# MODELING AND SIMULATION OF STATIC AND DYNAMIC ANALYSIS OF SPUR GEAR

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**Abstract -** *Gears are one of the most important machine component and are generally used where one need to transmit power with minimum transmission losses. In literature survey we have seen that a lot of research is carried out for gear i.e. modeling of gear using different modeling software's, on material selection of gears, on different helical profiles, on different loading conditions, and also the same are analyzed using some advanced analysis software for result validation.*
*Gears are used for power transmission in many mechanical devices. A spur gear is a gear in which leading edges are parallel to the axis of the gear. In present work the spur gear in mesh are analyzed when they are in running condition i.e., in dynamic motion.*
*We will use CATIA software for part modeling and also for assembly of parts to form the final product of two spur gears in mesh. The analysis part is to be carried out using ANSYS software in which we expect to found the deformation and stresses for spur gears in mesh for three different materials. In result part comparative graphs will be made both for deformation and stresses to find which material gears are best suitable to manufacture for gear applications.*

***Key Words*: Gear Pair, Dynamic Analysis, CATIA, Ansys14.0**

## 1. INTRODUCTION

Nowadays, gears are one of the most common and important components of machines of various kinds. Gears systems are used to reduce the rotational speed, increase the torque applied, change the direction of the power transmissions and distribute the available power between several machines.

Gears in a simple language can be defined as a member who has wheel with teeth. The primary function of gears is to turn and for which they are to be in mesh with other gears. They are basically used to transmit power of a moving part to the other.

### 1.1. Purpose of using Gears

Gears can increase or decrease the speed and power of a vehicle the main purpose of using gears are:

1. To reverse the direction of rotation
2. To increase or decrease the speed of rotation
3. To move rotational motion to a different axis
4. To keep the rotation of two axis synchronized

### 1.2. Types of Gears

1. Spur Gear: Are the one which have tooth parallel to the gear axis and are the simplest type of the gears known.
2. Bevel Gear: The tooth of gears are at some inclination with their axis and are generally used for 90 degree power transmission
3. Rack and Pinion Gear: It has a stationary straight rack over which a moving rotory pinion rotates. It is generally used to transmit power linearly.
4. Worm and Worm Wheel: the axis of worm and worm wheel are intersecting type and are used where one need to transmit power at a highly reduced speed. Worm is the driven member where as the worm wheel rotates with respect to worm.
5. Compound Gears: When one or more intermediate gears are placed between driving and driven members with purpose such as in case of gear box.
6. Gear Trains: A gear train consist of a combination of two or more gears mounted on rotating shaft to transmit power and as speed reducer or speed increaser. They may be simple, compound, reverted and epicyclic gear trains.
7. Idler Gear: To transmit power in same direction as in driving gear an idler gear is used.

### 1.3. Manufacturing of gears

Manufacturing of gears desires many process operations in consecutive stages relying upon the material and kind of the gears and quality desired.

Those stages usually are :

- Pre-forming the blank without or with teeth
- Annealing of the blank, if needed, as just in case of forged or cast steels

- Preparation of the gear blank to the specified dimensions by machining
- Manufacturing teeth or finishing the preformed teeth by machining
- Full or surface hardening of the machined gear (teeth), if needed
- Finishing teeth, if needed, by shaving, grinding etc.
- Examination of the finished gears

## 2. LITERATURE REVIEW

Pawar, P. B[1]manufactured metallic gears of steel alloy and Aluminium Silicon carbide composite and found that Composites provide much improved mechanical properties such as better strength to weight ratio, more hardness, and hence less chances of failure.Sánchez, M. B [2] evaluated the meshing stiffness of spur gear pairs, considering both global tooth deflections and local contact deflections, at any point of the path of contact and approximated by an analytical, simple function.Pawar, P. B [3] done themodeling and finite element analysis of gear using CATIA and ANSYS 14.0. In case of increased silicon carbide content, the hardness, and material toughness are enhanced. From the results it is concluded that composite material such as aluminum silicon carbide is one of the option as a material for power transmission gears.

