INTERNATIONAL JOURNAL OF SCIENTIFIC RESEARCH IN ENGINEERING AND MANAGEMENT (IJSREM)
VOLUME: 09 ISSUE: 04 | APRIL - 2025 SJIF RATING: 8.586 ISSN: 2582-3930

Stress Analysis of Spur Gear Using Ansys

[1]B prasad , G siva manga Prasad, G jagadeesh, K vara Prasad, K kishore kumar, G surya teja.

Mr k joseph noel
Department of Mechanical Engineering
Welfare Institute of Technology & Management, Pinagadi , Visakhapatnam

ABSTRACT

Gear drive plays vital role in power transmission industries. Gears are usually subjected to fluctuating loads. Due to these loads bending and compressive stresses will be developed in the gears. While designing the gear it is very important to analyze the stresses for safety operation, and weight reduction of gear is also one of the design criteria. In this project, the spur gear is modelled in "CATIA V5" and imported to "ANSYS" for static structural analysis and modal analysis. Static analysis is performed to determine the deformation and Von- misses stresses. Modal analysis is performed to determine the natural frequencies and mode shapes. The results were validated with theoretical calculations by Lewis equation.

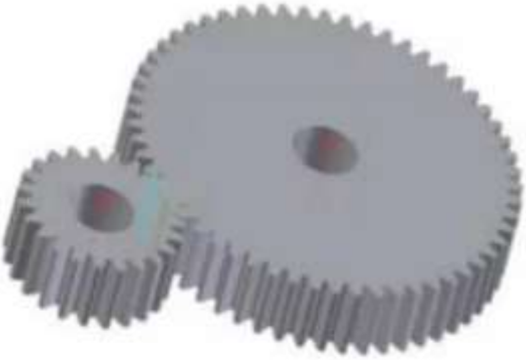
Analysis is done by considering different materials for gears like Cast iron, carbon steel, brass, copper. and results are compare

Key words: ANSYS, Static analysis

INTRODUCTION

Gear is an essential component in many machine parts; its application varies from small geared motor to and complicated aerospace accessories. Human has been familiar about the idea that the repeated bending of wood or metal back and forth with high amplitude could rupture it. He found that the repeated stress can produce fracture with stress within elastic limit of material. The fatigue analysis for structure designing relies on approach which has been progressed over the last 100 years or so. The very first fatigue analysis has been done by German mining engineer, W.A.S. Albert who performed number of repeated loading test on iron chain. Fatigue is the most important failure mode to be considered in a mechanical design. Fatigue is the process of continuous localized permanent structural change appearing in a material subjected to fluctuating stress conditions. If the loading limit does not exceed the elastic limit, the body will regain its original state. Designer should have a good knowledge of analytical and empirical techniques to get effective results in averting failure. Mechanical failure is observed mainly due to fatigue design therefore fatigue becomes an obvious design, consideration for many structure such as aircraft, rail cars, automotive suspension, Vehicle frame and bridges in normal conditions, contact fatigue is one of the most common failure modes for gear tooth surfaces. Gear tooth interaction causes adhesive wear throughout the life of gear drive.

I. SPUR GEAR



Spur gears are regularly used for speed reduction or increase, torque multiplication, resolution and accuracy enhancement for positioning systems. The teeth run parallel to the gear axis and can only transfer motion between parallel-axis gear sets. Spur gears mate only one tooth at a time, resulting in high stress on the mating teeth and noisy operation

Dimension Specifications:

Gears mate via teeth with very specific geometry. Pitch is a measure of tooth spacing and is expressed in several ways.

Diametral pitch (DP) is the ratio of the number of teeth to the pitch diameter of a gear; a higher DP therefore indicates finer tooth spacing. It is easily calculated by the formula **DP= (N+2)/OD**, where N is the number of teeth, and OD represents the circumferential measurement.

Circular pitch (CP) is a direct measurement of the distance from one tooth center to the adjacent tooth center. It can be measured by the formula **CP= Π + DP.**

Module (M) is a typical gear discipline and is a measurement of the size and teeth number of the gear. Gears measured in inches earn English module' distinction to prevent confusion**. M = OD/N**

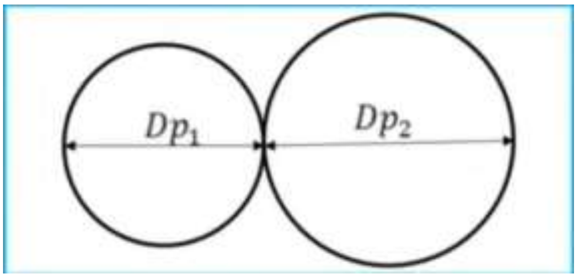
Pressure angle is the angle of tooth drive action, or the angle between the line of force between meshing teeth and the tangent to the pitch circle at the point of mesh.

Typical pressure angles are 14.5° or 20°

A. *Design For Space Constrains*

The designed gear system should fit within a space limit. So the designer could say if he sums pitch diameters of the mating gears, it should be less than or equal to allowable space limit as shown in figure below.

The blue rectangle represents space on which gear should get fit. One can take 80% of width of this space as allowable width for gear design. So following is the relation obtained by this condition.

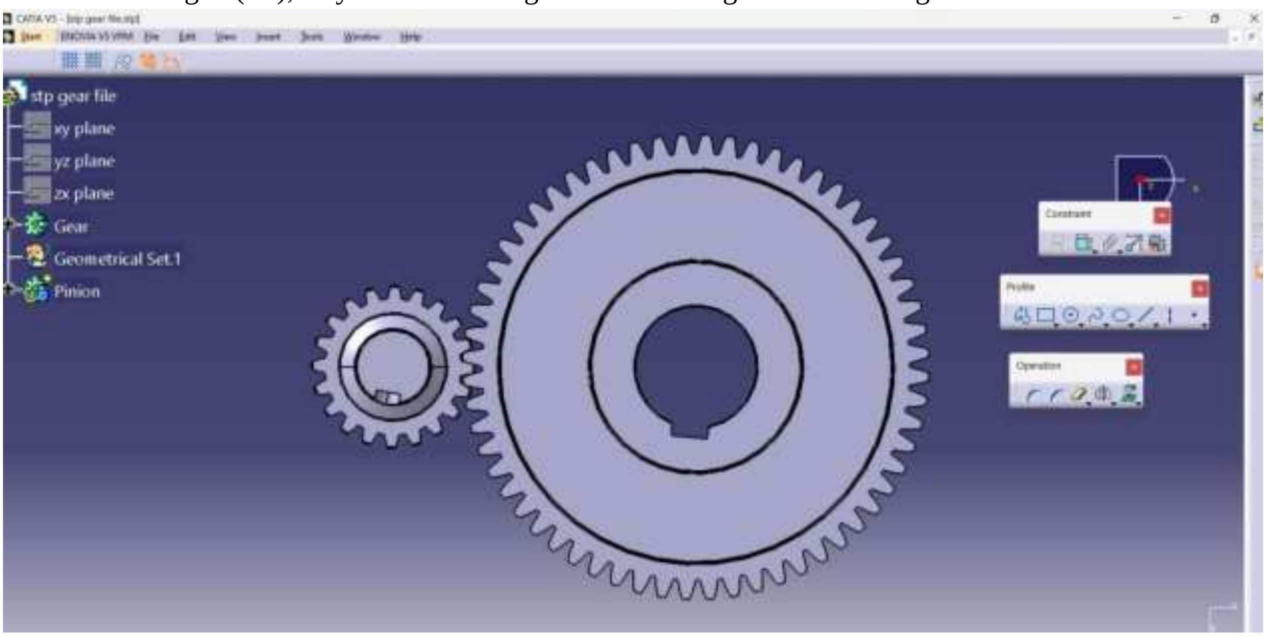


We also know speed ratio of gears, this will lead to one more relation in terms of pitch circle diameters.

Dp1/Dp2 = RPM2/RPM1

By solving above 2 equations simultaneously we can obtain pitch circle diameters of both the gears.Determination of Number of Teeth – Interference

Here we will understand how to determine number of teeth on both the gears. To do this we have to assume number of teeth on one gear(T1), say the smaller gear. Now using the relation given below we can determine number



of teeth on other gear, T2.

$$T_2 = \frac{T_1}{D_{p1}} D_{p2}$$

So we got number of teeth on both the gears, but one should also check for a phenomenon called interference if gear system has to have a smooth operation. Interference happens when gear teeth has got profile below base circle. This will result high noise and material removal problem. This phenomenon is shown in following figure.

If one has to remove interference, the pinion should have a minimum number of teeth specified by following relation.

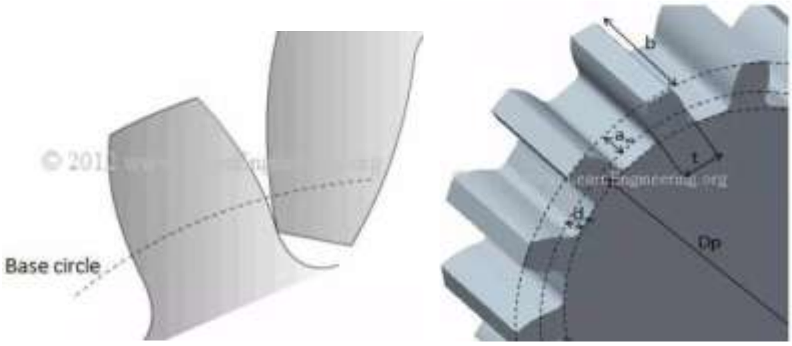


Fig.4A pair of gear teeth under interference

$$T_1 \geq \frac{2a_w \frac{1}{T_2} P_d}{\sqrt{1 + \frac{1}{T_2}\left(\frac{1}{T_2} + 2\right) sin^2\phi} - 1}$$

Where aw represents addendum of tooth. For 20 degree pressure angle(which is normally taken by designers) aw= 1 m and bw = 1.2 m. Module m, and pitch circle diameter Pd are defined as follows.

M=DP/T Pd =T/DP

Design for Mechanical Strength - Lewis Equation

Now the major parameter remaining in gear design is width of the gear teeth, b. This is determined by checking whether maximum bending stress induced by tangential component of transmitted load, Ft at the root of gear is greater than allowable stress. As we know power transmitted, P and pitch line velocity V of the gear Ft can be determined using following relation

F_tV = Power Transmitted

Here we consider gear tooth like a cantilever which is under static equilibrium. Gear forces and detailed geometry of the tooth is shown in figure below

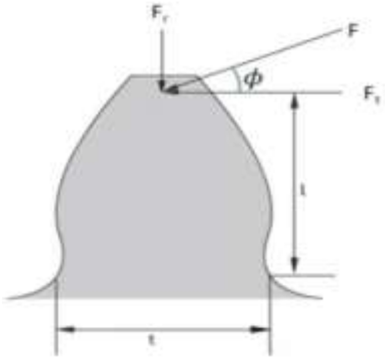


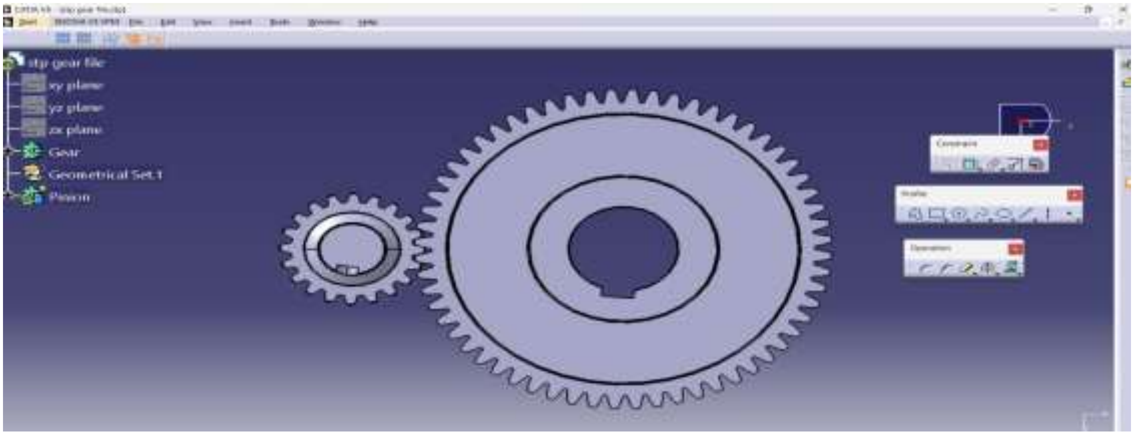
Fig.6 Geartooth underload

One can easily find out maximum value of bending stress induced if all geometrical parameters shown in above figure are known. But the quantities t and 1 are not easy to determine, so we use an alternate approach to find out maximum bending stress value using Lewis approach. Maximum bending stress induced is given by Lewis bending equation as follows.

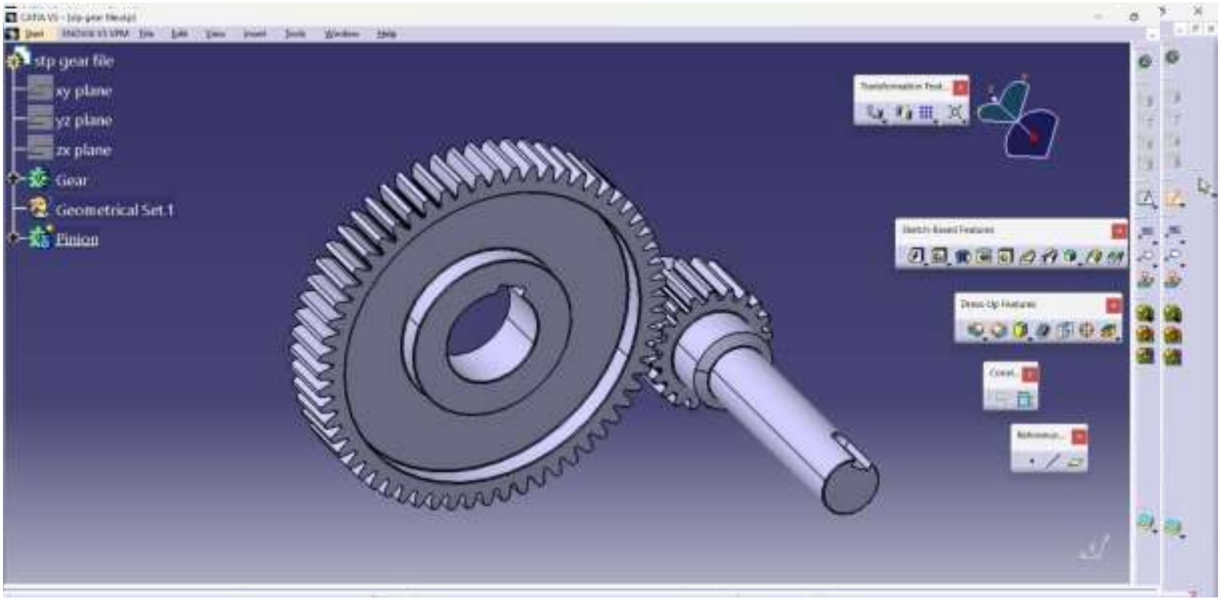
$$\sigma = \frac{F_t P_d}{bY}$$

Where Y is Lewis form factor, which is a function of pressure angle, number of teeth and addendum and dedendum. Value of Y is available as in form of table or graph. Using above relation one can determine value of b, by substituting maximum allowable stress value of material in LHS of equation. But a gear design obtained so will be so unrealistic, because in this design we are considering gear tooth like a cantilever which is under static equilibrium. But that's not the actual case. In next session we will incorporate many other parameters which will affect mechanical strength of the gear in order to get more realistic design.

