Design & Analysis of Helical Gear made of Stainless Steel & Nylon under Different Loading Conditions

Prafulla P. Jadhav
PG Student
Department of Mechanical Engineering
SRES College of Engineering
Kopargaon, Maharashtra, INDIA

Santosh V. Bhaskar
Associate Prof.
Department of Mechanical Engineering
SRES College of Engineering
Kopargaon, Maharashtra, INDIA

Abstract*— Gears are the back bone & one of the most critical components in mechanical power transmission systems. Among the contributors in gear set failure bending and surface strength are identified as one of the contributors which plays major role in it. Thus, to reduce the failure of gear & for optimization of gear design analysis of stresses resulted into major are of interest. In this paper the major focus is on studying stresses and pressure distribution using analytical and finite element analysis for the characteristics of an involute helical gear system. To estimate & identify the various stresses and other important parameters of helical gears, a three-dimensional solid model of set of gears is generated using the Computer Aided Design Software Pro/Engineer, which is currently being known as a strong tool for solid modeling. The analytical investigations are carried out by varying the load on gear. This paper also considers the study of contact stresses induced between two gears. To determine the contact stresses between two mating gears, the analysis is carried out on the two gears. A pair of standard helical gears is used and analysis is performed on by FEA Software Altair HYPERWORK OptiStruct solver using the following two conditions for the same power output (i) gear pair made of nylon; and (ii) other pair made of stainless steel. The results obtained from Altair HYPERWORK OptiStuct, are compared to investigate the possibilities of using Nylon as an alternative material for gears against traditional as steel in power and/or motion transmission sector.

Keywords*— *Computer Aided Design; Finite Element Analysis; Helical Gears, Hyper Mesh, HYPERWORKS, OptiStruct Stainless Steel

I. INTRODUCTION

Jitendra Singh Chauhan et al. (2016) [1], they have established the co-relation between AGMA equation and FEA Software Ansys for calculating bending stresses and contact stresses.

A.Y. Gidado et al. (2013) [2], has done co relation between statistical calculation of bending stress by Lewis formula against the FEA by Ansys FEA software. Also mentioned that for the failure of gear teeth, root bending stress plays major role in fatigue fracture while contact stresses are responsible for pitting failure of contact surface.

F.W. Brown (2011) [3], have done research to differentiate the bending and contact stresses between asymmetric and symmetric involute gear tooth. Both these gears were representative of helicopter main gear drives. They carried out test on both the gear system and observed that for both the asymmetric tooth form and optimized tooth form has higher bending strength when compared to baseline designs.

Pushpendra Kumar Mishra (2003) [4], have identified that bending strength and surface strength are the key factors which are involved in failure of the gear tooth of a gear sets. Also they identified & studied that various researchers have used different techniques for the optimization & calculation of stresses which are contributing in gear design.

Ashish N. Taywade, Dr. V. G. Arajpure (2014) [5], have studied and shown that Nylon66 is better option against the issues due to acetyl gear because of superior properties of Nylon.

Cheng, Y. et al (2002) [6], have investigated using Finite Element analysis contact and the bending stresses of helical gear set with localized bearing contact.

JianFeng L.et al (1999) [7], have established FEA models for analyzing of different parameters of helical gears such as helical gears for internal and external hobbing and slotting. Also they have presented the results of load distribution along the contact line, contact stresses. Also shown the co relation between the trend of gear tooth deformation against gears which are tested by using the dynamic speckle photography method

Tsay, C.B., and Fong, Z.H.(1991)[8], used Tooth contact analysis technique & Finite Element method used for contact & stress analysis of gears. They have investigated the stress analysis model for new pair of gearing.

Vijayarangan, S.,and Ganesan (1993)[9], has investigated the safe use of composite materials for helical provided the face width of the gear to be increased appropriately.

Huston, R.L., Mavriplis et al. (1992) [10], discussed the uses of gears with extension of non-tooth forms Rao, C.M., and Muthuveerappan (1993) [11], has studied the effect of varying face width and helix angle on the root stresses of helical gears. They investigated effect on different position of contact line for helical gears of the helix angle and face width.