Syzrantseva, K [4] calculated the stress-strain condition of the teeth based on the Finite Element Method. The stress loading of the contact lines of gearing is determined based on matching the photos of ISG (Integral Strain Gauge) reaction and the picture of compression strain distribution. Raptis, K. G [5] In this work, the maximum stresses, which are developed in the crucial toe section of spur gears being in contact and subjected to loading at the Highest Point of Single Tooth Contact (HPSTC), are numerically and experimentally studied.Patil, S. [6] an attempt has been made to study the effect of static coefficient of friction along the line of action of a spur gear.

A 3D finite element method was used for the modeling and analysis of the contact between the gear tooth flanks. The involute gear profile was generated using a macro in ANSYS APDL. Marimuthu, P [7] attempts are made for a reasonably accurate estimation of optimum profile shifts based on pinion and gear tooth fillet strength for a load at critical loading point considering the load sharing between the gear pair on asymmetric normal and high contact ratio asymmetric spur gear through direct design.

### 2.1. CONCLUSION FROM LITERATURE REVIEW:

**1.** The gears of Aluminum Silicon carbide composite can be made which offers improved properties over steel alloys and these can be used as better alternative for replacing metallic gears.
**2.** Ifwe use increased silicon carbide content, the hardness, and material toughness are enhanced. From the results it is concluded that composite material such as aluminum silicon carbide is one of the option as a material for power transmission gears.
**3.** The involute gear profile was generated using a macro in ANSYS APDL.
**4.** For optimum design of gear drive the maximum fillet stresses developed at the pinion and gear have to be equal.
**5.** The accuracy in manufacturing accuracy of the tooth profile is essential as the intensity of the excitation depends on it.

## 3. METHODOLOGY

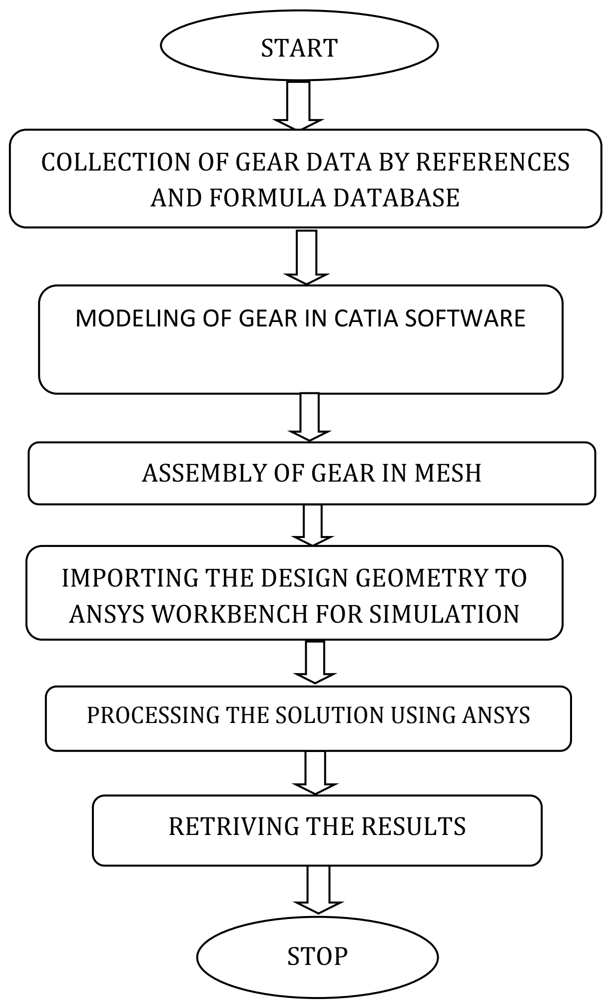


**Fig-1:** Flow chart of Gear Simulation

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Standard gear teeth-200 stub involute
Material = Steel
Poisson's Ratio μ = 0.27 to 0.30
Young's Modulus E = 2.1*105 N/mm$^2$
Module m = 10
Speed of pinion n1 = 1800 rpm
Gear ratio i = 1:3
Pinion no. of teeth Z1 = 12
Life of teeth LH= 20000 hr
Pressure angle (α) = 20̊
Power P = 18 KW