CATIA:



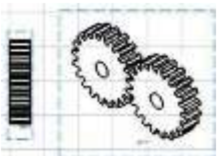
Views of Spur Gear & Pinion:

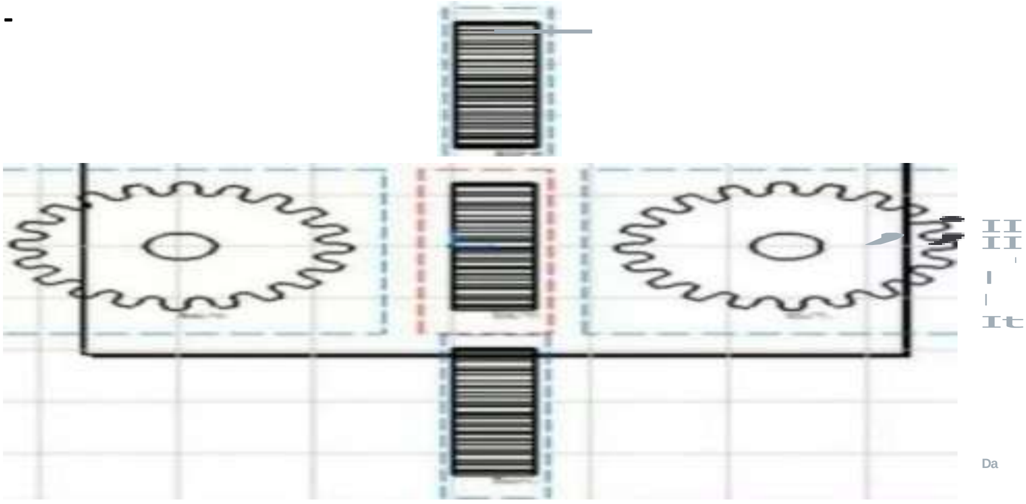


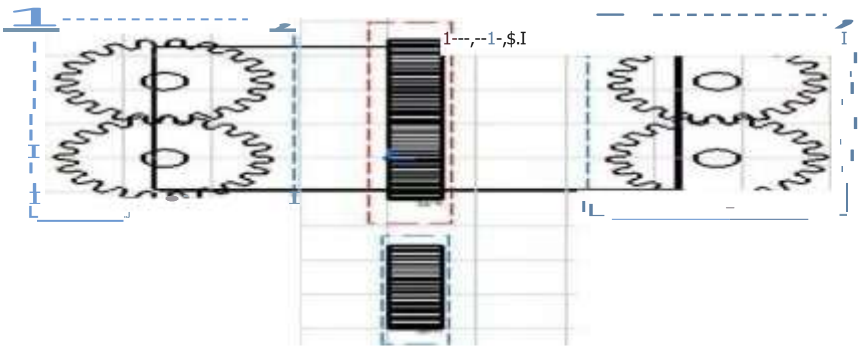
Bill of Material[BOM]

The Bill of Material, or parts list, corresponds to information on the product from which the views were generated. It consists of an itemized list of the parts of a structure shown on a drawing or on an assembly.

Views of spur gear:





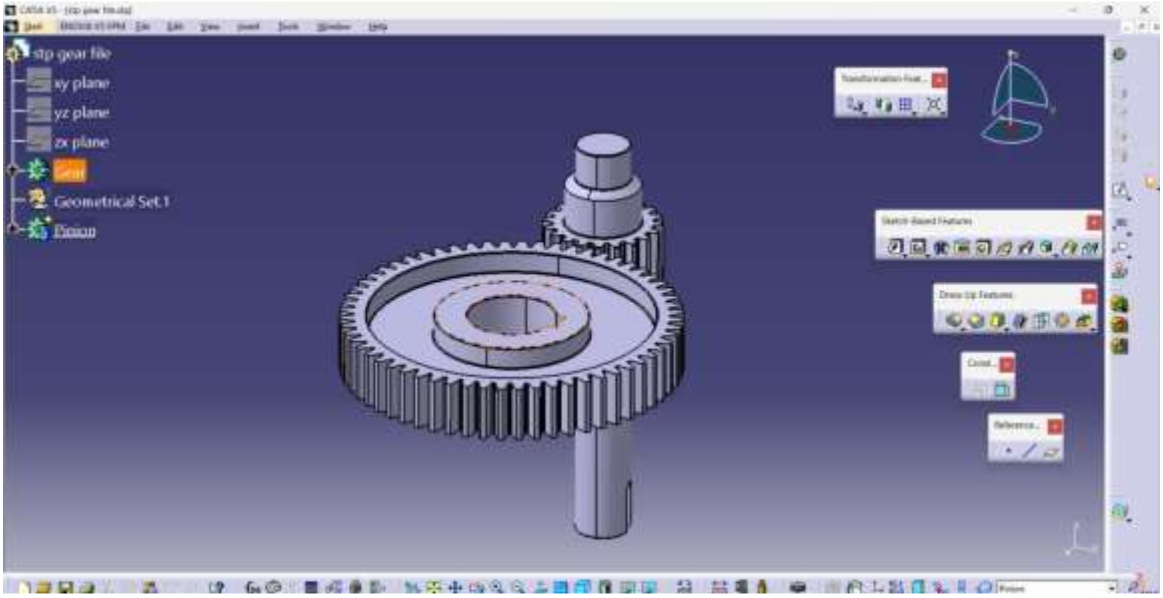


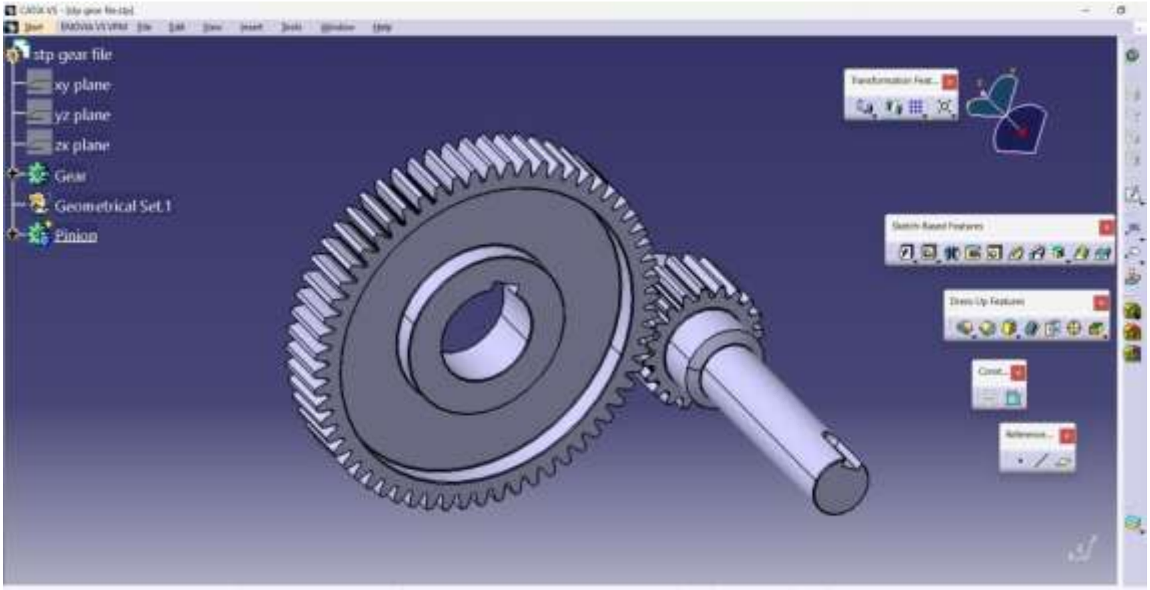
DRAFTINING VIE

DESIGN AND CONCLUSION

GEAR MODEL DESIGN IN CATIA

CATIA V5: A powerful 3D CAD software developed by Dassault Systèmes, used extensively in mechanical design, automotive, aerospace, and industrial equipment industries.

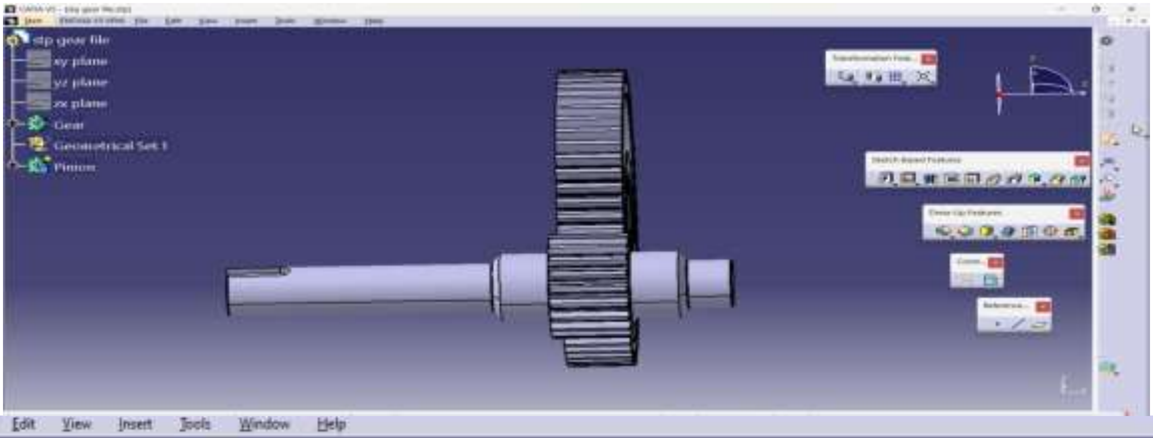


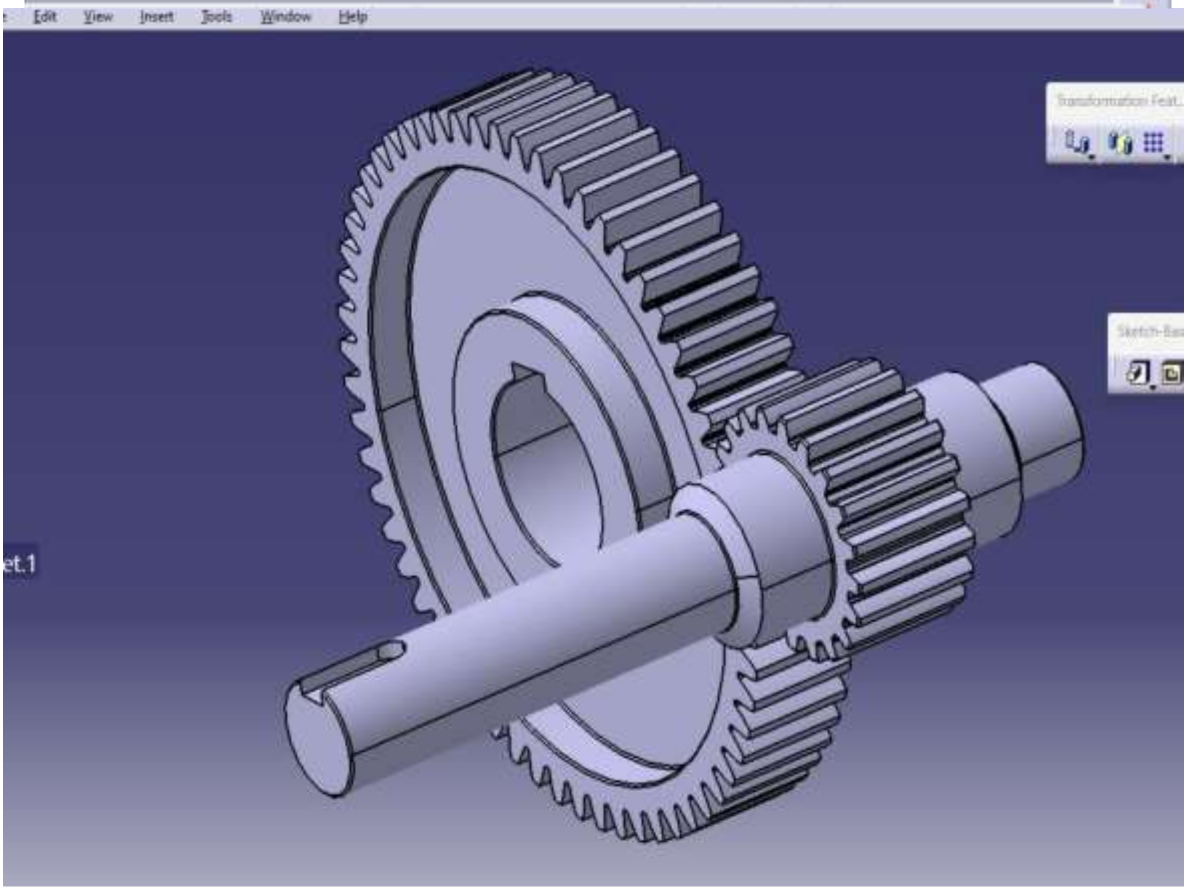


The main objective of this project is to design a 3D model of a gear using CATIA with proper parameters such as:

- Module
- Number of teeth
- Pressure angle

- Pitch circle diameter





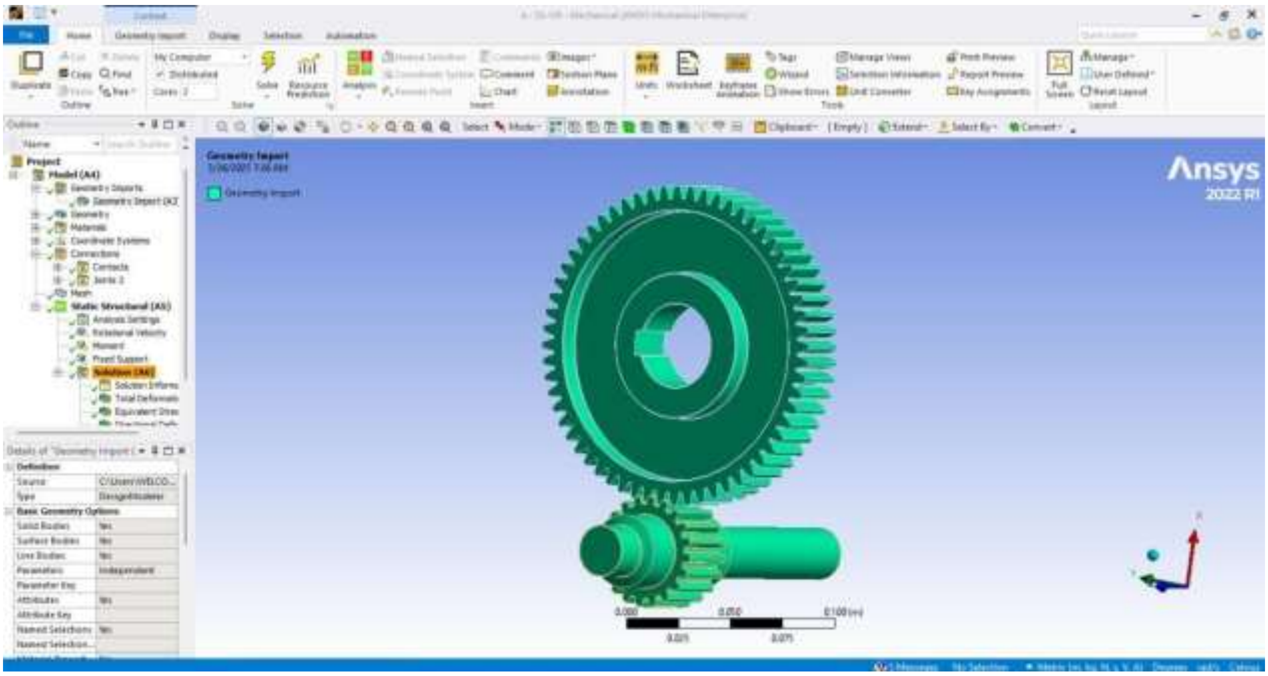
MODEL IMPORTING IN TO ANSYS 2022 R1

ANSYS 2022 R1 provides advanced tools for simulation, analysis, and product design. One of the first steps in any simulation workflow is the import of geometry or model data. Accurate import of geometry ensures a smooth simulation setup and reduces pre-processing time.

