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Design and analysis of different polymer gears in feed drive mechanism for all geared Lathe machine

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Abstract

Gears are the most vital parts in industrial machines as a power transmission component. Nowadays polymer gears are considered as a correct replacements to a conventional metallic gears. In all geared lathe machine for better performance metallic gears in drive mechanism is replaced by three different polymer materials viz. Nylatron (NSM), Nylon 66, PTFE (Polytetrafluroethelene). Gear design parameter like module, face width, and pitch circle and etc. considered to optimize the design parameter in geared model to make polymer as feasible gear model. An attempt is made in this paper to analyze the reconstruction of drive mechanism in all geared lathe machine considering the contact stresses, bending stresses, equivalent(Von-Mises) stress, equivalent elastic strain, total deformation. Static analysis of a 3D model has been performed using ANSYS 15R. It is concluded that use of high strength engineering polymer in the industrial machine like lathe machine will make the machine more efficient, smoother for operation and noiseless as compared to conventionally used metallic gears.

Keywords:- Nylatron(NSM), Nylon 66, PTFE, Spur gear, Lathe feed drive mechanism, Ansys

1. Introduction

Gears are a critical component in the rotating machinery industry. Gears are the most common mean of transmitting power in mechanical engineering. Nowadays it is need of time to use non-metallic gears due to their special properties as compared to metallic gears. Polymer material like Nylatron, Nylon, Acrylic, PEEK, POM.PTFE etc. . are used for gears due to their reducing weight and noiseless operation. These polymer material have high strength, excellent wear resistance, self-lubricant, good mechanical property.

Due to such properties one attempt is made to replace metallic spur gear train of lathe machine feed drive mechanism. Study and accurate analysis of the gear system is valuable to get the effective gear transmission. There are many types of stresses occurs in loaded and rotating gear teeth. So one should take account all the possibilities, so that the gears are proportional to keep all the stresses with in design limit. Each gear tooth should be considered as a cantilever beam, while transmitting the load subjected to bending. The main aim is to determine the contact stresses, bending stresses and total deformation. Analysis of spur gear by Ansys software, for this spur gear train is prepared in catia software.

2. Literature Review

Ashwin Chopane et al. [1] calculated the beam strength, wear strength, effective load for designing the arrangement of rack and pinion for racing car. He prepared a 3D model of the steering system and accumulated it in software like Creo, and ANSYS is used for performance analysis of tetrahedral mesh element to gear better precision results. It concluded that with polymer gears arrangements in this steering system with supra car seems very beneficial as it gives reduced weight, vibration, and noise for the system.

Ashish Taywade et al. [2] made design analysis for both metal and polymer helical gears. It showed that in Nylon 66 helical gear stress induced was less as compared to metallic gear and also strength to weight ratio is higher in case of Nylon 66 gear. Due to polymer material, it provides convenience for manufacturing. It gives economic benefits, less weight, less noise and vibration, self-lubricating, and many more.

Ashish Taywade et al. [3] designed and developed Nylon 66 plastic helical gear for the replacement of metallic gear for automobile application. Due to superior properties of Nylon 66, it met to extreme performance challenges. For design and development tooth load is calculated first by using Lewis equation and with the help of Buckingham's equation dynamic tooth load is calculated for the gear and then Von-Mises stress is found.

R. Yakut, H. Düzcükoğlu*, M.T. Demirci et al [4] In this study, load carrying capacity and occurring damages of gears which are made of PC/ABS blends were investigated. PC is hard material and ABS is soft material. The usage of materials limits these drawbacks. However PC and ABS polymers combine each other, the PC/ABS blends have suitable mechanical properties for gear applications in the industrial areas. In this study, usability of PC/ABS composite plastic materials as spur gear was investigated. PC/ABS gears were tested by applying three different loading at two different numbers of revolutions on the FZG experiment set.

V. Siva Prasad et al.[5] This paper describes design and Analysis of spur gear and it is proposed to substitute the metallic gears of sugarcane juice machine with polymer gears to reduce the weight and noise. A virtual model of spur gear was created in PRO-E, Model is imported in ANSYS 10.0 for analysis by applying normal load condition. The main purpose of this paper to analysis the different polymer gears namely nylon, polycarbonate and their viability checked with counterpart metallic gear like as cast iron. Concluding the study using the FEA methodology, it can be proved that the composite gears, if well designed and analyzed, will give the useful properties like as a low cost, noise, Weight, vibration and perform its operation similar to the metallic gears. Based on the static analysis Nylon gear are suitable for the application of sugarcane juice machine under limited load condition in comparison with cast iron spur gears.

Vivek Karaveer et al.[6] This paper presents the stress analysis of mating teeth of the spur gear to find maximum contact stress in the gear tooth. The results obtained from finite Element analysis are compared with theoretical Hertz equation values. The spur gear are modeled and assembled in ANSYS DESIGN MODELER and stress analysis of Spur gear tooth is done by the ANSYS 14.5 software. It was found that the results from both Hertz equation and Finite Element Analysis are comparable. From the deformation pattern of steel and grey cast iron, it could be concluded that difference between the maximum values of steel and grey CI gear deformation is very less.