**Calculation:**

Speed of gear $n_2$ = 1800/3= 600 rpm
Gear no. of teeth $Z_2$ = 12*3 = 36
Pitch circle diameter for pinion,
d1 = m*$z_1$ = 10*12 = 120 mm
Pitch circle diameter for Gear,
d2 = m*$z_2$ =10*36 = 360 mm
Center distance of gear & pinion
a = ($z_2$+$z_1$)m/2 = (36+12)10/2 = 240 mm
Base circle diameter for Pinion,
$db_1$ = $d_1$*cosα =120*cos200 =112.76 mm
Base circle diameter for Gear,
$db_2$ = $d_2$*cosα = 360*cos200 = 338.3 mm
Tip circle diameter for pinion,
$da_1$ = $d_1$ + 2m =120+2*10 = 140 mm
Tip circle diameter for gear,
$da_2$ = $d_2$ + 2m=360+2*10 = 380 mm
Root circle diameter for pinion,
$df_1$ = $d_1$ – 2*1.25m = 120-2*1.25*10 = 95 mm
Root circle diameter for gear,
df2 = d2 – 2*1.25m = 360-2*1.25*10 = 335 mm

**FOR PINION:**
Pitch line velocity
V = π *d1*n1/(60*1000)=11.31 m/sec

Pinion life L = 60 n1LH = 60*1800*20000
= 2160000000 revolutions
= 216*10$^7$rev

Pinion Torque
Mt = P*60 / (2 πn1) = 10000*60/(2*3.14*1800)
= 95.5 N m

$$\sigma_b = \frac{1.4Kbl}{nK\sigma}\,\sigma$$

= (1.4*0.7*532.25) / (2.5*1.5) = 139.1N/mm$^2$

**FOR GEAR**
Gear life L = 60*600*20000 = 7.20000000 rev.
= 72*10$^7$rev.

Gear Torque Mt = P*60/(2 πn2) = 286.48 Nm
Material : CS 85, case hardness 55 RC and core hardness greater than 350 BHN.
Design stresses

$$\sigma_b = \frac{1.4Kbl}{nK\sigma}\,\sigma$$

= (1.4*0.7*393.2) / (2.0*1.2) = 160.556 N/mm$^2$

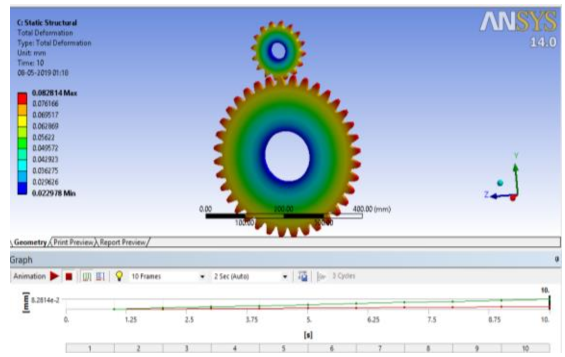


Figure 1 – Obtained deformation of pinion and gear assembly (Considering torque on pinion)

Figure 1 represents the Obtained deformation of pinion and gear assembly considering torque on pinion. It is observed that the total maximum deformation is 0.08 mm which is negligible as per the gear mesh is concerned.

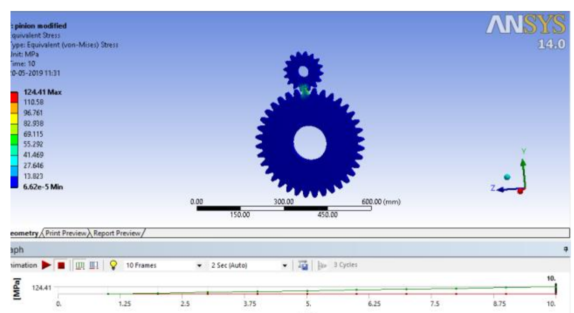


Figure 2 – Equivalent von misses stress of gear and pinion assembly
(Considering torque on pinion)

Figure 2 represents the Equivalent von misses' stress of gear and pinion assembly obtained as 124.41 N/mm$^2$. It is a little low as compared to the numerically obtained data that is 139.1 N/mm$^2$ this may be due to environmental factors like friction between gears and due to frictionless support considered at its hub.