Lu, J. et al. (1993) [12], have done the finite element models for the pinion and gear respectively. The stress analyses for aligned and misaligned have been compared with those obtained by other approaches Elastic deformation of teeth and the stress analysis of the double circular are helical gears are accomplished by using finite element method.

Jianfeng L.et al(1999)[13], they have established the 3D solid model of gear teeth based on the analysis of the profile formulation principle of internal and external mesh gears of hobbing and slotting.
Zhang, J.J. et al (1999)[14], used an approach for the analysis of teeth contact and load distribution of helical gears with crossed axis. The approach was based on a tooth contact model that accommodates the influence of tooth profile modifications, gear manufacturing errors and tooth surface deformation on gear mesh quality
Vera, N.S. et al. (2003)[15], have described enabling the precise calculation of gear pair pitting resistance by using the shape of function which defines the change of contact stresses on tooth flanks along the path of contact for a tooth pair.
Anders Flodin (2000) [16], carried out an investigation of the real wear development on a set of gear wheels was carried out where plastic replicas of the flanks were made to avoid dismounting the gear wheels
Based on the literature review of past work it was observed that very less work has been done on analytical study of helical gears specially when the two gears are in mesh condition. Also, very few reports are available on studying the stress generation analysis of gears made of different materials. The following are the identified research Gaps:

i) Very less reports on studying stress analysis in helical gears in mesh and finding the effects of between the teeth.
ii) Very less reports on identifying the different materials of gears using the software tools.
iii) Very less reports in studying the effects of loading condition on stress generation of gears.
iv) No reports have been found on comparing the stress generation behavior of helical gear made of stainless steel & nylon under different loading conditions.
v) To study & conclude the usefulness and better of use of nylon as an alternative gear material for transmitting power and/or motion in different loading conditions.

Based on above gap analysis of literature review, the objective of work on the gears has been identified as-
(a) To study of the behavior of helical gear made of stainless steel & nylon under different loading conditions.
(b) To understand the characteristics of an involute helical gear system.
(c) To model the gear pair in Computer Aided Design Software PRO/Engineer and meshing to be done in HYPERMESH software.
(d) To analyze the gear pair in Altair OptiStruct solver
(e) To study pattern of the contact stresses between two gears with same no. of teeth
(f) To identify the Nylon as alternative material for Gears made of stainless steel in power transmission

II. METHODOLOGY

a. Modelling the gear as per specifications in PRO/Engineer software
b. Importing the developed design on HYPERMESH software without affecting its original geometry.
c. Fine grain meshing of the developed design to perform various further analyses over it.
d. Providing the suitable and correct material to the developed gear design using the standard library of materials.
e. Perform comparative study of the effect of varying load on the various stresses, strains on the helical gear model.
f. Analyse and Compare the various stresses, strain and safety factors of the model of helical gears made of stainless steel and Nylon using FEM tool Altair OptiStruct solver

III. DESIGN OF HELICAL GEARS

A standard gear set of helical type has been selected with following characteristics & parameters as shown in Table II for the power and RPM mentioned in Table I

TABLE I POWER AND RPM

Case	Power kW	RPM n_p
1	2.5	6500
2	4	3000
3	1.5	5000

As for the automotive particular applications following parameters are knowing
a=center to center distance=52.624mm
z_g=no. of teeth on gear=20
z_p=no. of teeth on pinion=20
ψ=helix angle=18.7°

So normal module of the gear can be calculated as [17],

$$m_n = \frac{a\ 2\cos\psi}{z_g + z_p}$$

$$m_n = \frac{a\ 2\cos\psi}{z_g + z_p}$$

$$m_n = \frac{52.624\ X\ 2\cos 18.7°}{20 + 20}$$

$$m_n = 2.6mm$$

So nearest module for the gear selected is as 2.5mm

$$m_n = 2.5mm$$

After getting module, all other parameters of the gear have been calculated and tabulated as shown in Table II