Mahebub Vohra et al.[7] In this paper, Metallic material Cast iron and Non Metallic material Nylon are investigated. The stress analysis of the lathe machine headstock gear box are analyzed by finite element analysis. Analytical bending stress is calculate by two formula Lewis formula and AGMA formula. Analytical results is compared with the finite element method result for validation. Concluding the study, we observed that finite element method software ANSYS have values of stress distribution were in good agreement with the theoretical results. Besides non-metallic material can be used instead of metallic material because non-metallic material provide extra benefits like as less cost, self-lubricating, low noise, low vibration and easy manufacturing.

4. Specification of Problem

In this project work it is proposed to substitute the metallic gears of feed drive mechanism in lathe machine with

plastic gears to reduce the total feed drive system's weight and also to mi

types of plastic materials were used i.e. Nylatron(NSM), Nylon 66, PTFE(Polytetrafluroethelene) and their viability is checked with their counterpart metallic gear(cast iron). Considering the static analysis of gears of these polymer materials compared to each other. A virtual model of spur gear was created in CATIA V5 and then this model is imported in Ansys 15R for analysis purpose and then normal load conditions were applied. On the basis of deflection and stressed from analysis all spur gear of these plastic material were compared.

5. Feed drive Gear system design

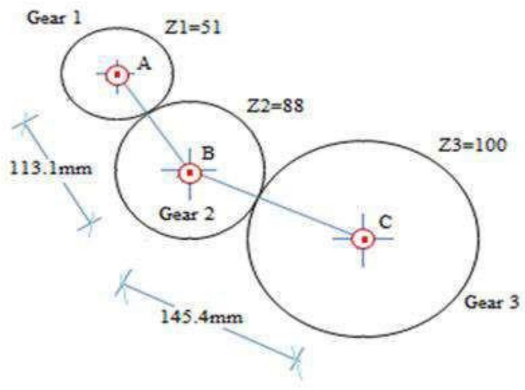


Fig. 1. Existing steel gear train configuration

Fig. 2. Actual Image of Feed drive mechanism in Lathe machine

There are three gears i.e. Gear 1, Gear 2, Gear 3 Mounted on shaft A, B and C. The number of teeth on gears are 51,88 and 100 respectively.

Table 1. Input Parameters of existing gear train

Parameter	Value
Gear 1	51 teeth
Gear 2	88 teeth
Gear 3	100 teeth
Center distance 1-2	113.1 mm
Center distance 2-3	145.4 mm
2 HP motor connected to Gear 1	1440 rpm

Types of gear:

Existing steel gear in feed drive mechanism is of spur gear type so for plastic gears same type of spur gear is selected to avoid axial load as well as for easy machining and simplicity. A 20^0 full depth involute profile system was selected due to there merits

Interference reduction Reduces risk of undercutting

Due to this involute profile system, Gear tooth becomes somewhat wider at the root place and hence strong tooth and load carrying capacity increases and give good length of contact.

5.1 Plastic gearing

5.2 Material Properties

Three different materials are chosen. Table no. 2 shows the properties of all material.

Table 2. Properties of materials

Properties	**Steel**	**CI**	**Nylon 66**	**PTFE**	**Nylatron NSM**
Hardness	106.8	105	118-120	52-56	80
Tensile Strength (N/mm^2)	67.70	65.5 -420	85	13-333	75.84
Flexural Yield Strength (MPa)	40	421-641	145-310	14.0-27.6	110
Elongation At Brake (%)	13	1 to 15	5-640	300-550	20
Melting Point (Celsius)	1470	1540° C	260	317-337	420
Thermal Conductivity (w/m-k)	46	42	0.53	0.250	0.29
Tensile Modulus (MPa)	240	170	5500	550	2826
Density (g/cm^3)	7.8	7.7	1.4	2.2	1.14
Yield Stress (N/mm^2)	285	159-221	82.8	42	80
	0.3	0.240-0.330	0.37 to 0.40	0.42	0.40

Nomenclature:

Z	=	Number of teeth	d	=	Pitch circle diameter
T_R	=	Transmission Ratio	v	=	Pitch line velocity
V_R	=	Velocity Ratio	Peff	=	Effective load on gear tooth
C_V	=	Velocity Factor	Cs	=	Service factor
m	=	Module	Pt	=	Tangential load on gear tooth
Pd	=	Dynamic load	C	=	Deformation factor
Ø	=	Pressure angle	σb	=	Permissible bending stress
µ	=	Poission Ratio	Y	=	Lewis form factor
E	=	Modulus of Elasticity	Tr	=	Torque transmitted
Sb	=	Beam strength	fos	=	Factor of safety

6. Design of gear train for Nylatron NSM as a material for all the gears in lieu of steel. Properties of Nylatron NSM:-

Ultimate tensile strength (S_{ut}) = 80 Mpa Yield tensile strength (S_y) = 80 Mpa Modulus of elasticity (E) = 3150 Mpa

Let the gear tooth system be **20^0 full depth involute tooth system** for the **Nylatron NSM.**