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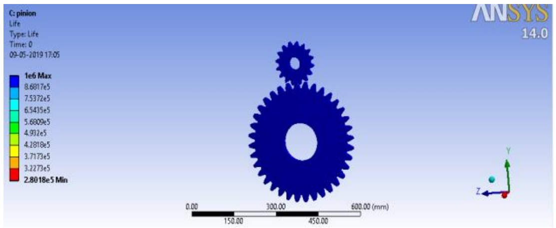

Figure 3 – Life of the gear-pinion assembly considering torque on pinion

Figure 3 represents the fatigue life or durability of the gears in mesh with respect to the given boundary conditions and it is obtained of the range of $10^6$ cycles.

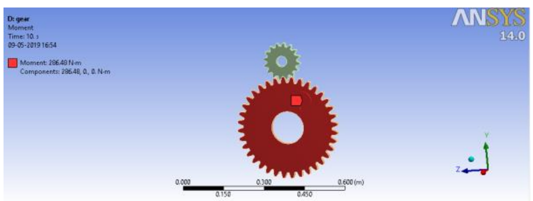

Figure 4 – Moment of 286.48 N-m for pinion in anticlockwise direction

The figure 4 above represents the moment provided at the gear and it taken as 286.48 N-m which is same as taken in the above numerical data.

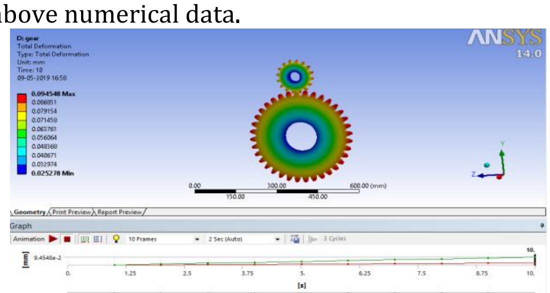

Figure 5 – Obtained deformation of pinion and gear assembly
(Considering torque on gear)

Figure 5 represents the Obtained deformation of pinion and gear assembly considering torque on gear. It is observed that the total maximum deformation is 0.09 mm which is negligible as per the gear mesh is concerned.

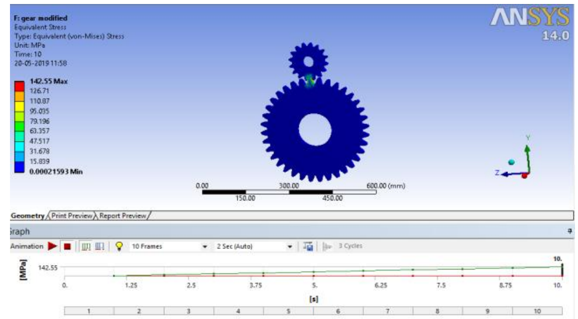

Figure 6 – Equivalent von misses stress of gear and pinion assembly (Considering torque on gear)

Figure 6 represents the Equivalent von misses' stress of gear and pinion assembly obtained as 142.55 N/mm$^2$considereing torque on gear. It is a little also found low as compared to the numerically obtained data that is 166.556 N/mm$^2$ this is again due to environmental factors like friction between gears and due to frictionless support considered at its hub.

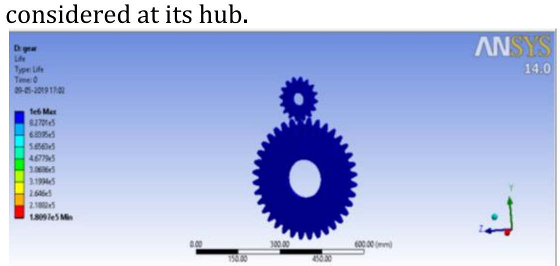

Figure 7 – Life of gear considering gear torque.

Figure 7 represents the fatigue life or durability of the gears in mesh with respect to the given boundary conditions considering torque on gear and it is obtained of the range of $10^6$ cycles.