GEOMETRY IMPORT

Using ANSYS Workbench:

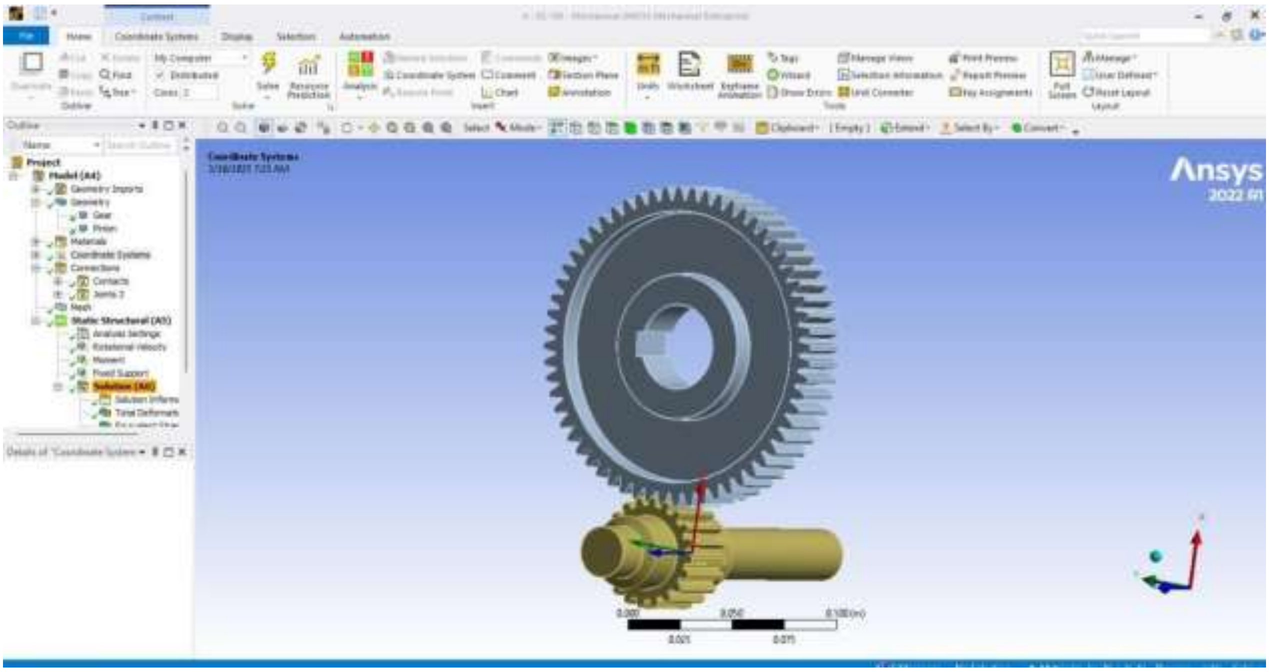
1. Open ANSYS Workbench.
2. Drag and drop the Geometry component into the Project Schematic.
3. Double-click Geometry to open Design Modeler or Space Claim.
4. Use File > Import External Geometry File to load the model.
5. Choose the appropriate file format and locate your model.



GEOMETRY IMPORT

COORDINATE SYSTEM:

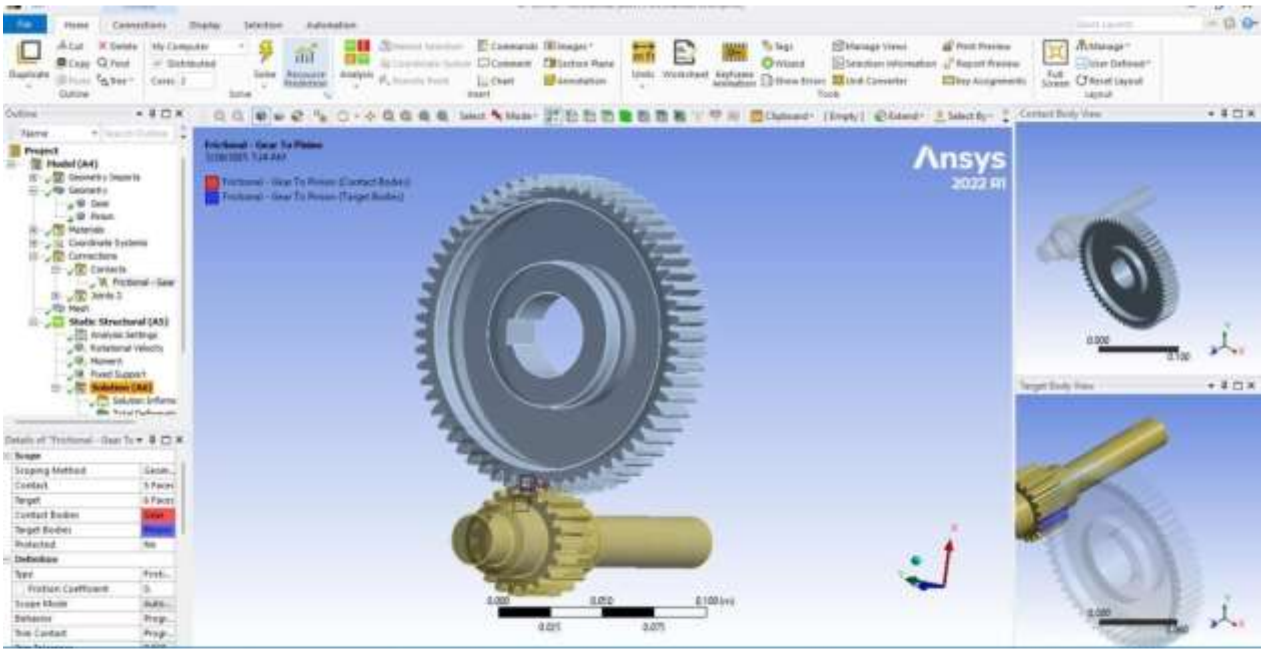
In ANSYS, coordinate systems play a crucial role in defining geometry, boundary conditions, loads, results interpretation, and more. Understanding and managing coordinate systems ensures accurate modeling and analysis, especially in complex simulations involving motion, symmetry, or non-Cartesian conditions



COORDINATE SYSTEM

FRICTIONAL -GEAR TO PINION:

In mechanical systems involving power transmission, gear and pinion pairs are commonly used to transfer motion and torque. Modeling the frictional contact between gear and pinion teeth is essential for accurate simulation of stresses, deformation, contact pressure, and power loss.
ANSYS 2022 R1 provides robust contact modeling tools that allow simulation of frictional behavior between mating parts like gears and pinions.



FRICTIONAL -GEAR TO PINION

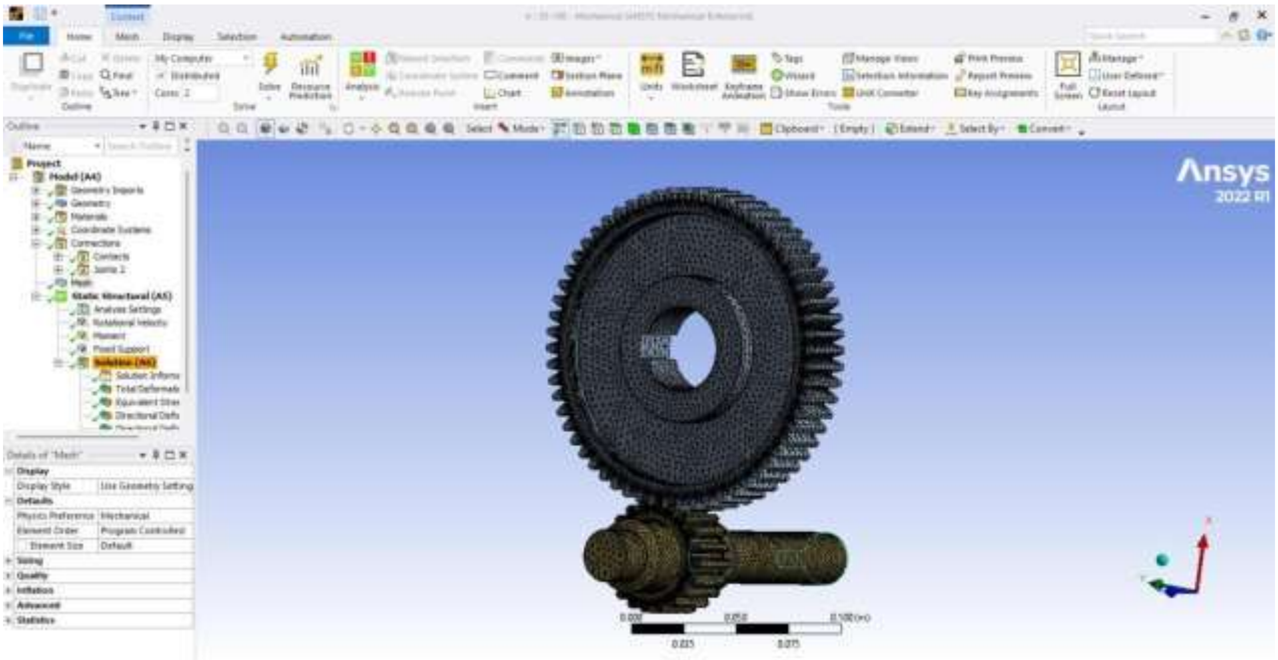
REVOLUTE -GROUND TO PINION:

In multibody dynamics and motion simulations, a revolute joint is used to allow rotational motion between two components while restricting all translational movement. When applied between the ground (fixed frame) and a pinion, the revolute joint simulates the pinion's rotation around a fixed axis—common in gear mechanisms and rotating shafts

REVOLUTE -GROUND TO PINION

MESHING:

Meshing is a crucial step in Finite Element Analysis (FEA). It involves dividing the geometry into smaller, discrete elements that the solver uses to approximate physical behavior under various conditions (such as stress, heat, or fluid flow). The quality of the mesh directly affects the accuracy, convergence, and efficiency of the simulation.



MESS

SELECTED MATERIAL FOR GEOMETRY MODEL :

In any engineering simulation, accurate material selection is critical for predicting real-world behavior. For this geometry model, Stainless Steel has been selected as the material due to its excellent mechanical properties, corrosion resistance, and widespread use in structural and mechanical applications.

Applications of Stainless Steel in Simulation:

- Structural Analysis: To evaluate stress, strain, and deformation under loads.
- Thermal Analysis: To study heat distribution and temperature effects.
- Fatigue and Life Estimation: For components subjected to cyclic loading.
- Contact Analysis: When used in assemblies (e.g., stainless steel gear systems).

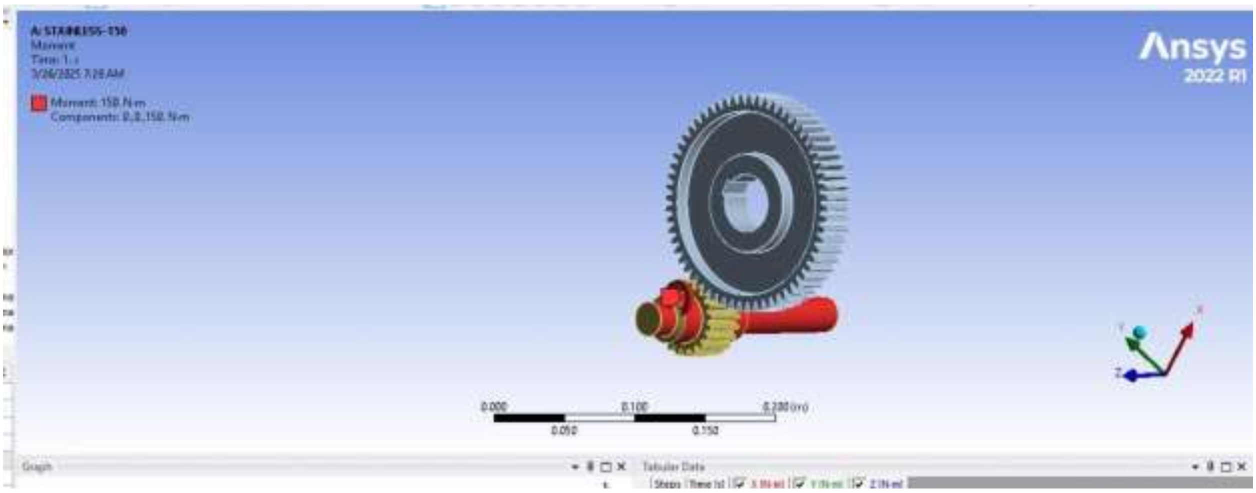
Stainless steel rpm 150

This simulation involves analyzing a stainless steel component subjected to rotational motion at 150 RPM. The purpose is to evaluate the structural response of the component under steady rotational loading. Key results studied include momentum, total deformation, equivalent stress, and directional deformations.

MOMENTEM:

Momentum in this context refers to the rotational momentum (angular momentum) generated by the rotating stainless steel component.

· The component exhibits a steady-state rotational momentum due to its constant angular velocity
· This is influenced by the mass moment of inertia and angular speed (150 RPM).
· The momentum remains stable over time, indicating dynamic balance in the system.