TABLE II. SPECIFICATION OF GEAR USED FOR ANALYSIS

Parameter	Gear 1	Gear 2
No. Teeth	20	20
Normal Pitch (d_P) (mm)	10.16	10.16
Transverse d_P	9.65	9.65
Circular Pitch (mm)	7.8540	7.8540
Module (mm)	2.5000	2.5000
Normal Pressure Angle	20°	20°
Transverse Pressure Angle	21.020°	21.020°
Helix Angle	18.7°	18.7°
Pitch Diameter	52.787 mm	52.787 mm
Outside Diameter	58.287 mm	58.287 mm
Centre Distance	52.624	

Based on the above parameters with the help of Pro/Engineer Software solid model of Gear has been generated which is shown in Figure 1.

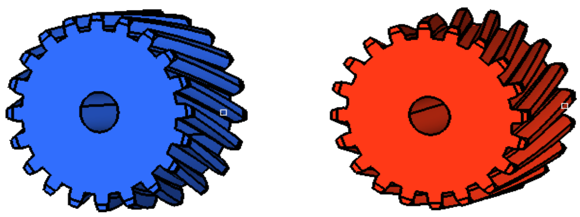

Fig 1. Gear Design

A. *Material Properties*

For the analysis of gears Stainless Steel & Nylon 66 are used. The properties of the both material are tabulated as in Table III

TABLE III MATERIAL PROPERTIES OF STAINLESS STEEL & NYLON 66

Parameter	Stainless Steel	Nylon66
Density	7860kg/m^3	1400kg/m^3
Poissons Ratio	0.2	0.39
Modulus of Elasticity	2X10^5 Mpa	3540Mpa
Tensile Strength	1310Mpa	83Mpa
Compressive Strength	1120MPa	86Mpa

B. *Load Calculations*

CASE I

Effective load on tooth can be calculated based on M_t which is transmitted torque and can be given as[17]

$$M_t = \frac{60 \times 10^6 \times kW}{2\pi n}$$

=3574.67 N-mm

$$P_d = \frac{e n_p z_p \, b r_1 r_2}{2530\sqrt{r_1{}^2 + r_2{}^2}}$$

Where
P_d= dynamic load N
n_p=rpm of pinion
z_p= no. of teeth of pinion
z_g=no. of teeth of gear
b= face width, mm
r1=radial pitch of gear 1, mm
r2= redial pitch of gear 2, mm
ψ =helix angle
α_n =normal pressure angle
e=total error
Cs=service factor

After calculating total error, following is the dynamic load on the gear 2,[17]

$$P_d = \frac{21.43 \times 6500 \times 20 \times 26.39 \times 26.39}{2530\sqrt{26.39^2 + 26.39^2}}$$

P_d =513.94N

$$P_{eff} = C_s P_t + P_d \cos\alpha_n \cos\psi$$

P_{eff} =416.2N

Similarly, calculations for 2nd case and 3rd case has been performed & results are tabulated in Table V

TABLE V –FORCE & TORQUE DETAILS

Case	Power kW	RPM	Transmitted Torque (N-mm)	Dynamic Load at gear tooth (N)	Effective force on Gear Tooth P_{eff} (N)
1	2.5	6500	3574.67	513.98	416.2
2	4	3000	12738.85	237.2	819.7
3	1.5	5000	2866.242	395.36	322.4

Table V shows Force and Torque details which are boundary conditions for this analysis. P_{eff} is the load which is going to be applied on gear tooth

IV. ANALYSIS OF GEARS

A. *Meshing of Gears*

Meshing of both the gears done using HYPERMESH software with element size of 10. The meshed gears are shown as below in figure 2

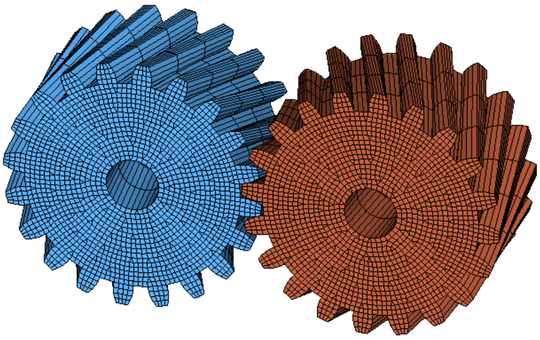

Figure 2-Meshed Gears

B. Application of Loads on Gear

Based on calculations, P_{eff} load is applied as shown in below Figure 3 on tooth of Gear 2