We know,

$$Transmission\ Ratio = TR = \frac{angular\ velocity\ of\ first\ driving\ gear}{angular\ velocity\ of\ last\ driven\ gear}$$

i.e,

$$TR = \frac{n_1}{n_3} = \frac{z_3}{z_1} = \frac{100}{51} = 1.96$$

Now, for two stage gear box, the velocity ratio (VR) is given by;

$$VR = \sqrt{\mathrm{TR}} = \sqrt{1.96} = 1.4$$

$\Rightarrow$

$$VR = \frac{n_1}{n_2} = \frac{z_2}{z_1} = 1.4$$

$$\frac{1440}{n_2} = \frac{88}{51}$$

$$n_2 = 834.54 \text{ rpm}$$

Similarly,

$$VR = \frac{n_2}{n_3} = \frac{z_3}{z_2} = 1.4$$

$$\Rightarrow \quad n_3 = 734.4 \text{ rpm}$$

On comparing the torque transmitted by shaft A, B, & C, we found that more torque is transmitted by shaft B & C, therefore Gear2 and Gear3 are to be designed.

Now the center distance between Gear 2 and Gear 3 must be 145.4 mm.

i.e. $$\frac{d_2}{2} + \frac{d_3}{2} = 145.4 \text{ mm}$$

$$\Rightarrow \quad d_2 + d_3 = 290.8 \text{ mm} \quad \ldots\ldots\ldots\ldots\ldots\text{(I)}$$

Also we know,

$$VR = \frac{n_2}{n_3} = \frac{z_3}{z_2} = \frac{d_3}{d_2} = \frac{100}{88} = 1.136$$

$$\Rightarrow \frac{d_3}{d_2} = 1.136 \;\ldots\ldots\ldots\ldots\; \text{(II)}$$

Now solving equ. (I) & (II), we get,

$$d_2 = 136.14 \text{ mm} \quad \& \quad d_3 = 154.65 \text{ mm}$$

Similarly we have, $$\frac{d_1}{2} + \frac{d_2}{2} = 113.1 \;\ldots\ldots\ldots\; \text{(III)}$$

Solving, we get, $$d_1 = 90.06 \text{ mm}$$

Now Gear2 and Gear3 are made up of same material, therefore Gear2 will be weaker and will be the criteria of design.

Pitch line velocity of Gear2 (v_2),

$$v_2 = \frac{\pi \times d_2 \times n_2}{60} = \frac{\pi \times 0.13614 \times 834.54}{60}$$

$$v_2 = 5.94 \text{ m/s}$$

Velocity factor (C_v),

$$Cv = \frac{3}{3 + v_2} = \frac{3}{3 + 5.94}$$

$$Cv = 0.335$$

Now, let's assume that the number of teeth on pinion2 will be in the range of 20 to 30 teeth.

Therefore, the form factor (Y) will vary from 0.32 to 0.358.

Let's take the intermediate value of form factor (Y),

Y=0.34

Now let's calculate the Beam Strength (S_b),

$$Sb = \mathrm{m} \times \mathrm{b} \times \sigma\mathrm{b} \times \mathrm{Y} = \mathrm{m} \times \mathrm{b} \times \sigma\mathrm{b} \times \mathrm{Y}$$

Where,

m = module of gear

b= face width of gear = 10m

ut/3
σb = S

$$\because \quad Sb = \text{m} \times 10\text{m} \times 80/3 \times 0.34$$

$$Sb = 90.66\, m^2 \ldots\ldots\ldots\ldots\ldots\text{eq. (A)}$$

We know torque transmitted by Gear2 (Tr)

$$\text{Tr} = \frac{\text{P} \times 60}{2 \times \pi \times n_2} = \frac{1491 \times 60}{2 \times \pi \times 834.54}$$

$$\text{Tr} = 17.06\, N - m$$

$$\text{Tr} = 17060.89 \text{ N} - \text{mm}$$

$$Pt = \frac{2 \times \text{Tr}}{d_2} = \frac{2 \times 17060.89}{136.14} = 250.63 \text{ N}$$

$$Peff = \frac{\text{Cs}}{Cv} \times Pt = \frac{1.25}{0.335} \times 250.63 = 935.186 \text{ N} \;\ldots\ldots\ldots\ldots\text{eq. (B)}$$

Let's take factor of safety (fos) "1.5" for design

We know,

$$\text{fos} = \frac{Sb}{Peff}$$

$$\Rightarrow \quad Peff \times \text{fos} = Sb \ldots\ldots\ldots\ldots\ldots\ldots\ldots\text{ eq. (C)}$$

Now putting the values from eq. A & B in eq. C, we get

$$935.186 \times 1.5 = 90.66\, m^2$$

$$\Rightarrow \quad \mathbf{m = 3.93} \text{ mm}$$

Now taking the choice-1 preferred value for module, i.e.