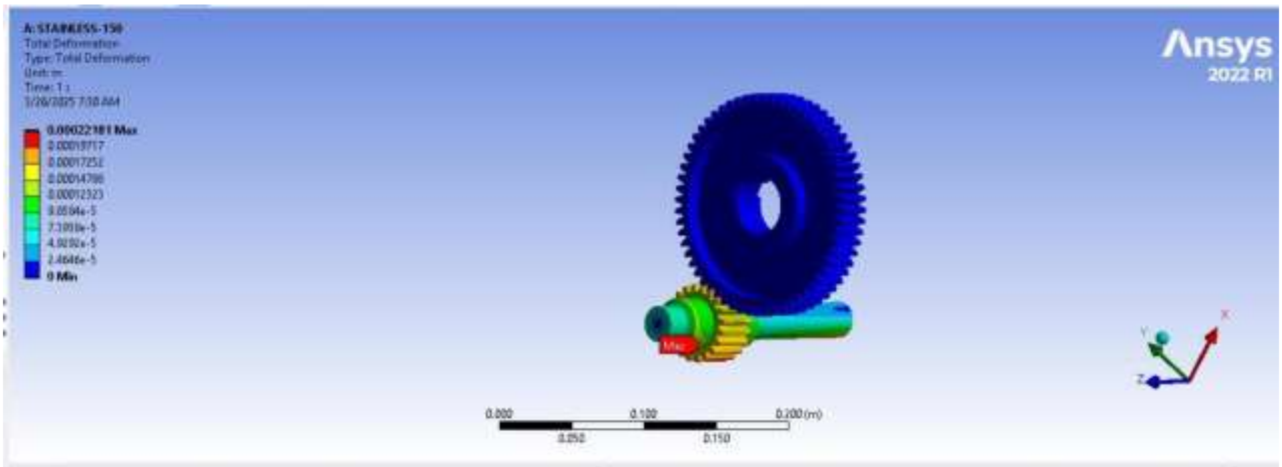
II. MOMENTEM

TOTAL DEFORMATION:

Total deformation refers to the overall displacement of any point on the component from its original position due to applied rotational loads.

- The highest deformation occurs at the outermost edges (farthest from the axis of rotation).
- Deformation increases with distance from the center due to centrifugal effects.

Deformation remains within elastic limits, indicating no permanent deformation for stainless steel under the current conditions.



TOTAL DEFORMATION

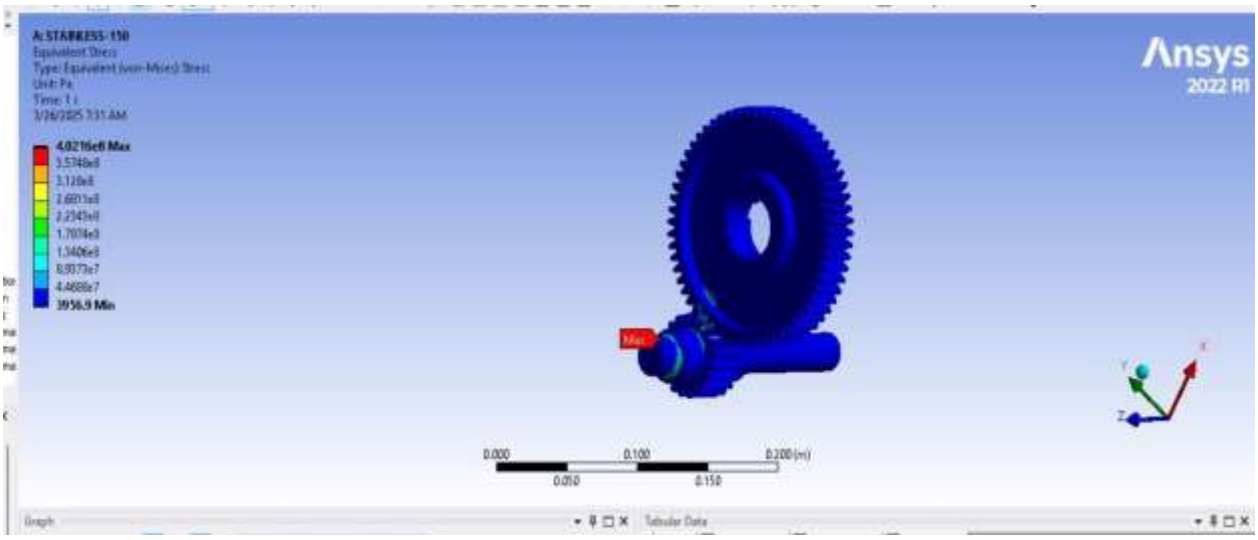
EQUIVALENT STRESS:

Equivalent (or von Mises) stress is a scalar value used to predict yielding of materials under complex loading conditions.

· High-stress concentrations were observed at sharp edges or contact areas.

· Maximum equivalent stress remains below the yield strength of stainless steel (~215 MPa).

· The component is safe and structurally sound under 150 RPM.

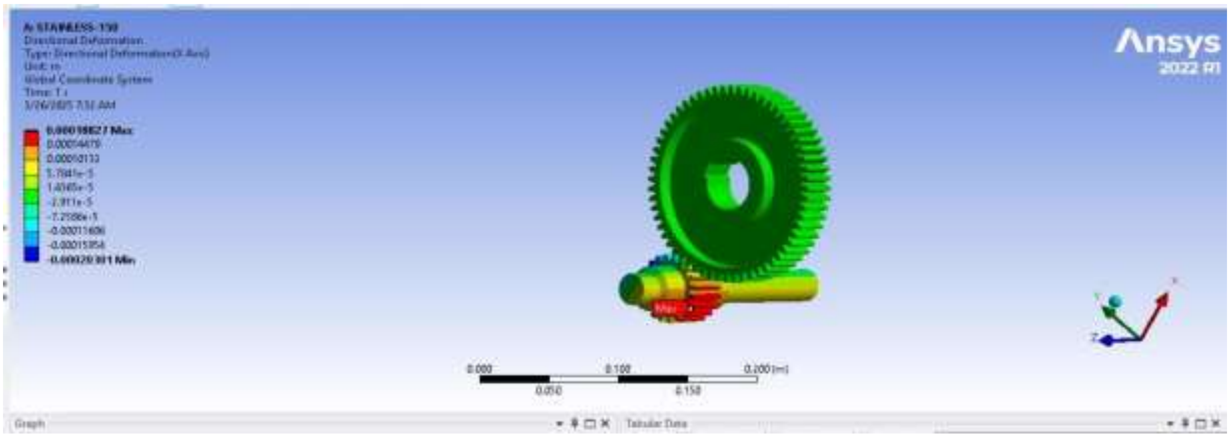


EQUIVALENT STRESS

DIRECTIONAL DEFORMATION

This is the displacement of the component in the X-direction due to rotational and inertial effects.

- Symmetric pattern due to balanced rotation.
- Slight deformation indicates minimal displacement in the X-direction.Highest deformation near the boundary or outer surface aligned with the X-axis.



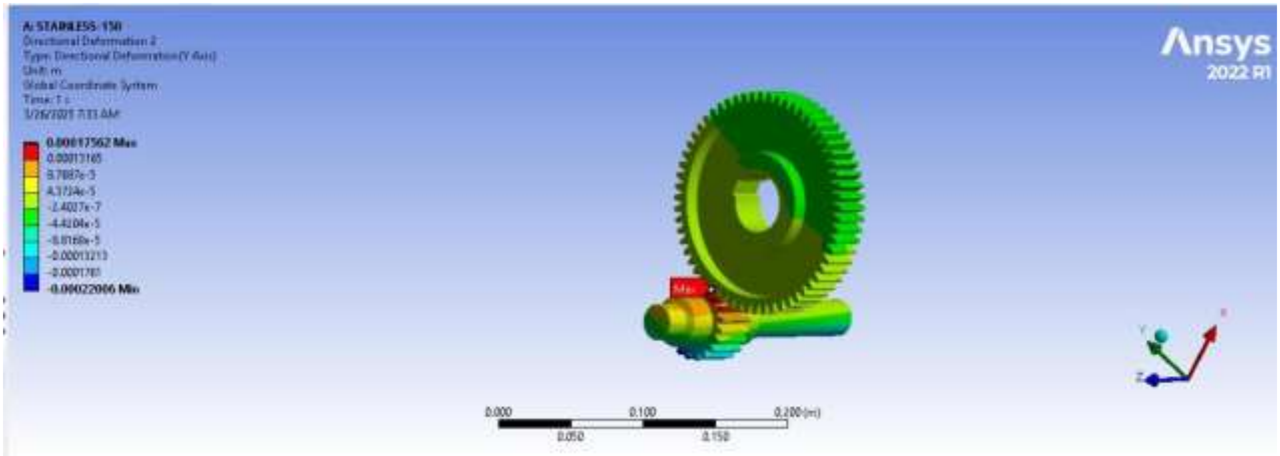
DIRECTIONAL DEFORMATION 1

DIRECTIONAL DEFORMATION 2

This represents the deformation in the Y-direction.

- Deformation pattern is similar to X-direction but dependent on geometry orientation.

- Shows how the material is displaced radially due to rotation.
- Useful for understanding lateral distortion or misalignment risks.



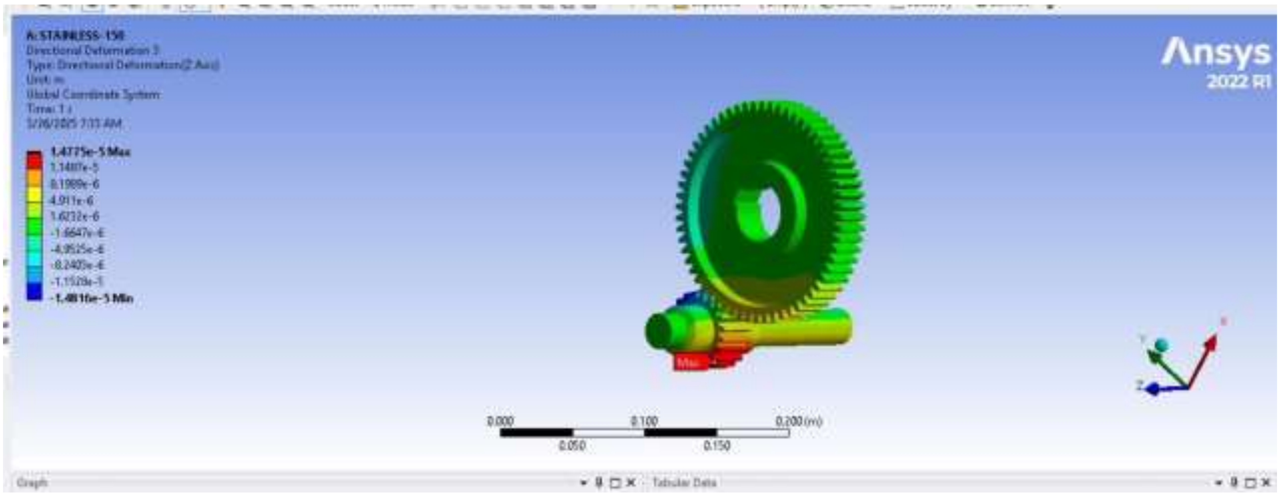
DIRECTIONAL DEFORMATION 2

DIRECTIONAL DEFORMATION 3

Z-directional deformation corresponds to movement along the axis of rotation.

- Minimum deformation in the Z-direction since the load is primarily radial.
- Important to check for axial elongation or compression.

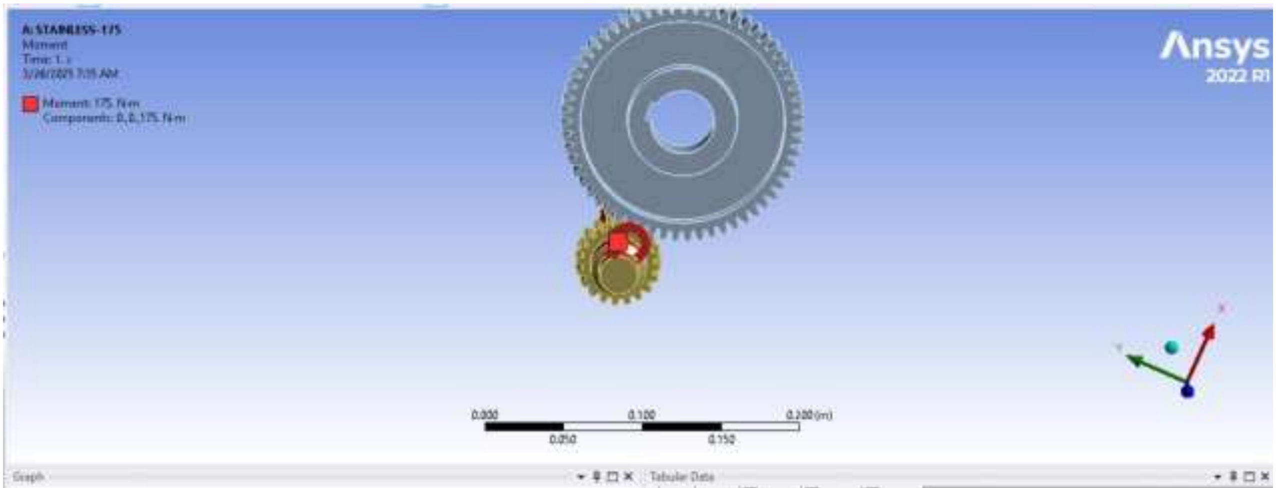
Confirms structural stability along the axis under 150 RPM.