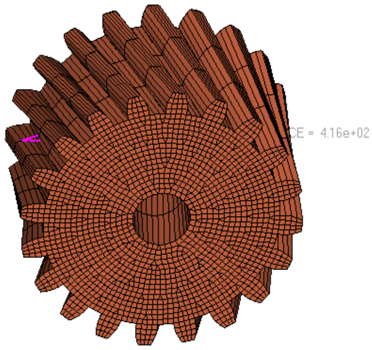

Figure 3- Application of Load on Gear

V. RESULT & DISCUSSION

A. Results of Case 1

For Stainless Steel

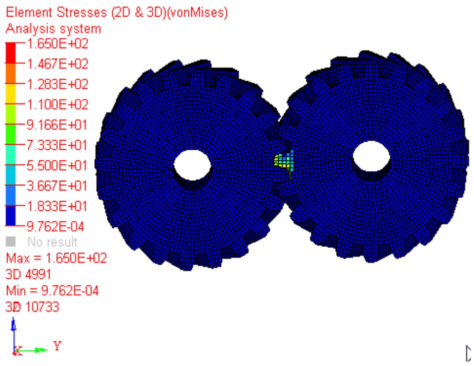

Figure 4- Case I-2.5kW- Von Mises stress on Stainless Steel Gear

As shown in figure 4, Von Mises stress developed on gear is 166Mpa for power of 2.5kW

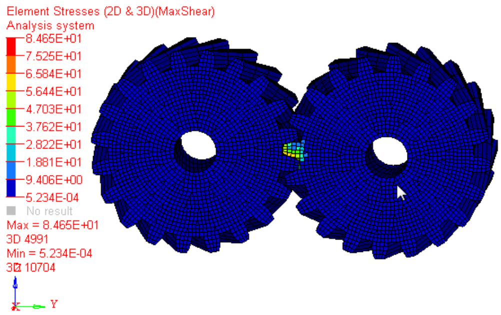

Figure 5 Case I-2.5kW -Max. Shear stress on Stainless Steel Gear

As shown in figure 5, Max shear stress developed on gear is 84.65Mpa for power of 2.5kW