$$\mathbf{m = 4} \textbf{ mm}$$

Now the Gear dimensions will be as follows,

m = **4** mm

b = 10× m = **40** mm

$$z_1 = \frac{d_1}{m} = \frac{90.06}{4} = 22.515 \approx \mathbf{22}$$

$$= \frac{}{m} = \frac{136.14}{4} = 34.035 \approx \mathbf{34}$$

d_2
z_2

d_3
z_3

$$= \frac{}{m} = \frac{154.65}{4} = 38.66 \approx \mathbf{39}$$

Now let's check the design

We have,

z1=22, z2 = 34, & z3 = 39

d1= 88 mm, d2= 136 mm & d3 = 156 mm b= 40 mm

Let's take Grade

Now we have,

-8 finish for manufacture

$$Peff = (Cs \times Pt + Pd)$$

Where,

Cs = Service Factor = 1.25

Pt = 250.63 N

$$Pd = \frac{21.v.[C.e.b \quad Pt]}{+\sqrt{[C.e.b + Pt]}} +$$
$$21.v$$

Where,

C = deformation factor

e = sum of error for gear manufacture b = face width

v = pitch line velocity

Now the deformation factor **C** is given by,

$$\mathbf{C} = \frac{K}{\frac{1}{Ep}+\frac{1}{Eg}} = \frac{0.111}{\frac{1}{3150}+\frac{1}{3150}} = 174.825 \quad \dots\dots\dots \text{(a)}$$

Where, K= constant depending upon the form of tooth = 0.111 (for 20^0 full depth system)

Ep & Eg = modulus of elasticity for pinion & gear material respectively = Ep = Eg = 3150 N/ mm^2

For grade 8, we have

$$e = ep + eg$$

$$ep = 16 + 1.25 \ [m + 0.25\sqrt{d_2}]$$

$$ep = 16 + 1.25\ [4 + 0.25\sqrt{136}]$$

$$ep = 24.64\ \mu m$$

$$eg = 16 + 1.25\ [m + 0.25\sqrt{d_3}]$$

$$eg = 16 + 1.25\ [4 + 0.25\sqrt{156}]$$

$$eg = 24.90\ \mu m$$

$$e = ep + eg$$

$$e = 24.64 + 24.90$$

$$e = 49.54\ \mu m$$

$\times$ mm …….(b)

Now, $n_1 = 1440$

We have,

e = 49.54 10^{-3}

$$VR = \frac{n_1}{n_2} = \frac{z_2}{z_1}$$

$$\frac{1440}{n_2} = \frac{34}{22}$$

$$n_2 = 935.06\ rpm$$

Similarly

$$n_3 = 820.22\ rpm$$

Therefore,

$$v_2 = \frac{\pi \times d_2 \times n_2}{60} = \frac{\pi \times 0.13614 \times 935.06}{60}$$

$$v_2 = 6.65\ m/s \qquad …….(c)$$

$$Pd = \frac{21.v.[C.e.b \quad Pt]}{\sqrt{[C.e.b + Pt]}} +$$

$$21.v +$$

Now putting values from (a), (b) & (c) in above, we have

$$Pd = \frac{21\times 6.65\,[174.825\times 0.049543\times 40+250.63]}{21\times 6.65+\sqrt{[174.825\times 0.049543\times 40+250.63]}} = 508.18\text{ N}$$

Now,

$$Peff = (Cs \times Pt + Pd)$$

$$Peff = (1.25 \times 250.63 + 508.18) \quad = 821.47\text{ N}$$

Now,

$$Sb = m \times b \times \sigma b \times Y = 4 \times 40 \times 80/3 \times 0.370$$ **= 1578.66 N**

………………(Y=0.370 for z=34)

$$fos = \frac{Sb}{Peff} = \frac{1578.66}{821.47}$$

fos = 1.92 Therefore design is satisfactory

All the three materials, i.e. Nylatron NSM, Nylon 66 & PTFE will have the same dimensions Table 3. Gear train specification for polymer gears

Gear Dimension	Module (m)	Pressure angle (α)	Face width b	No. of teeth	Pitch circle dia	Addendum circle dia (da)	Dedendum circle dia (df)	Clearance circle	Base circle dia
Gear 1	4 mm	20^0	40	22	88 mm	96 mm	78 mm	80 mm	82.69 mm
Gear 2	4 mm	20^0	40 mm	34	136 mm	144 mm	126 mm	128 mm	127.79 mm
Gear 3	4 mm	20^0	40 mm	39	156 mm	164 mm	146 mm	148 mm	146.59 mm

7. CAD PROTOTYPING

As per the dimensions obtained by precise calculations, 3D model of gear and gear assembly were modeled and assembled in CATIA V5. Shaft hole of gears and keyway is as per the available set as shown in figure.