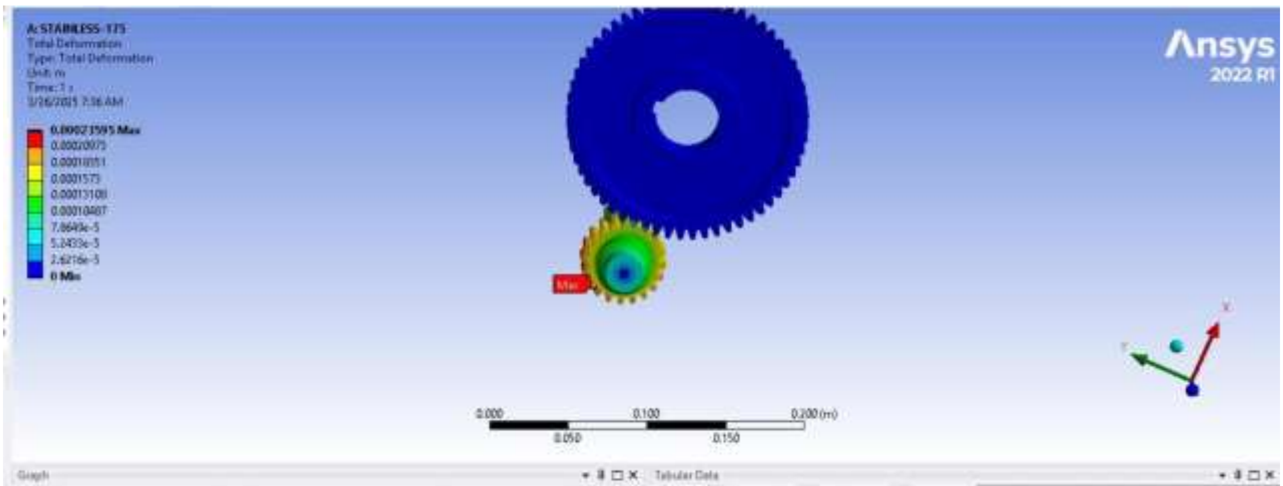
DIRECTIONAL DEFORMATION 3

RESULTS FOR STAINLESS STEEL 150 RPM							
stainless steel	150 rpm	sno	total deformation	equivalent stress	directional deformation	directional deformation	directional deformation
		min	0	3835.7	-0.0002	-0.00022	-1.4372
		max	0.000219	4.08E+08	0.000186	0.00017	1.43E-05

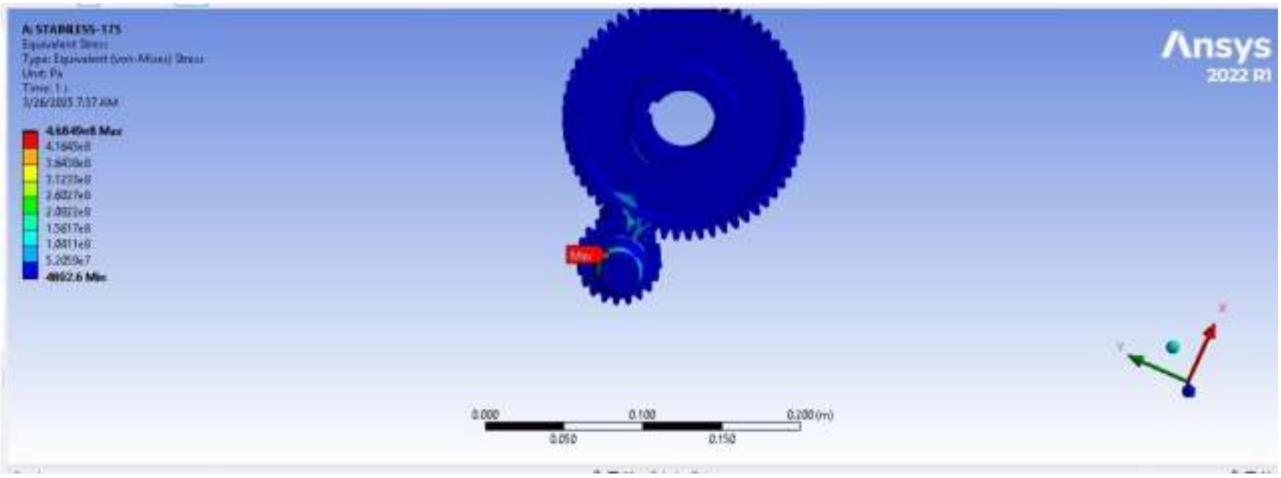
STAINLESS STEEL 175 RPM



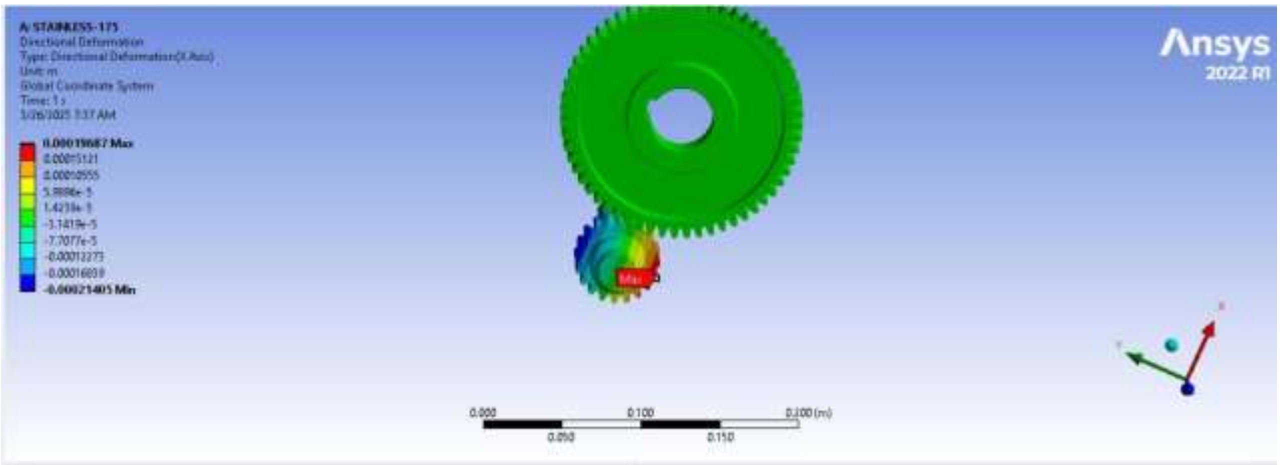
MOMENTUM



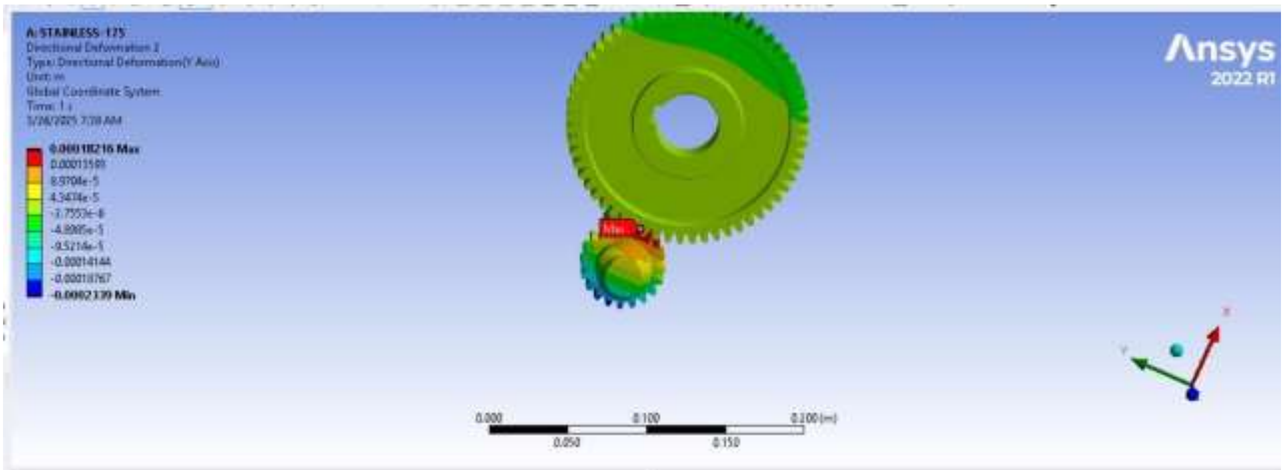
TOTAL DEFORMATION



EQUIVALENT STRESS

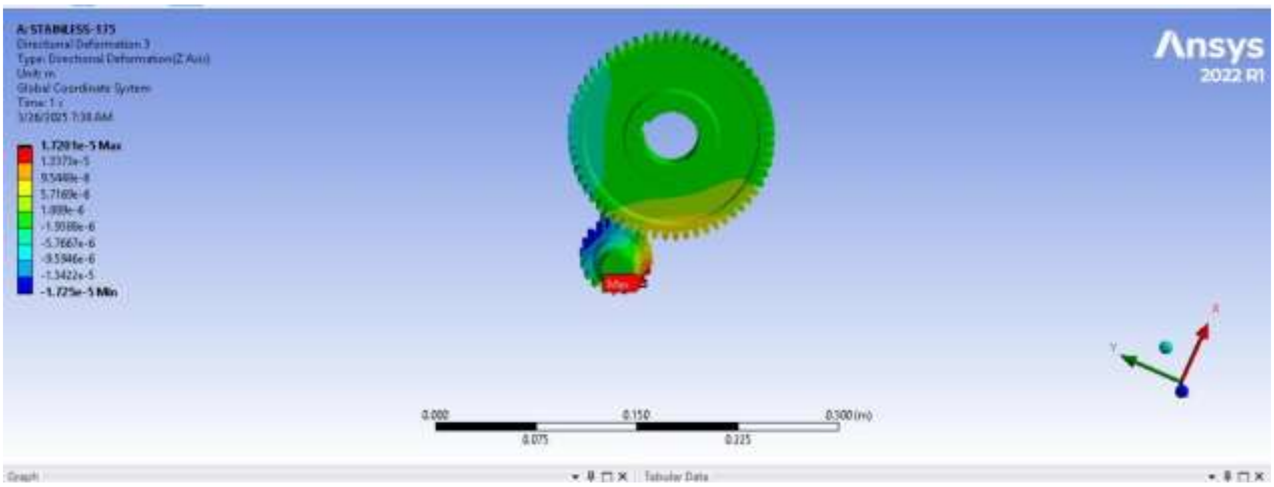


DIRECTIONAL DEFORMATION 1



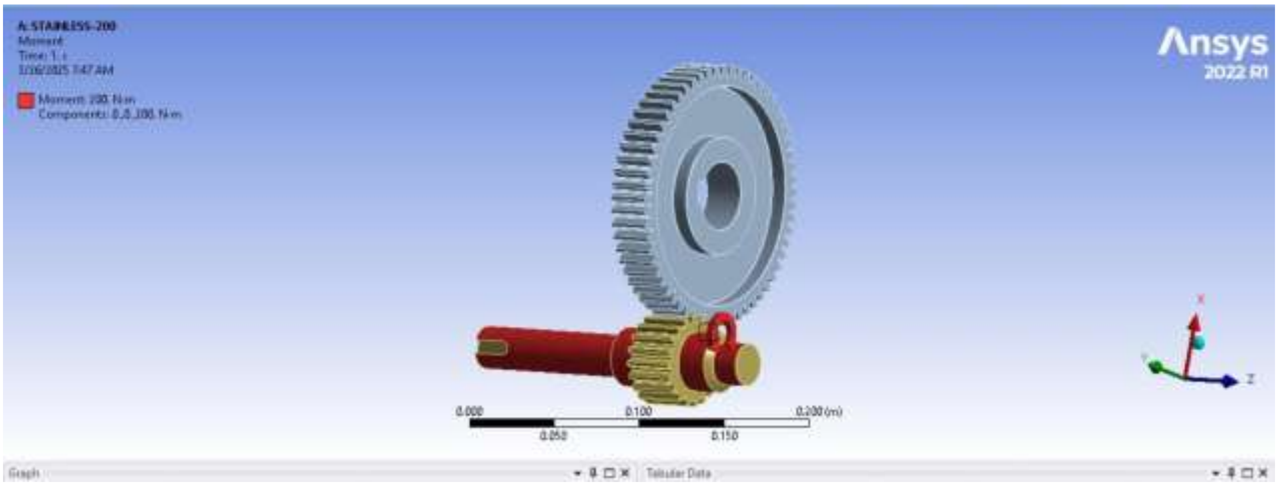
DIRECTIONAL DEFORMATION 2

VOLUME: 09 ISSUE: 04 | APRIL - 2025 SJIF RATING: 8.586 ISSN: 2582-3930

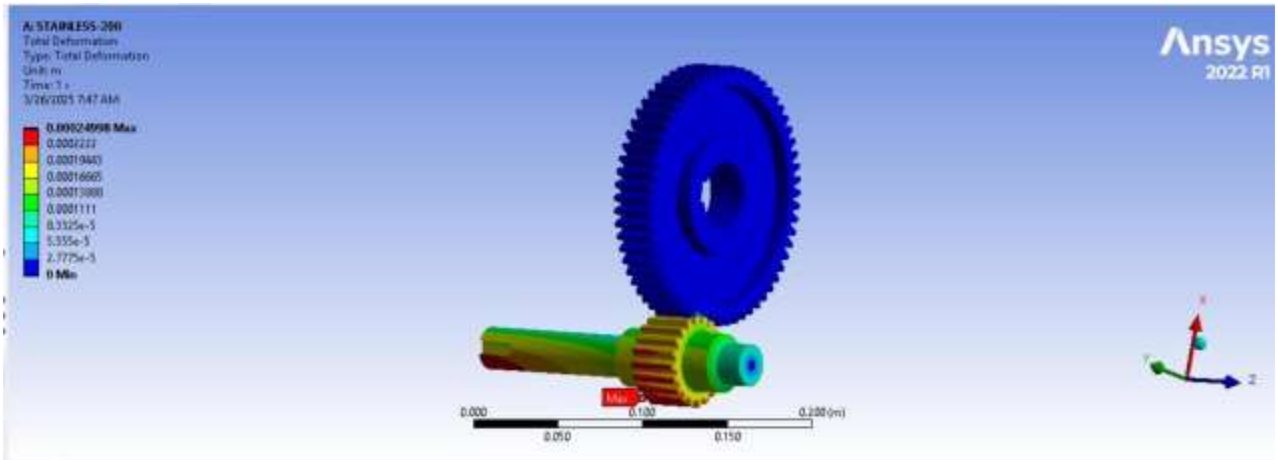


RESULTS FOR STAINLESS STEEL 175 RPM							
stainless steel	175 rpm	sno	total deformation	equivalent stress	directional deformation	directional deformation 2	directional deformation 3
		min	0	4802.6	-0.00021405	-0.0002339	-1.73E-05
		max	0.00023595	4.68E+12	0.00019687	0.00018216	1.72E-05