**Table-1:**

| Sr. No. | Time Step (S) | Respective Deformation |
|---|---|---|
| 1 | 1 | 2.1835e-010 |
| 2 | 2 | 9.0344e-006 |
| 3 | 3 | 1.826e-005 |
| 4 | 4 | 2.7482e-005 |
| 5 | 5 | 3.6704e-005 |
| 6 | 6 | 4.5926e-005 |
| 7 | 7 | 5.5148e-005 |
| 8 | 8 | 6.437e-005 |
| 9 | 9 | 7.3592e-005 |
| 10 | 10 | 8.2814e-005 |

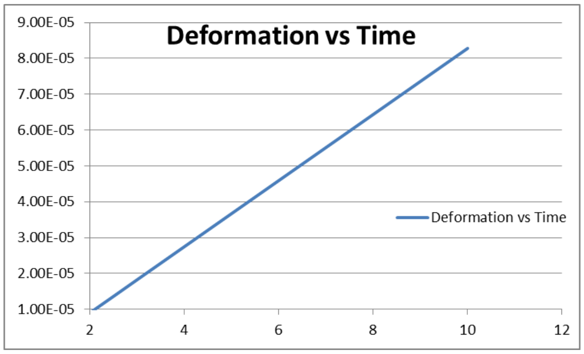


Figure 8- Figure Representing value of deformation with respect to time in seconds for torque in pinion

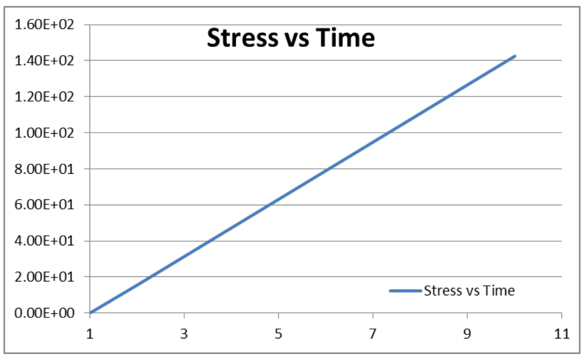


Figure 9- Figure Representing value of stress with respect to time in seconds for torque in pinion

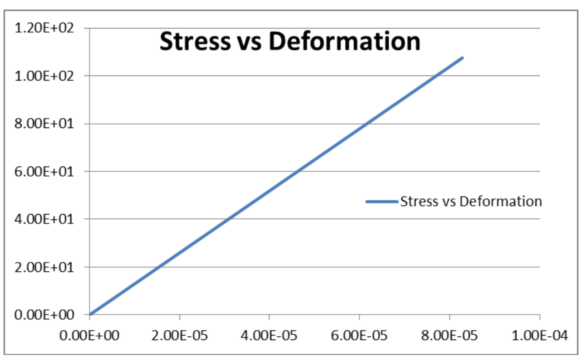


Figure 10- Figure Representing value of stress with respect to deformation at different time intervals for torque in pinion

TABLE 1 : Comparative von misses stress between numerical and FEA results

| **Sr. No.** | **Parameter** | **Numerical Result** | **FEA Result** | **% Difference** |
|---|---|---|---|---|
| 1 | Stress generated for pinion | 139.1 N/mm$^2$ | 124.41 N/mm$^2$ | 10.5 % |
| 2 | Stress generated for gear | 160.556 N/mm2 | 142.55 N/mm$^2$ | 11.21 % |

Table 2 – Material properties of structural steel taken into account

**Structural Steel > Constants**

| | |
|---|---|
| Density | 7.85e-006 kg mm^-3 |
| Coefficient of Thermal Expansion | 1.2e-005 C^-1 |
| Specific Heat | 4.34e+005 mJ kg^-1 C^-1 |
| Thermal Conductivity | 6.05e-002 W mm^-1 C^-1 |
| Resistivity | 1.7e-004 ohm mm |

**Considering Buckingham's Dynamic Load Case**

Table 2: Comparative von misses stresses for Buckingham's dynamic load case

Fd = Ft +[0.164 Vm (c*b + Ft)/[0.164 Vm + 1.486√(c*b+Ft)]]