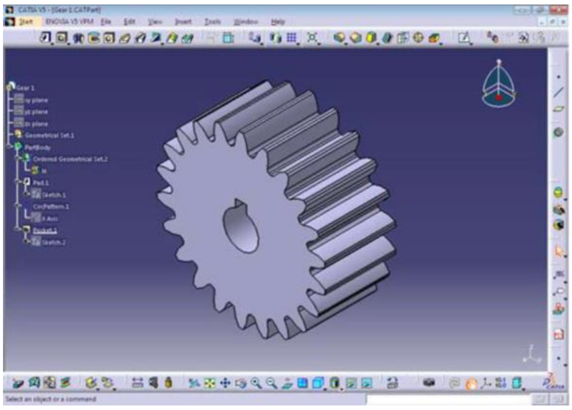


Fig. 3. Gear 1

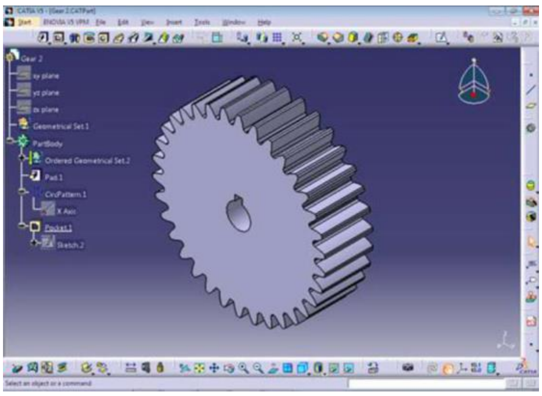


Fig.4. Gear 2

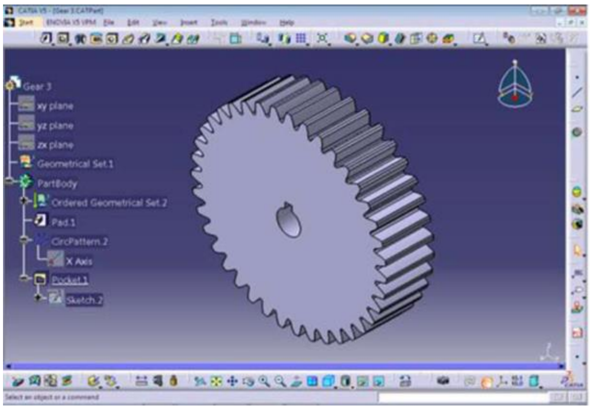


Fig.5. Gear 3

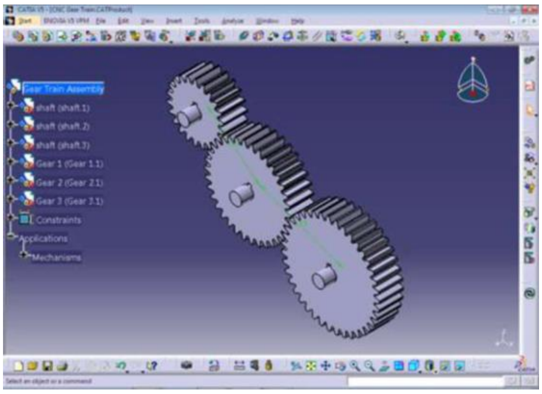


Fig.6. Gear Assembly

8. Finite Element Analysis Of Spur Gear :

Finite element modeling is described as the representation of the geometric model in terms of a finite number of element and nodes. It is actually a numerical method employed for the solution of structures or a complex region defining a continuum. Solutions obtained by this method are rarely exact. However, errors in the approximate solution can be minimized by increasing the number of equations till the desired accuracy obtained. This is an alternative to analytical methods that are used for getting exact solution of analysis problems. The solution of general problem by finite element method always follows an orderly step-by-step process. for analysis in ANSYS R 15.0. The loading conditions are assumed to be static.

Nalyatron NSM gear train

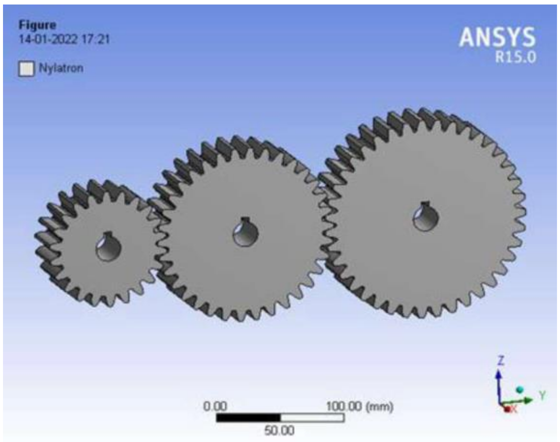


Fig.7. Loads and boundary condition

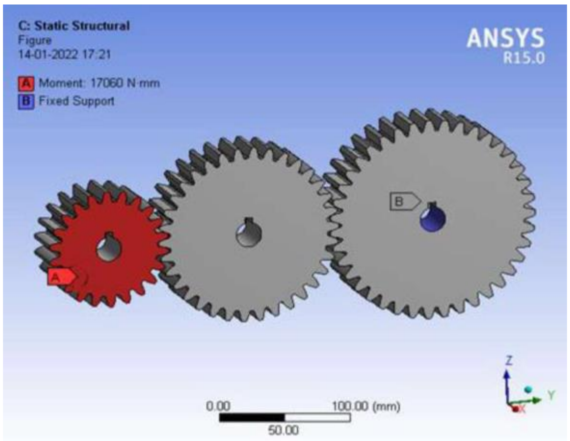


Fig. 8. Moment constraining of diagram in ansys

9. Result and Discussion

A] From the static analysis using Ansys the Total deformation, Equivalent strain and Von-mises stress for plastic material such as Nylatron (NSM), Nylon 66 and PTFE are obtained as following figures and tables below.