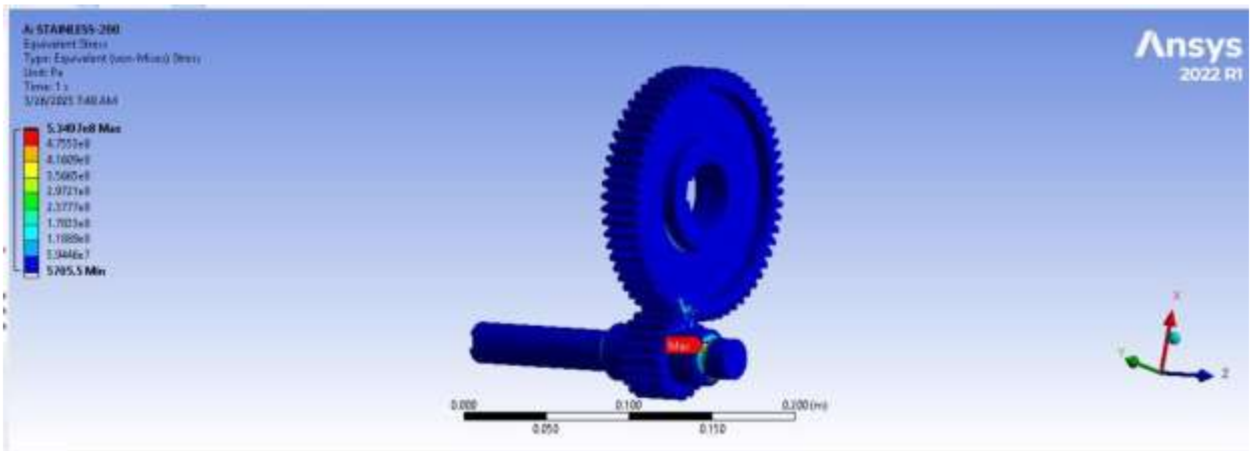
STAINLESS STEEL 200 RPM



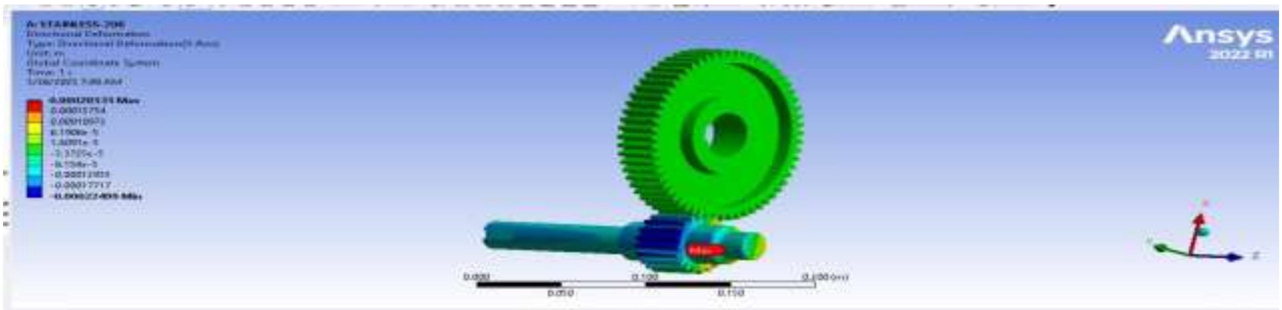
MOMENTUM



TOTAL DEFORMATION

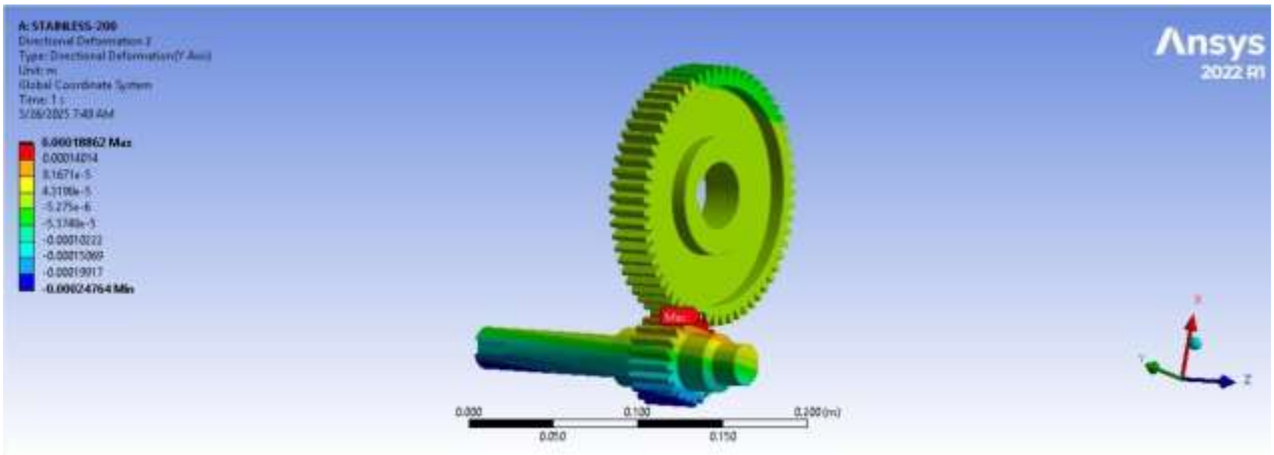


EQUIVALENT STRESS

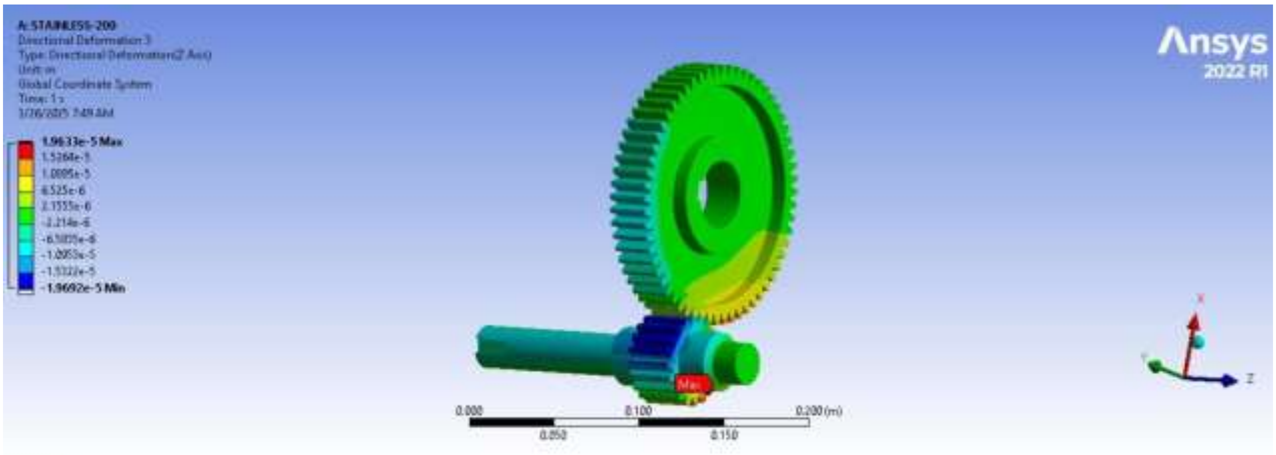


DIRECTIONAL DEFORMATION 1

VOLUME: 09 ISSUE: 04 | APRIL - 2025 SJIF RATING: 8.586 ISSN: 2582-3930



DIRECTIONAL DEFORMATION 2



DIRECTIONAL DEFORMATION 3

RESULTS FOR STAINLESS STEEL 200 RPM							
stainless steel	200 rpm	sno	total deformation	equivalent stress	directional deformation	directional deformation 2	directional deformation 3
		min	0	5705.5	-0.00022499	-0.00024764	-1.97E-05
		max	0.00024998	5.35E+08	0.00020535	0.00018862	1.96E-05

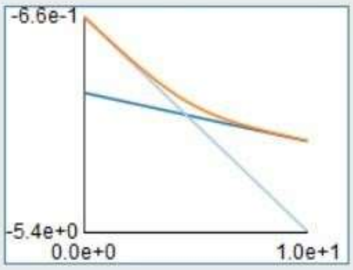
STRUCTURAL STEEL

It's a steel that's shaped into specific cross sections (like I-beams, H-beams, angles, channels, etc.) and is designed to carry loads in construction. It's known for being:

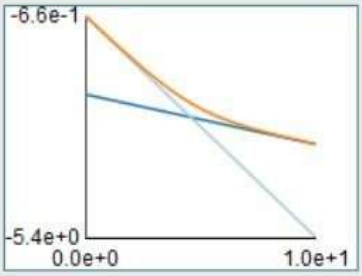
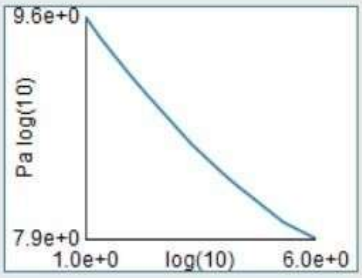
- Strong
- Ductile (can bend without breaking)
- Weldable
- Recyclable

Structural Steel

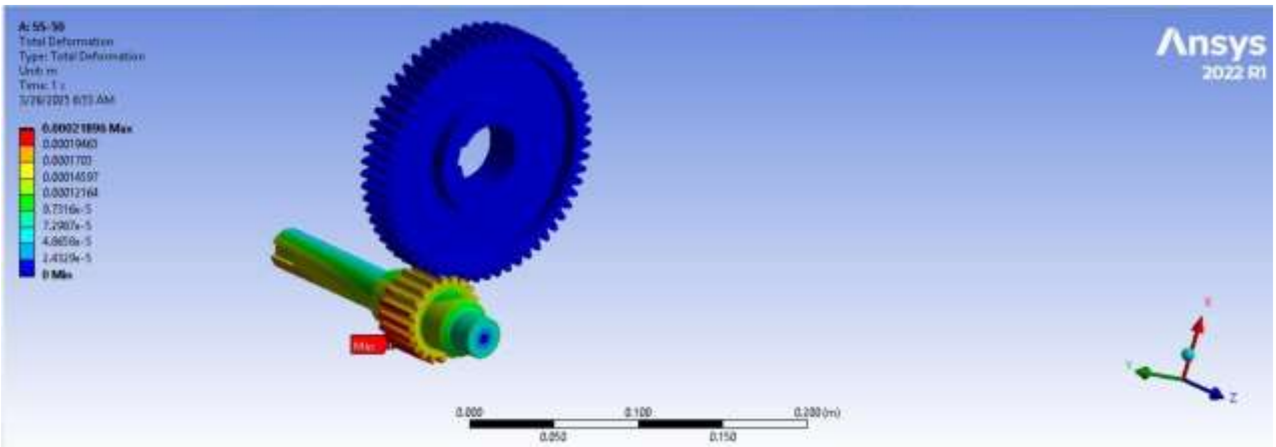
Fatigue Data at zero mean stress comes from 1998 ASME BPV Code, Section 8, Div 2, Table 5-110.1

Density	7850 kg/m³
Structural	
Isotropic Elasticity	
Derive from	Young's Modulus and Poisson's Ratio
Young's Modulus	2e+11 Pa
Poisson's Ratio	0.3
Bulk Modulus	1.6667e+11 Pa
Shear Modulus	7.6923e+10 Pa
Isotropic Secant Coefficient of Thermal Expansion	1.2e-05 1/°C
Compressive Ultimate Strength	0 Pa
Compressive Yield Strength	2.5e+08 Pa
Strain-Life Parameters	

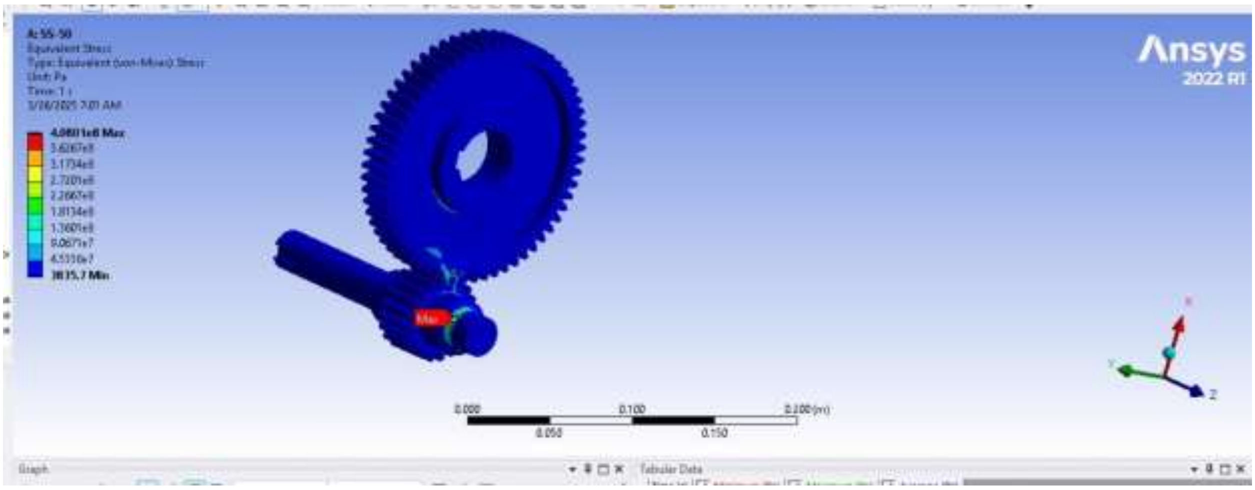
Structural Steel

Strain-Life Parameters	
S-N Curve	
Tensile Ultimate Strength	4.6e+08 Pa
Tensile Yield Strength	2.5e+08 Pa
Thermal	
Isotropic Thermal Conductivity	60.5 W/m·°C
Specific Heat Constant Pressure	434 J/kg·°C
Electric	
Isotropic Resistivity	1.7e-07 ohm·m