Where, vm = pitch line velocity, =11.31m/s or 678.6 m/min

Vm = 678.6 m/min

b = face width = 72mm = 0.072 m

c = factor depending on machining error = 12300e = 12300*0.03 = 369

Ft = transmitted load, kgf

Ft = HP*75/vm

= 24.14*75/11.31= 24.14 HP
= 160.08 kgf or 1570.4 N

Fd=160.08+[0.164*678.6 [369*0.072+160.08]/[0.164*678.6+1.486√[369*0.072 + 160.08]]]

= 160.08+[20988.5/131.6]

= 319 kgf or 3134.95 N

= 3.135 KN = 3135 N

Pinion Moment = 3135*70 /100= 2194 N-m

Gear Moment = 3135*190/100= 7476.5 N-m

σ = Ft /bYm

= 1570.4 / 94.25*0.379*.3 = 146.544 (For pinion)

=1570.4 / 94.25_0.34*.3= 163.35 (For gear)

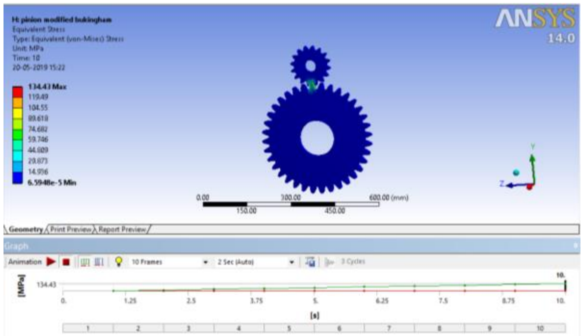

Figure 11: Equivalent von misses stresses Buckingham's dynamic load case for torque on pinion

Figure 11 represents the von misses stresses generated using Buckingham dynamic load case for providing a torque on pinion and it is observed that generated stress is around 8.27 % deviated from the actual numerical stress

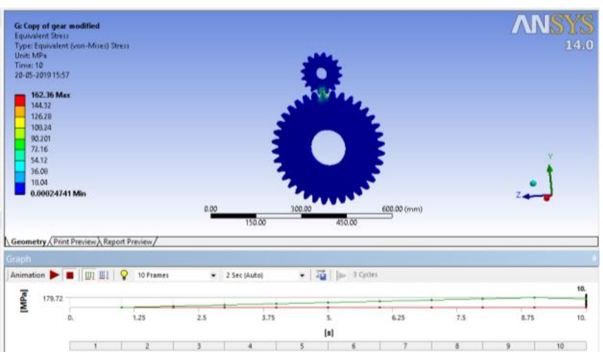

Figure 12: Equivalent von misses stresses Bukingham dynamic load case for torque on gear

Figure 12 represents the von misses stresses generated using Buckingham dynamic load case for providing a torque on gear and it is observed that generated stress is around 0.6 % deviated from the actual numerical stress.

**Considering Lewis Equation of dynamic load**

Fd = Ft* Cv
Ft = HP*75/ vm
vm = 11.31 m/s
P = 24.14 hp

Cv=(6+Vm)/6

= (6+11.31)/6 = 2.885

Ft = 24.14*75/11.31

= 160.08 kgf or 1570.4 N

Fd = 160.08*2.885

= 461.831 kgf

= 4530.6 N or 4.531 KN

Pinion Moment = 4530.6 *70 /100= 3171.42 N-m

Gear Moment = 4530.6*190/100= 8266.14 N-m

The Lewis equation indicates that tooth bending stress varies with the following:

= 1570.4 / 94.25*0.379*.3 = 146.544 (For pinion)

=1570.4 / 94.25*0.464*.3= 119.698 (For gear)

(1) Directly with load,

(2) Inversely with tooth width b,

(3) Inversely with tooth size p or m,

(4) Inversely with tooth shape factor y or Y.