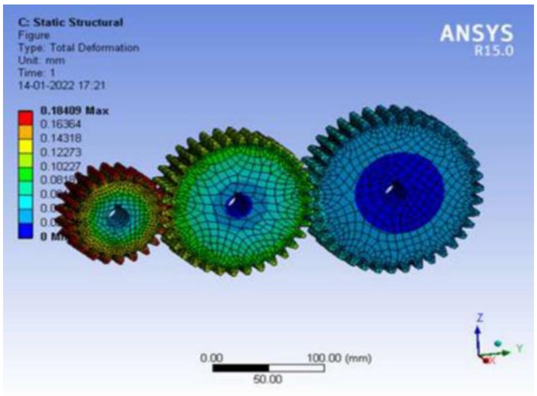


Fig. 9. Total Deformation

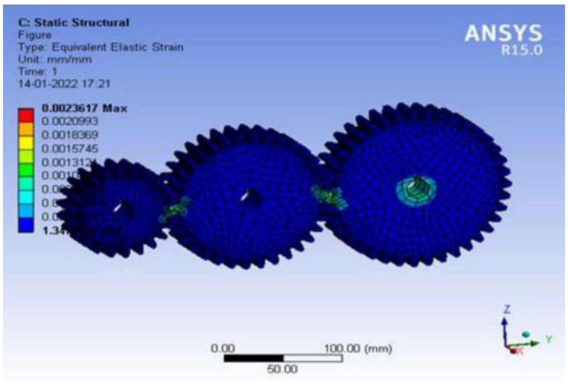


Fig.10. Equivalent Elastic Strain

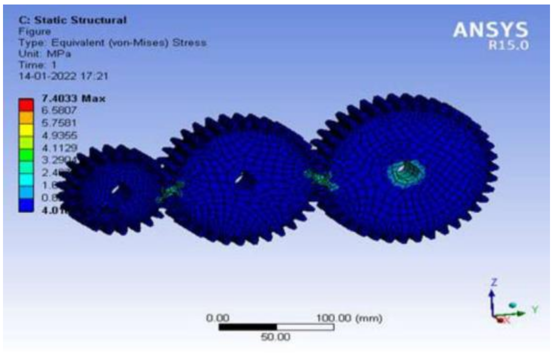


Fig. 11. Equivalent (von-mises) stress 1

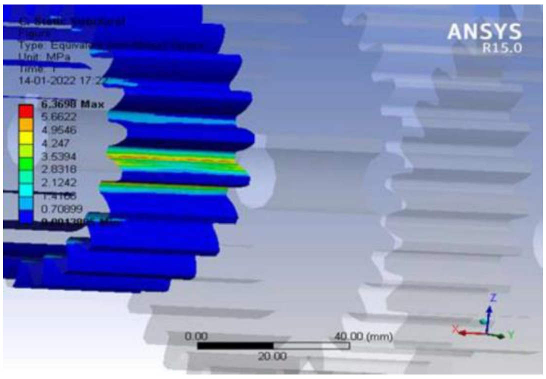


Fig. 12. Equivalent stress 2

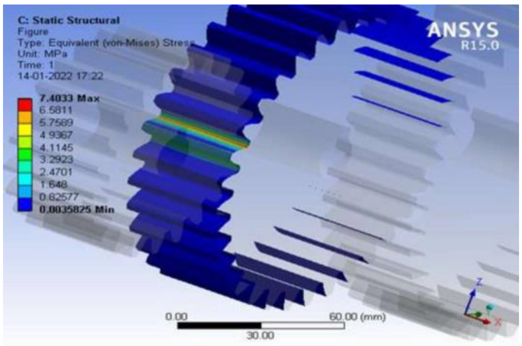


Fig. 13. Equivalent stress 3

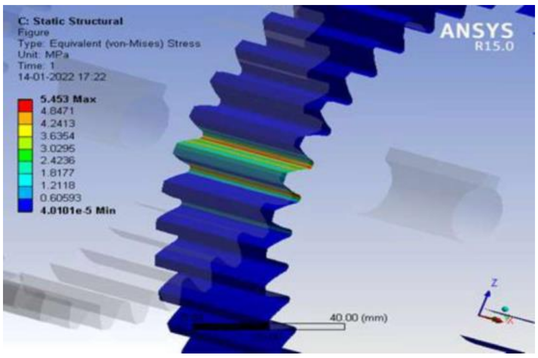


Fig. 14. Equivalent stress 4

Table 4. Ansys Results for plastic gear materials

Plastic material for Gear Train	**Von-Mises stress (Mpa)**	**Deformation (mm)**	**Strain**
Nylatron (NSM)	Stress (1) 7.4033 Stress (2) 6.3698 Stress (3) 7.4033 Stress (4) 5.453	0.18409	$2.3617\ e^{-3}$
Nylon66	Stress (1) 8.3511 Stress (2) 6.382 Stress (3) 7.5652 Stress (4) 8.3511	0.11471	$1.5522\ e^{-3}$
PTFE	Stress (1) 5.4588 Stress (2) 4.8875 Stress (3) 5.4588 Stress (4) 3.5144	0.88499	$9.9782\ e^{-3}$

B] From the static analysis using Ansys the frictional stress, pressure, sliding distance, penetration, and gap for plastic material such as Nylatron (NSM), Nylon 66 and PTFE are obtained as following figures and tables below.