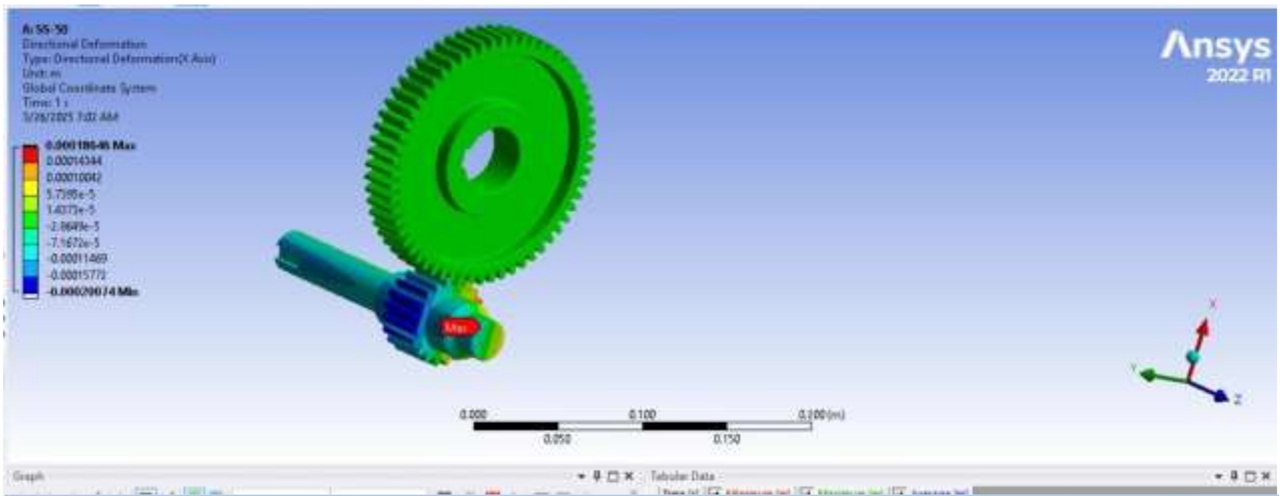
STRUCTURAL STEEL 50 RPM

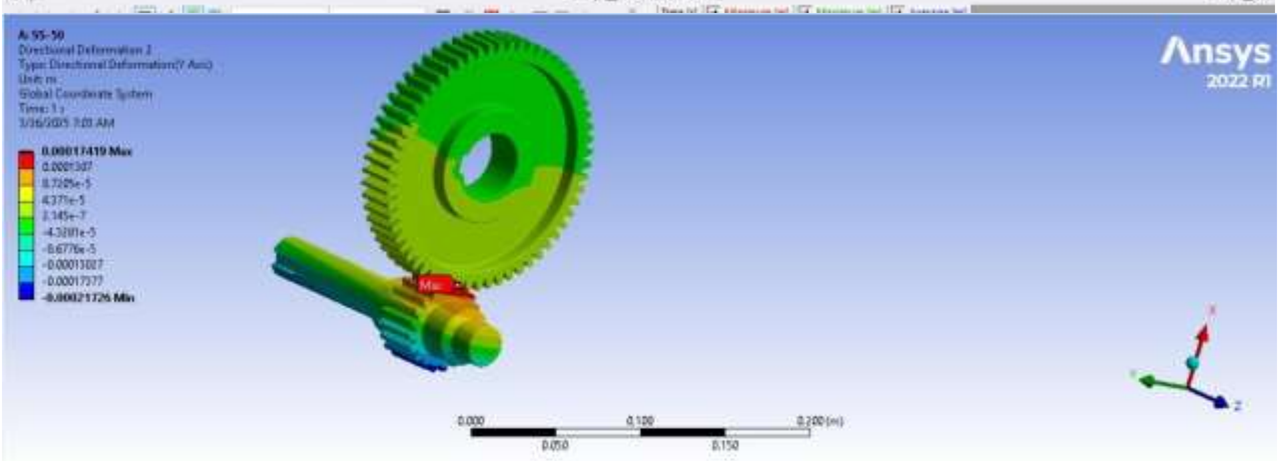


TOTAL DEFORMATION

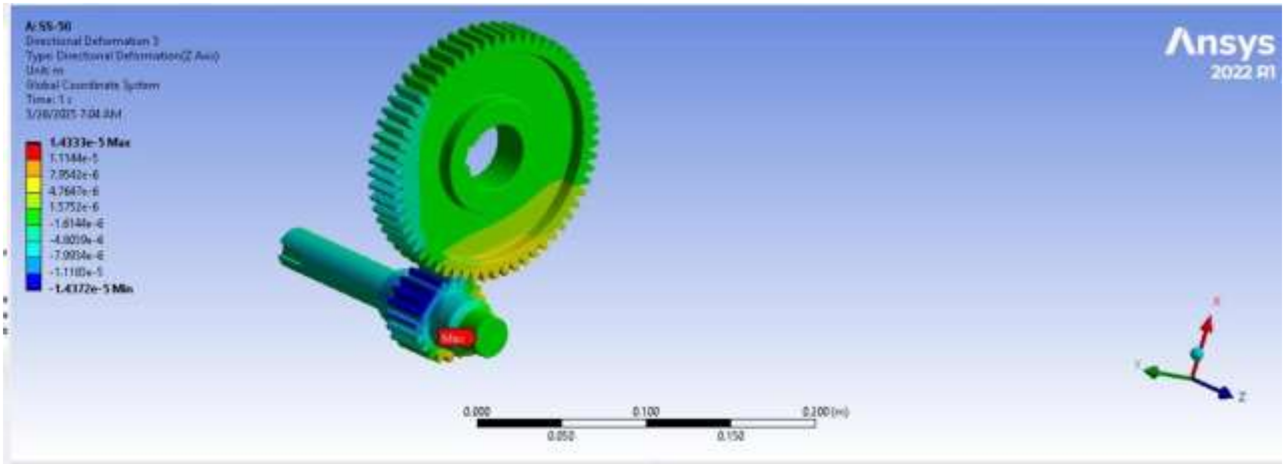


EQUIVALENT STRESS





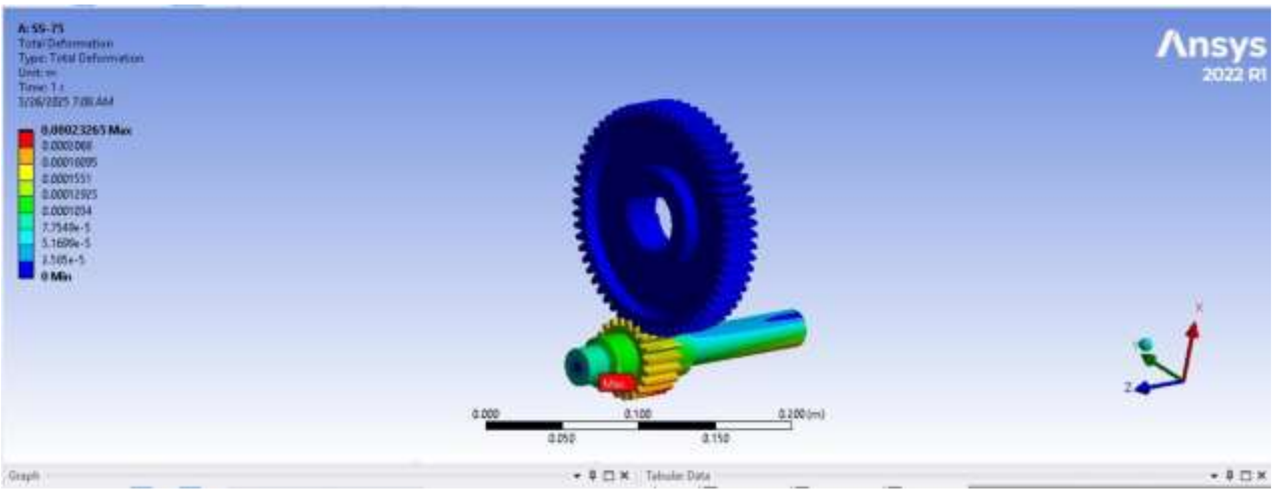
DIRECTIONAL DEFORMATION2



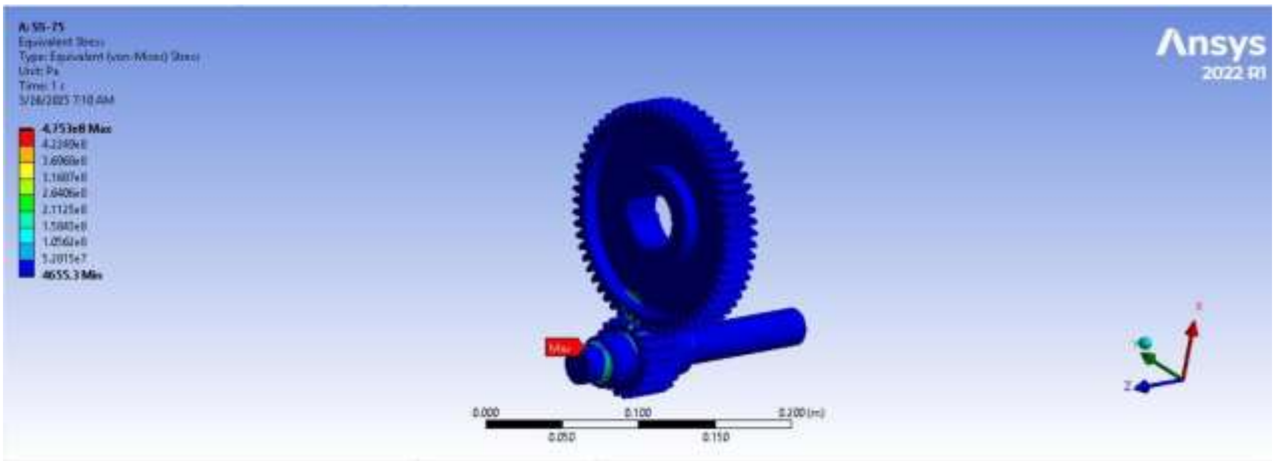
DIRECTIONAL DEFORMATION 3

RESULTS FOR STRUCTURAL STEEL 50 RPM							
structural steel	50rpm	sno	total deformation	equivalent stress	directional deformation	directional deformation 2	directional deformation 3
		min	0	3835.7	-0.00020074	-0.00021726	-1.44E-05
		max	0.00021896	4.08E+08	0.00018646	0.00017419	1.43E-05

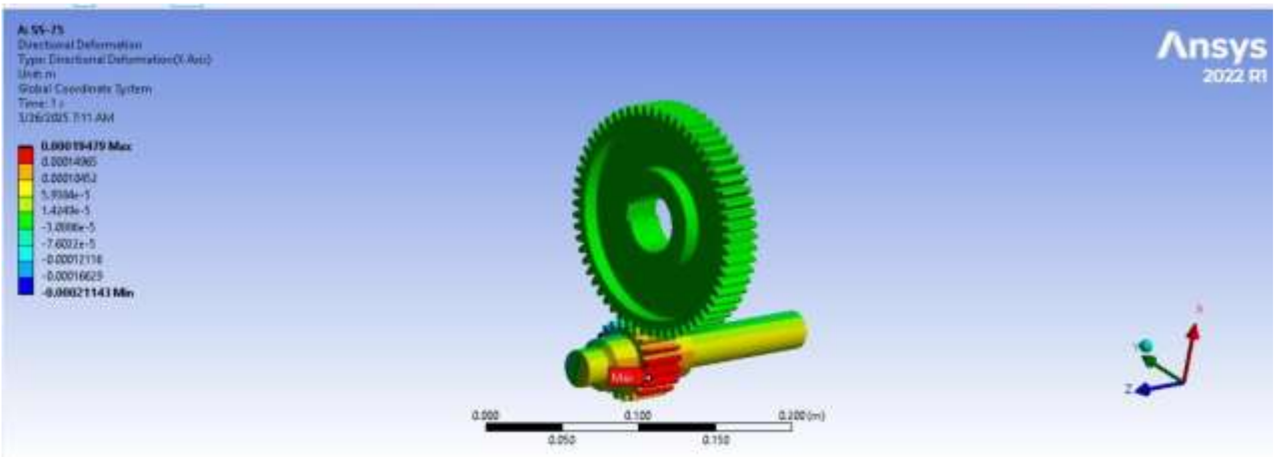
STRUCTURAL STEEL 170 RPM



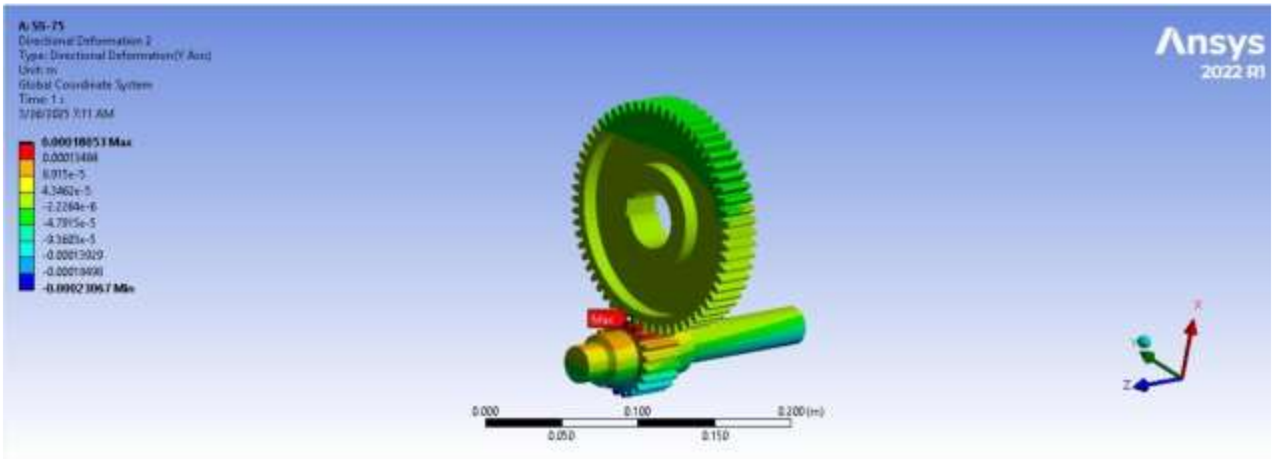
TOTAL DEFORMATION



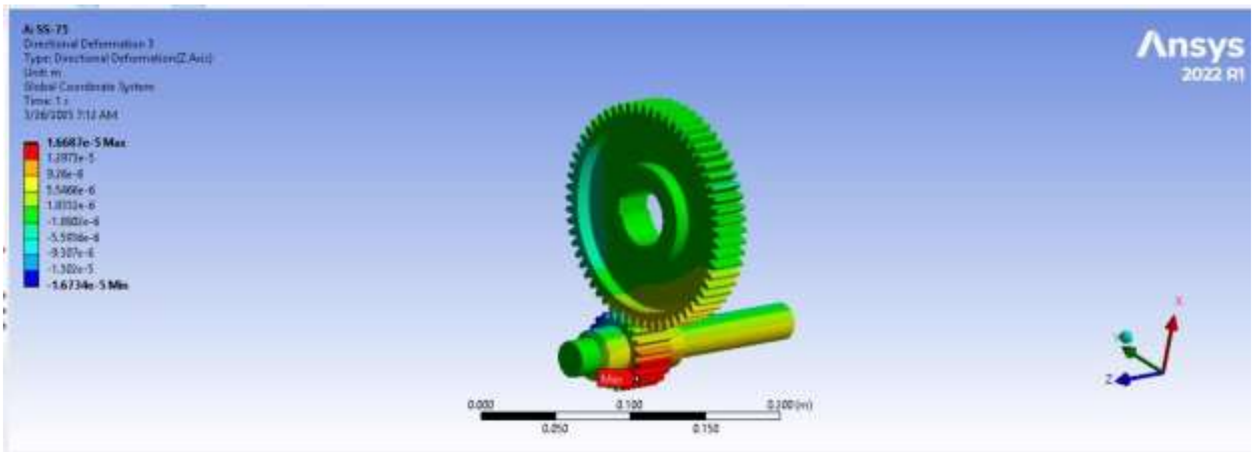
EQUIVALENT STRESS



Directional deformation1



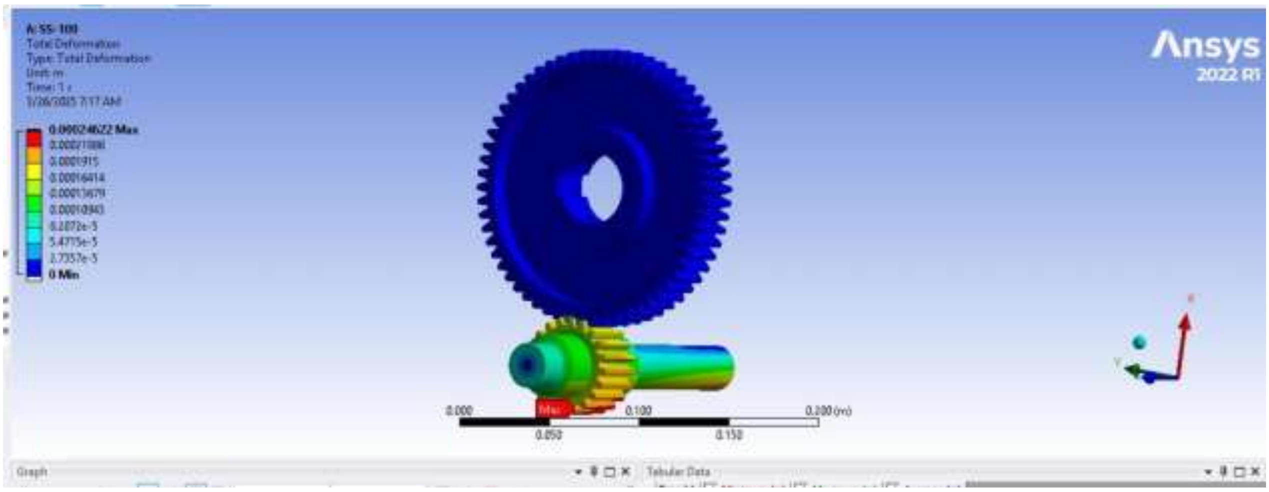
DIRECTIONAL DEFORMATION 2



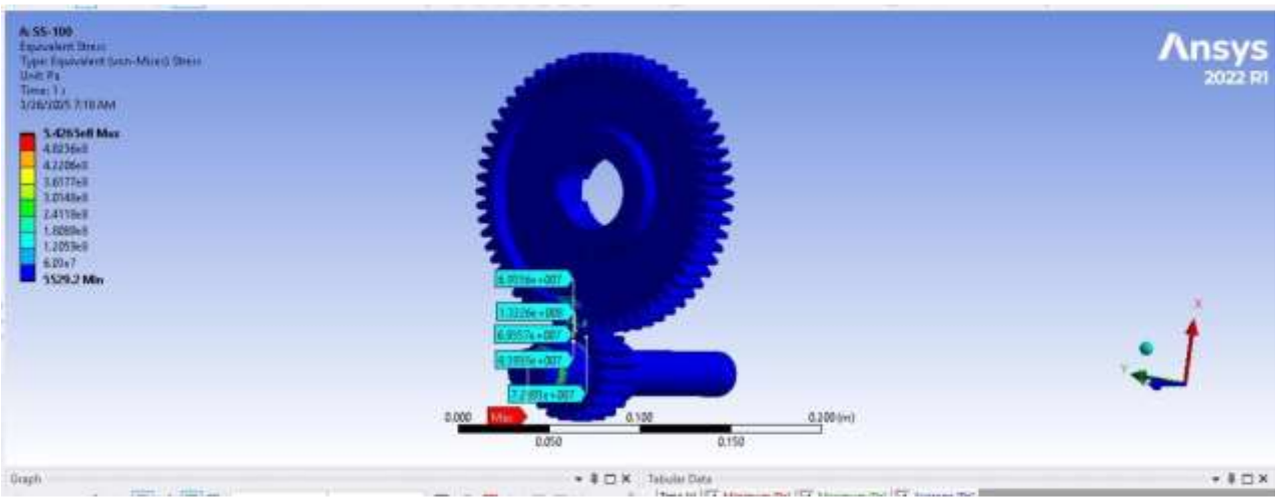
DIRECTIONAL DEFORMATION 3

RESULTS FOR STRUCTURAL STEEL 175 RPM							
structural steel	170 rpm	sno	total deformation	equivalent stress	directional deformation	directional deformation 2	directional deformation 3
		min	0	4655.3	-0.00021143	-0.00023067	-1.67E-05
		max	0.00023265	4.75E+08	0.00019479	0.00018053	1.67E-05