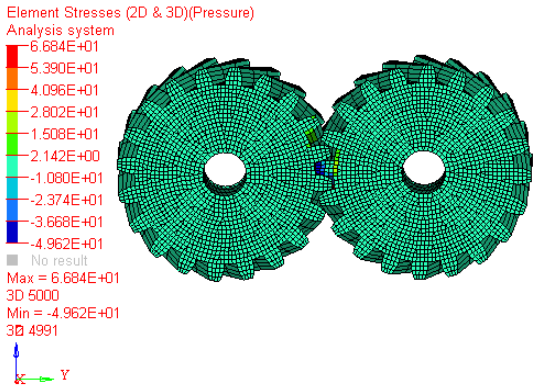

Figure 6 Case I-2.5kW- Pressure Distribution on Stainless Steel Gear

As shown in figure 6, pressure developed on gear is 66.8Mpa for power of 2.5kW

For Nylon 66

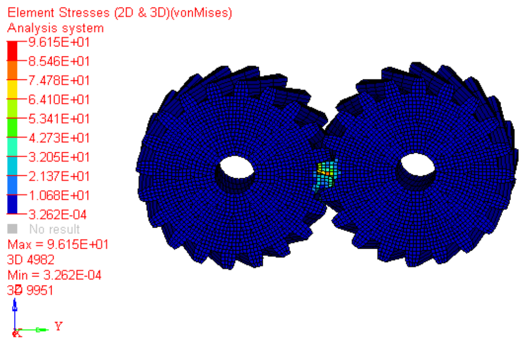

Figure 7 Case I-2.5kW- Von Mises Stress on Nylon 66 Gear

As shown in figure 7, von mises stress on gear is 96.1Mpa for power of 2.5kW

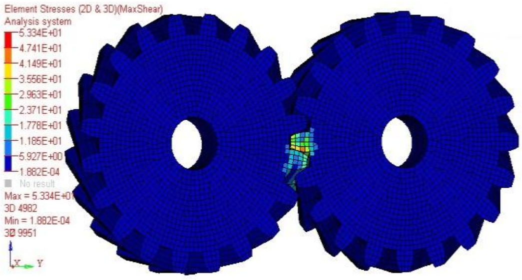

Figure 8 Case I-2.5kW- Max. Shear stress on Nylon66 Gear

As shown in figure 8, Max shear stress developed on gear is 53.34Mpa for power of 2.5kW

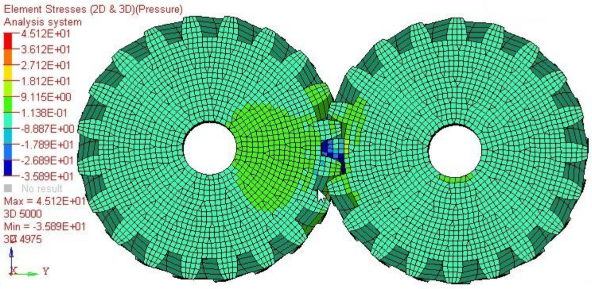

Figure 9 Case I-2.5kW- Max. Shear stress on Nylon66 Gear

As shown in figure 8, pressure distribution on gear is 45.12 Mpa for power of 2.5kW

B. Results of Case II

For Stainless Steel

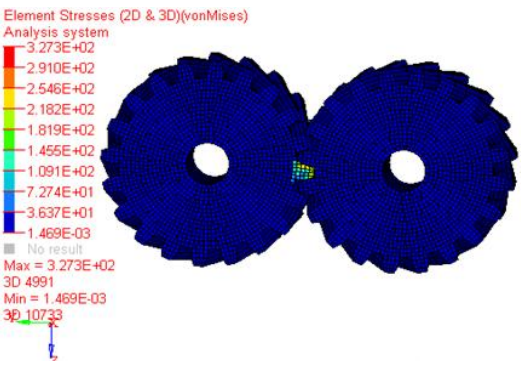

Figure 10 Case II-4kW- Von Mises stress on Stainless Steel Gear

As shown in figure 10, von mises stress on gear is 327.2 Mpa for power of 4kW

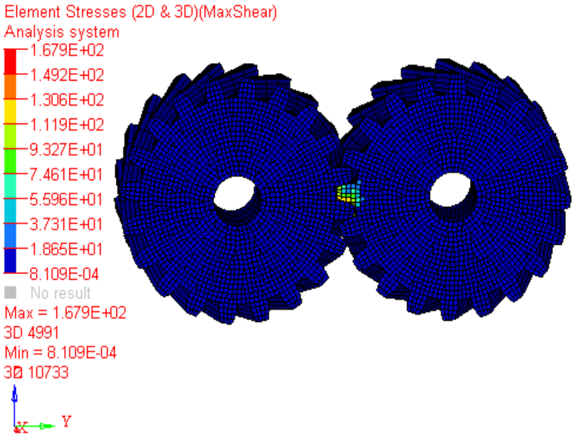

Figure 11 Case II-4kW- Max. Shear stress on Stainless Steel Gear

As shown in figure 11, max. shear stress on gear is 167.9 Mpa for power of 4kW

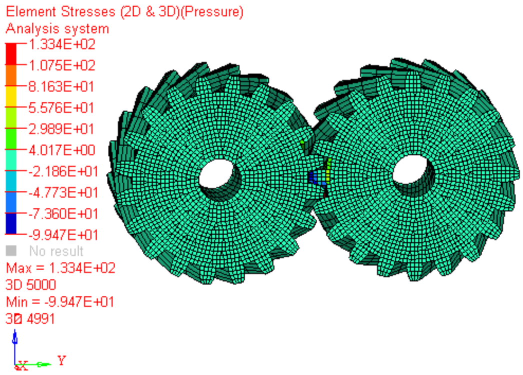

Figure 12 Case II-4kW- Max. Shear stress on Stainless Steel Gear

As shown in figure 12, pressure distribution on gear is 133.4 Mpa for power of 4kW