Gear 2 to Gear 1

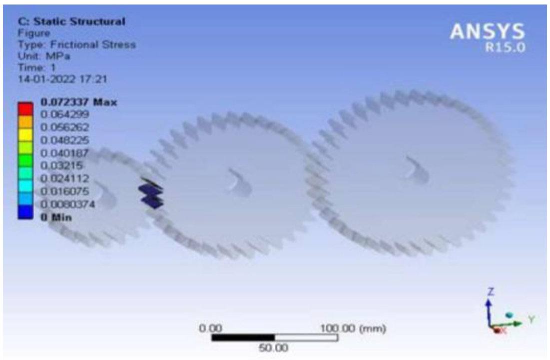


Fig. 15. Frictional Stress

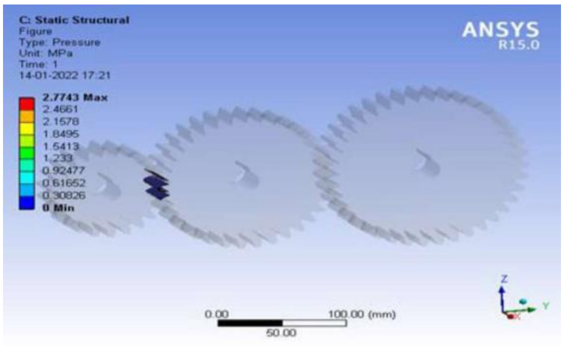


Fig. 16. Pressure

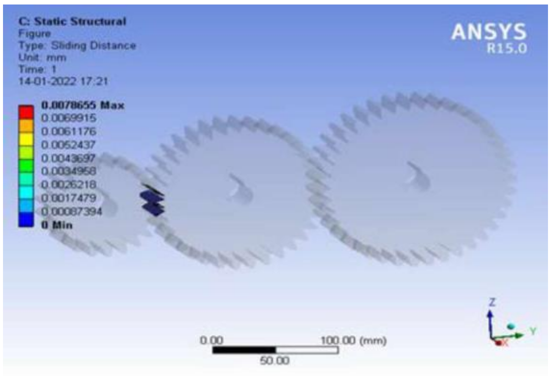


Fig.17.Sliding Distance

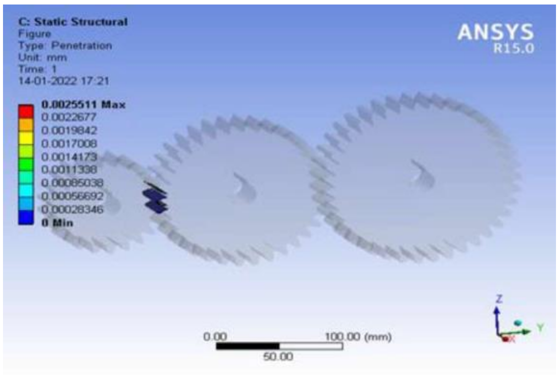


Fig.18. Penetration

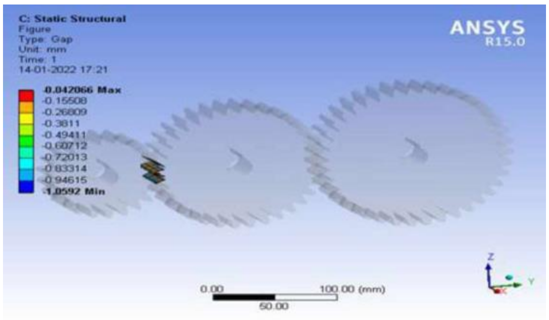


Fig.19. Gap

Gear 3 to Gear 2

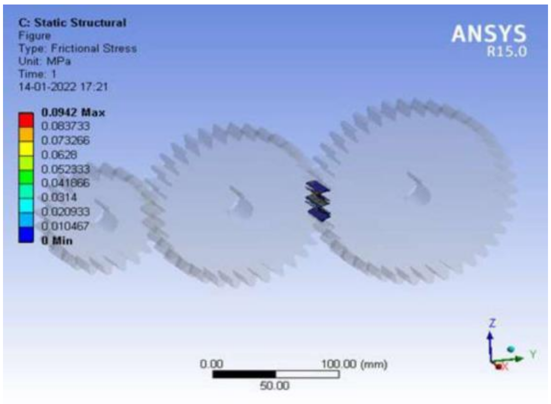


Fig.20.Frictional Stress

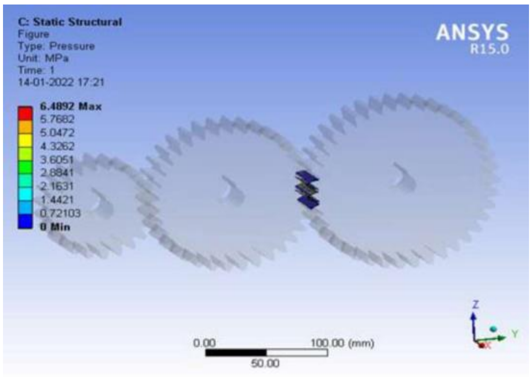
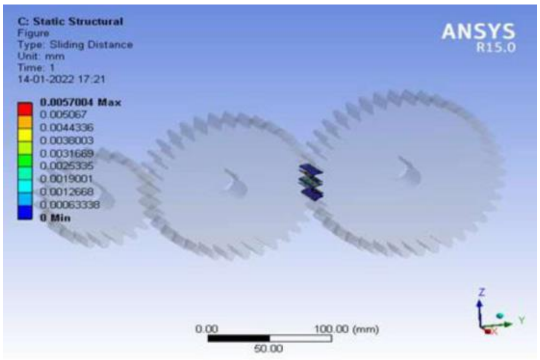


Fig.21. Pressure



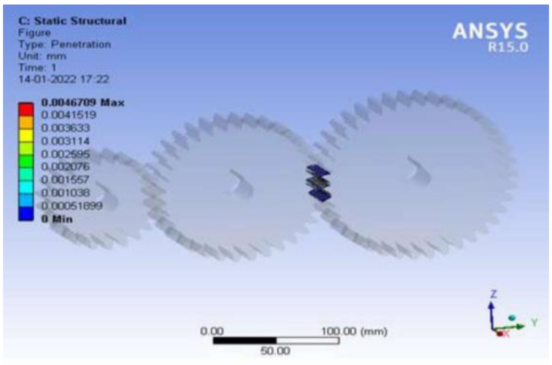


Fig. 22. Sliding Distance

Fig.23. Penetration

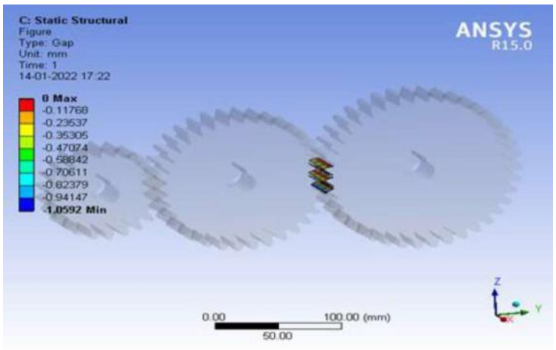


Fig.24. Gap

Table 5. Ansys Results for plastic gear materials

Plastic Material for Gear Train	**Position**	**Frictional stress (Mpa)**	**Pressure (Mpa)**	**Sliding Distance (mm)**	**Penetration (mm)**	**Gap (mm)**
Nylatron (NSM)	Gear 2 to 1	7.2337e^{-2}	2.7743	7.8655e^{-3}	2.5511e^{-3}	-4.2066e^{-2}
	Gear 3 to 2	9.42e^{-2}	6.4892	5.7004e^{-3}	4.6709e^{-3}	0
Nylon 66	Gear 2 to 1	5.5441e^{-2}	2.7692	5.0781e^{-3}	1.4584e^{-3}	--4.5865e^{-2}
	Gear 3 to 2	0.15323	9.3412	5.1216e^{-3}	3.8509e^{-3}	0
PTFE	Gear 2 to 1	0.27208	2.1867	0.13136	1.1517e^{-2}	-8.3393e^{-3}
	Gear 3 to 2	0.137	4.0649	3.7772e^{-2}	1.6758e^{-2}	0

10. Conclusion