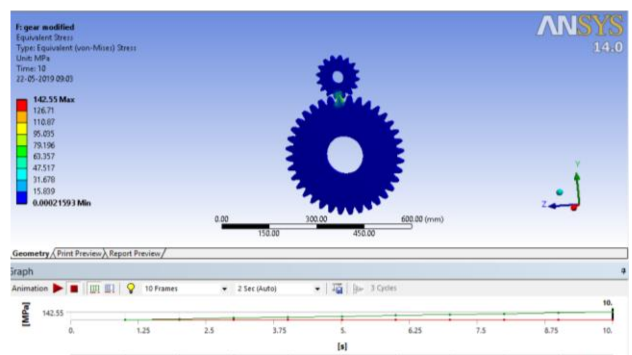

Figure 13: Equivalent von misses stresses Lewis dynamic load case for torque on pinion

## 4. RESULT AND DISCUSSIONS:

After studying the literature and knowing the facts about spur gear one may conclude the main points on which work is actually done till date is

1. As seen from literature review the material properties of gear as Aluminium silicon carbide will provide improved properties over steel alloys and can be implemented to see the comparison between the performances of the two.
2. Proper generation of tooth profile is must as the intensity of excitation depends on it.
3. Static as well as explicit analysis are not performed on the spur gear so far in virtual environment and can be

simulated to see the behaviour of gear in static and dynamic conditions.

Table 4.1: Comparative von misses stress between numerical and FEA results

| Sr. No. | Parameter | Numerical Result | FEA Result | % Difference |
|---|---|---|---|---|
| 1 | Stress generated for pinion | 146.544 N/mm2 | 142.55 N/mm2 | 2.74 |
| 2 | Stress generated for gear | 119 N/mm2 | 124.41 N/mm2 | 4.36 |

Table 4.2: Comparative von misses stresses for Buckingham's dynamic load case

| Sr. No. | Parameter | Numerical Result | FEA Result | % Difference |
|---|---|---|---|---|
| 1 | Stress generated for pinion | 139.1 $N/mm^2$ | 124.41 $N/mm^2$ | 10.5 |
| 2 | Stress generated for gear | 160.556 N/mm2 | 142.55 $N/mm^2$ | 11.21 |

Table 4.3: Comparative von misses stresses for Lewis dynamic load case

| Sr. No. | Parameter | Numerical Result | FEA Result | % Difference |
|---|---|---|---|---|
| 1 | Stress generated for pinion | 146.544 $N/mm^2$ | 134.43 $N/mm^2$ | 8.27 % |
| 2 | Stress generated for gear | 163.35 N/mm2 | 162.36 N/mm2 | 0.60 % |

## 5. CONCLUSION:

**1.** It is observed that stress obtained for pinion numerically is 139.1 N/mm2, while in software based solution it is 124.41 N/mm2. This is around 89.5 % closer to that of numerical one.

**2.** It is observed that stress obtained for gear numerically is 160.5 N/mm2, while in software based solution it is 142.5 N/mm2. This is around 88.79 % closer to that of numerical one.
**3.** The deformation is almost negligible and hence our design is safe.
**4.** The fatigue life obtained numerically is of the range of 107 cycles which is also justified by simulation as software gives a fatigue life of around 106 cycles.
**5.** The possible reason for the deviation of results from numerical data may be due to certain reasons like we have not consider friction while simulating our gear assembly, We have taken a fine meshing size but it may be possible that by changing the mesh size even more finer may give more better results.
**6.** A lot of boundary conditions are required to simulate a simple gear-pinion problem so one needs to be
**7.** well known with the theory behind to simulate any real world problem.
**8.** The value of stress as well as deformation changes linearly w.r.t time. The same correlation can be seen in stress vs. deformation graph.
**9.** It can be observed that using Buckingham and Lewis load cases the numerical and FEA results are even more closer and are satisfactory closer to validate the ANSYS solver.

## 6. FUTURE SCOPE:

1. The effect of changing the mesh size can be done to get closer results w.r.t numerical calculations.
2. The friction between gears and pinion as well as between gears/pinion and shaft can be taken into account.
3. The gear material can also be changed to see which material would last with greater fatigue life cycles.
4. The same analysis can be performed at different load cases and at different angular velocities.
5. Instead of pinion one can run gear by providing a suitable moment to it to get the simulation results.

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