For Nylon 66

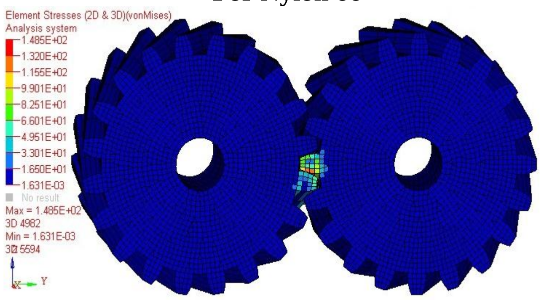

Figure 13 Case II-4kW- Von Misses Stress on Nylon 66 Gear

As shown in figure 13, von misses stress on gear is 148.5 Mpa for power of 4kW

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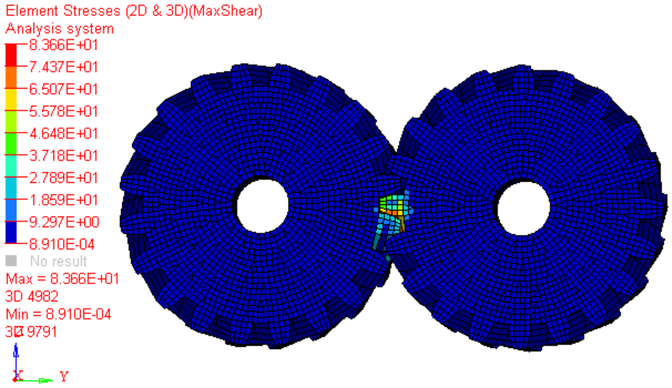

Figure 14 Case II-4kW- Max. Shear stress on Nylon 66

As shown in figure 14, max. shear stress on gear is 83.66 Mpa for power of 4kW

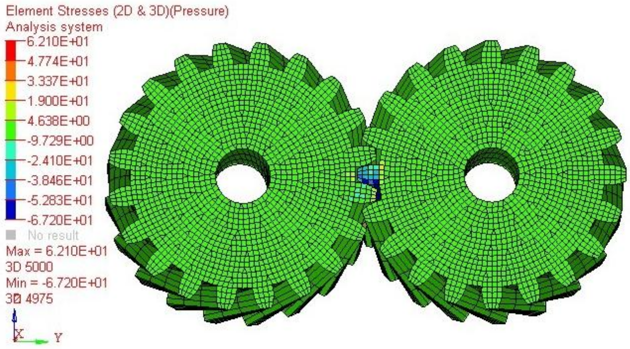

Figure 15 Case II-4kW-pressure distribution on Nylon 66

As shown in figure 15, pressure distribution on gear is 62.1 Mpa for power of 4kW

C. *Results of Case III*

For Stainless Steel

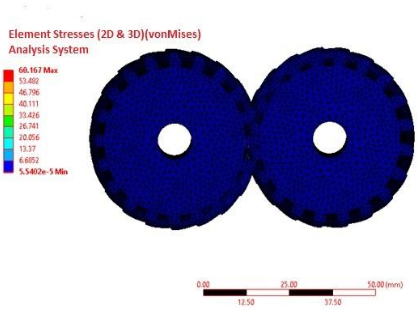

Figure 16 Case III-1.5kW- Von Misses Stress on Stainless Steel

As shown in figure 16, von misses stress on gear is 60.17 Mpa for power of 1.5kW

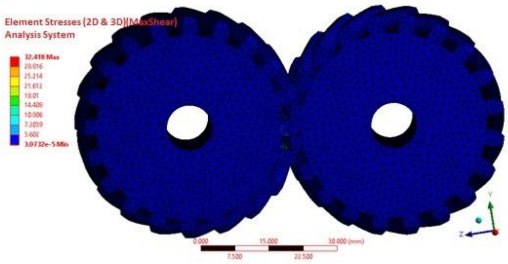

Figure 17 Case III-1.5kW- Max. Shear Stress on Stainless Steel

As shown in figure 17, Max. Shear stress on gear is 32.418 Mpa for power of 1.5kW