In this paper reconstruction of gear train in feed drive mechanism for lathe machine is proposed; in which design and analysis of polymer gears for drive mechanism with use of Nylatron, Nylon 66, PTFE has been presented.

a) As a result of FEA analysis of these polymer gears for feed drive mechanism, the following conclusions were reached.

b) As the driving torque or moment 17.06 N-m is applied to gear then maximum contact stress (von-mises) produced is 7.7033 mpa for Nylatron, 8.3511 mpa for Nylon 66 and 5.4588 mpa for PTFE.

c) The Maximum total deformation value reaches upto the 0.18409 mm for Nylatron, 0.11471 mm for Nylon 66 and 0.88499 mm for PTFE

d) The Equivalent strain value is 2.3617e^{-3} for Nylatron, 1.5522e^{-3} for Nylon 66 and 9.9782e^{-3} for PTFE.

e) The frictional stress is 9.42e-2 mpa for Nylatron, 0.15323 mpa for Nylon 66 and 0.137 mpa for PTFE.

f) The maximum pressure is 6.4892 mpa for Nylatron, 9.3412 mpa for Nylon 66 and 4.0649 for PTFE.

g) Sliding distance, penetration and gap values for gear pair are at acceptable limit.

With above result, It is concluded that the use of polymer gears in the drive mechanism for lathe machine is very beneficial as system becomes noiseless, less weight and reducing vibration. All the stresses produced by applying the given moment are at acceptable limit for given m

aterials.

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