated gear is 3.816 Kilograms. Hence, the perforated gear offers less weight by 0.21 Kilograms (210 grams)

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