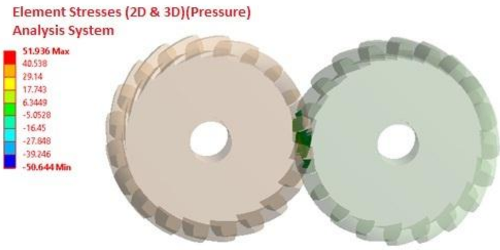

Figure 18 Case III-1.5kW- Pressure distribution on Stainless Steel

As shown in figure 18, Pressure distribution on gear is 51.9 Mpa for power of 1.5kW

For Nylon 66

Figure 19 Case III-1.5kW- Von Misses Stress on Nylon 66

As shown in figure 19, Von Misses Stress on gear is 53.891 Mpa for power of 1.5kW

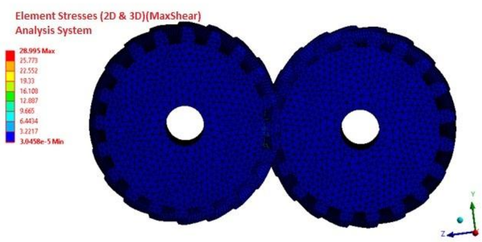

Figure 20 Case III-1.5kW- Max. Shear Stress on Nylon 66

As shown in figure 20, Max. Shear Stress on gear is 28.99 Mpa for power of 1.5kW

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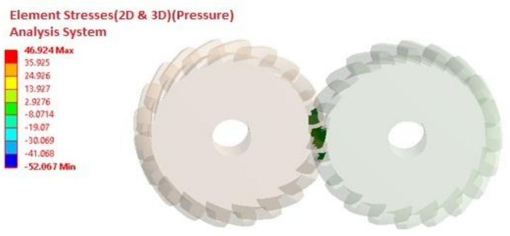

Figure 21 Case III-1.5kW- Pressure Distribution on Nylon 66

As shown in figure 21, Pressure distribution on gear is 46.92 Mpa for power of 1.5kW

VI. RESULT AND DISCUSSIONS

TABLE VI- VON MISSES, SHEAR STRESS & PRESSURE DISTRIBUTION DETAILS

Case	Stainless Steel			Nylon 66		
	Von Mises Stress MPa	Shear Stress MPa	Pressure Distribu t-ion MPa	Von Mises Stress MPa	Shear Stress MPa	Pressure Distribu t-ion MPa
2.5 kW	165	84.65	66.84	96.15	53.34	45.12
4.0 kW	327. 3	167.9	133.4	148.5	83.66	62.1
1.5k W	60.1 67	32.418	51.936	53.891	28.99	46.924

For both case I, case II, case III results are tabulated as above in Table 4 of Von Misse Stress, Max. Shear Stress, Pressure Distribution for both materials analyzed viz Stainless Steel and Nylon66. From the results it observed that stress induced on Nylon 66 gear tooth are lesser than that of induced on Stainless Steel gear. As mentioned in Introduction of this paper due to increase in area of applications of gears these results will help to use Nylon as alterative material.

VII. CONCLUSION

After analysis of Helical gear made of Stainless Steel and Nylon 66 it has been clear from all the above loading cases that stress on Nylon 66 gears are lesser than stainless steel for the same loading. It has been also observed that for first two load cases loading conditions the stress values on nylon 66 gear are more than that of tensile strength of the material. But for third case Nylon 66 has lower stress values against yield limit so for that load case Nylon 66 can be considerd as alternative material for helical gear against Stainless Steel. In terrm of other advantages,Nylon 66 also has properties like low cost, less weight, ease of manufacturing & self-lubrication. This analysis helps to check Nylon 66 as alternative material for most of the automobile applications.

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