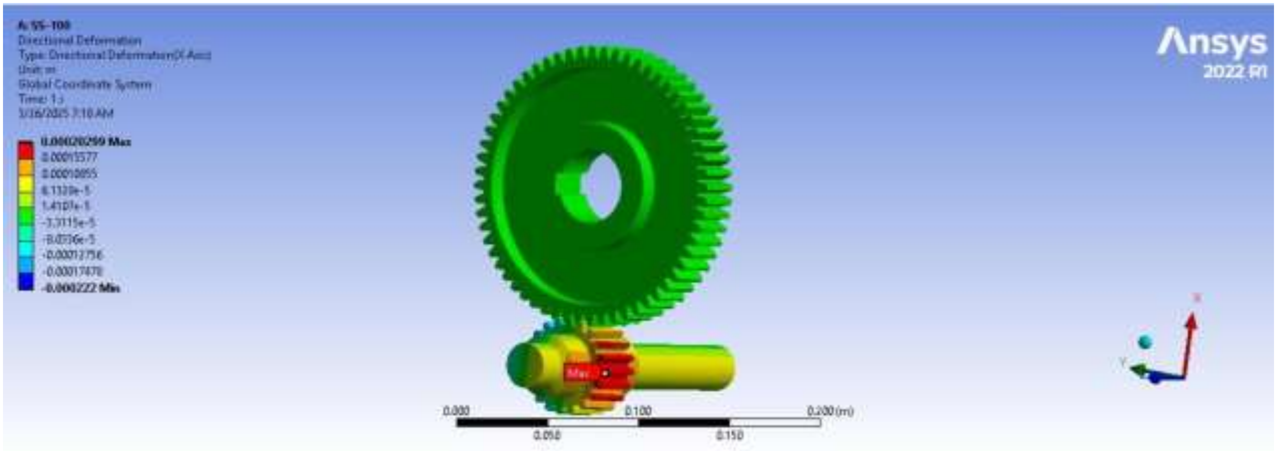
STRUCTURAL STEEL 100 RPM



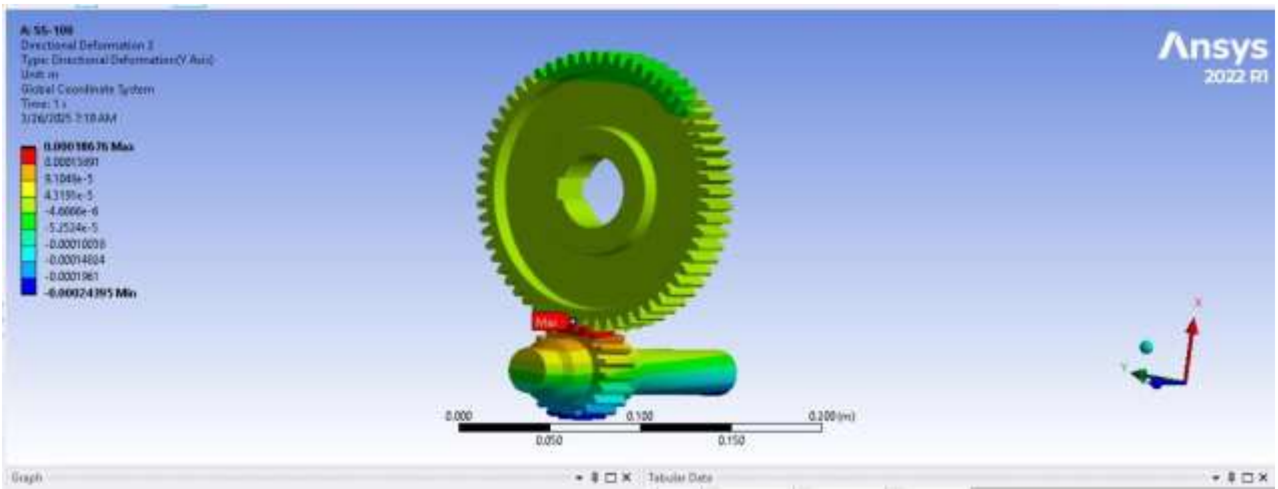
TOTAL DEFORMATION



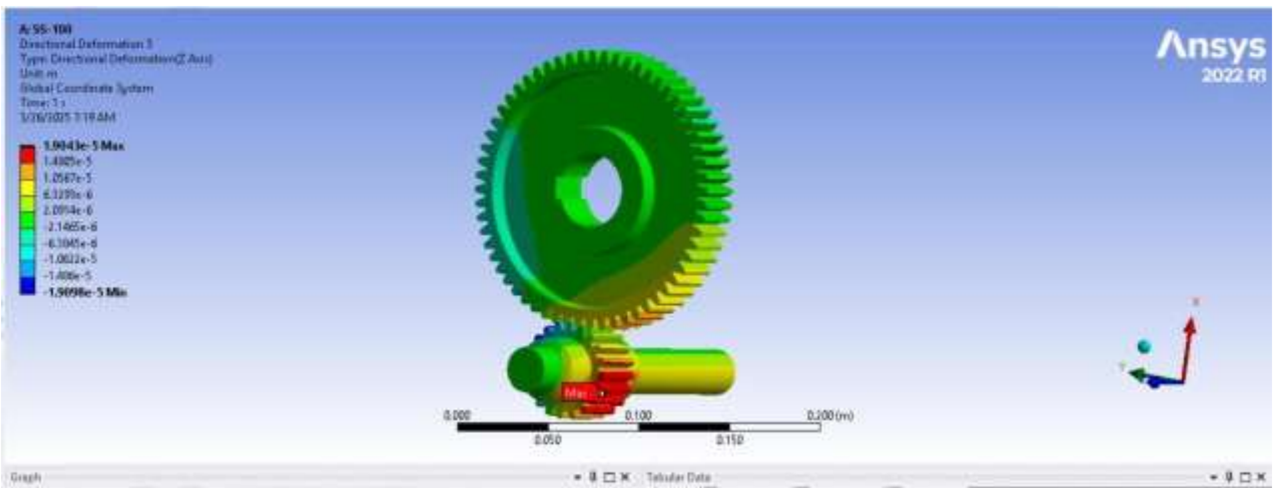
EQUIVALENT STRESS



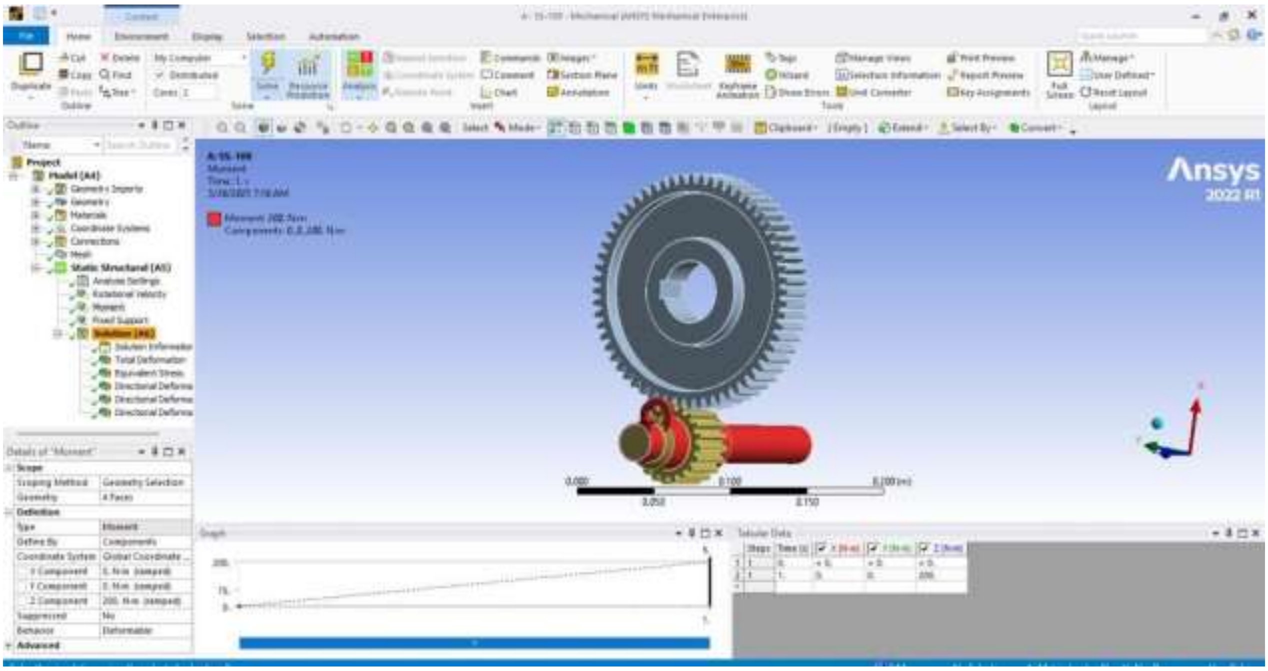
DIRECTIONAL DEFORMATION1



DIRECTIONAL DEFORMATION2



DIRECTIONAL DEFORMATION 3



MOMENTUM

RESULTS FOR STRUCTURAL STELL 100 RPM							
structural steel	100 rpm	sno	total deformation	equivalent stress	directional deformation	directional deformation 2	directional deformation 3
		min	0	5529.2	-0.000222	-0.00024395	-1.91E-05
		max	0.00024622	5.43E+08	0.00020299	0.00018676	1.90E-05

In the present study, a spur gear pair was modeled and analyzed using ANSYS 2021 to evaluate the stress distribution and deformation under different torque conditions. The pinion acts as the driving member rotating at 2500 RPM, while the gear is the driven member. A torque of 50 Nm, 175 Nm, and 200 Nm was applied on the pinion. The gear system has a gear ratio of 3, with the pinion diameter being 50 mm and the gear diameter being 150 mm. The face width of both gears is 25 mm.

A. Power Transmitted at Different Torque Levels

Torque (Nm)	Speed (RPM)	Power Transmitted (kW)
50	2500	13
175	2500	39
200	2500	52

B. Equivalent Von Mises Stress

Torque (Nm)	Max Von Mises Stress (MPa)	Location	Remarks
50	137	Pinion shaft (at step)	Stress concentration due to geometry step; can be reduced with fillet design
175	475	Pinion shaft (at step)	Same as above
200	542	Pinion shaft (at step)	Same as above
All Cases	≤132	Gear-pinion tooth contact region	Within safe limits for structural and stainless steel

Note: The high stresses at the shaft step are considered negligible from a design safety perspective as they occur due to geometric discontinuity. These can be mitigated by introducing suitable fillets or radii.

Contact Region Stress

Torque (Nm)	Stress at Gear-Teeth Contact (MPa)	Material Limit	Safety Status
50	<100	>250 MPa	Safe
175	<100	>250 MPa	Safe
200	132	>250 MPa	Safe

C.

Torque (Nm)	Stress at Gear-Teeth Contact (MPa)	Material Limit	Safety Status
50	<100	>250 MPa	Safe
175	<100	>250 MPa	Safe
200	132	>250 MPa	Safe

D. Total Deformation

E.

Torque (Nm)	Total Deformation (mm)	Remarks
50	0.00013	Very small, negligible in practical use
200	0.00026	Still within acceptable tolerance

F. Directional Deformation

G.

Torque (Nm)	Directional Deformation (mm)	Direction	Remarks
50	0.000148	X/Y/Z	Very minimal
200	0.000186	X/Y/Z	Within safe operational limits

H. Final Observations and Conclusion

I.

• The maximum equivalent von Mises stress in the gear-pinion contact zone does not exceed 132 MPa, which is well below the allowable stress for both structural steel and stainless steel.

• Peak stress at the pinion shaft step is attributed to geometric discontinuity. This can be addressed with design improvements (fillets).

• Both total and directional deformations are extremely small and do not affect the performance or integrity of the gear set.

• The gear set is capable of transmitting power efficiently at various torque levels up to 52 kW, with safe stress and deformation levels.

Conclusion:

J.

The spur gear pair design considered in this study is mechanically safe and efficient. All stresses and deformations fall within acceptable limits for the materials used. The modeling in CATIA and analysis in ANSYS confirm that the system performs reliably under the applied loading conditions. This validates the design for real-world applications in mechanical power transmission systems

REFERENCES

1. Darle W. Dudley(1954),Practical Gear Design, McGraw-Hill Book Company.
2. Khurmi Gupta RS(2000),"MachineDesign",Khanna Publication.
3. Khurmi RS(1997),"Theory of Machine", Khanna Publication.
4. Machine Design DataBook(2003), PSG Publication.
5. .Rattan SS(1998),"Theory of Machines",Dhanpat Rai Publication.
6. Romlay F RM(2008), "Modeling of a Surface Contact Stress for Spur Gear Mechanism Using Static and Transient Finite Element Method", Journal of Structural Durability & Health Monitoring (SDHM), Vol. 4, No. 1, Tech Science Press.
7. Shanavas S (2013), "Stress Analysis of Composite Spur Gear", International Journal of Engineering Research & Technology (IJERT), ISSN: 2278-0181
8. Shinde SP,Nikam A and Mulla TS(2012), "Static Analysis of Spur Gear Using Finite Element Analysis", IOSR Journal of Mechanical and Civil Engineering(IOSR-JMCE)