Experimental Investigation And FEA Of Wear In Gear At Torque Loading Conditions

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Abstract- *Gear is machine element, which used to transmit the power in the modern engineering era. They differ from small in size used for the watches and the large gears used in all types of trains, bridge lifting mechanism, and industrial application with an outspread velocity ratio. The gears are a main constituent element that provides necessary support for mechanism in many machines like automobile sector, civil sector and various engineering application like mills, industries, hoisting and transmitting machinery and marine engines, etc.*

Gear element is used usually to transfer circular motion by variation of torque loadings. During their process gear teeth are undergo various contact and bending stresses. Gear imparts high pressure on pinion resulting in wear causing rough surfaces, pitting etc., this location is prone to crack initiation causing fatigue failure. In this project, work investigation of stresses at a specified torque loading using Experimental and finite element technique. Design and CAD Modeling of an existing gear pair using CATIA V5 software. Discretization (Meshing) of modeled spur gear done by using ANSYS software. Torque loading done with the help of ANSYS software to analyze stresses. Setup is prepared to mount gears for replicating torque loading. Gear surface analyzed to measure wear caused due to torque loading. Comparative analysis of spur gear between FEA and Experimental results. Conclusion and further future is proposed.

Keywords- Spur Gear, Wear rate, Gears, Wear investigation, experimental setup, conclusion, future scope.

I. INTRODUCTION

A gear is a gyrate machine element which having teeth, those teeth mesh with another toothed part gyrate for transmitting power or torque. Geared system can change the torque, speed and direction of a power source. Gears are mostly produce a change in torque value, creating mechanical advantages, gear ratios, and it considered simple machine. The teeth of two meshing gears having nevertheless shape. Continuously mate gears, working in a series that called a transmission. The gears continually engaging with other toothed gears, known as rack, that producing translation motion instead of rotational motion. The gear elements, which transmit the motion from one shaft to other, are equivalent to the wheels in a crossed, belt and pulley drive. The main advantages of gears that avert slippage. When one gear is mesh with other in that one is bigger than the other in that mechanical advantages achieve, with the torques and rotational speeds, of the two gears differing in part to their diameters. In transmissions with several gear ratios like a motorcycles, bicycles, and cars. For gear failure mode occurring tooth bending fatigue, surface scoring and wear, contact fatigue. There are two different types of gear teeth devastation occur in gears under several freight due to fatigue known as tooth breakage in a root and teeth damage, teeth breakage of teeth is clearly worst damage case, the gear hampered operating condition or destroyed, because of this, the localized stresses in a tooth should be conceptually studied in all gear application. The fatigue process develops to tooth breakage is divers in to two types like crack propagation period and crack imitation; However, the crack imitation period commonly account for the most service life of gear, particularly in high cycle fatigue. The beginning crack can be induced due to diver's reasons. The many common reasons will be material defect, defect due to thermal or mechanical and material fatigue to short term overload. The first crack then prorogates under impetus loading until little critical length is extend, that a complete tooth breakage occurs. The service life of a gears to which a crack in the tooth root can be determined practically or numerically with the help of finite element method. Wear is associated with one surface rub over another and the removal of material and distortion of material on a veneer as a result of mechanical action of the opposite surface that's why consequently tooth of gear gets weakened. Scoring is spur gear teeth surface fatigue failure. Due to misalignment of gear shaft, selection of wrong viscosity of the lubricant, and contact stresses growing on the contacting surface that causes fatigue strength of geometry of the contacting surfaces. These assumptions are needed for the traditional procedure, their use raises question about the accuracy and applicability of the results. It found that investigating coupling effect between gear dynamics and surface wear in the warn out surfaces are gives internal excitation incorporated with certain kind of dynamic gear model. The dynamic model in the above studies only included the torsional deflection in gear shaft systems. Shaft bending and bearing radial deflections which will not considered in dynamic models, which may degenerate the prediction accuracy of the dynamic analysis. To guarantee the prediction accuracy, a comprehensive translational, rotational, coupled, dynamic nonlinear model with three degree of freedom for a spur gear, this system is proposed and then combined with a quasi-static wear. Based on this combined dynamic surface wear determination methodology, the affected surface wear evaluate on the dynamic behavior of spur gear system.

II LITERATURE REVIEW

Mrs. C.M. Meenakshi

This paper dealing the objective to study the various stress state of spur gear and finding the tangential and radial forces which acted on different point that basis we can analyze by applying the forces. By using Ansys software contact and bending stress on the tooth of spur gear drive is found. Gears are machine elements used to radiate the power between rotating shafts by means of sponsalia of reckoning called teeth. They different from a small in size used in clocks or watches and larger gears used in automobile section, heavy duty applications, bridge lifting machine and rail road. In real world The gears are main and important element (or hurt) of machines such, cargo loading, tractors, metal cutting machine, rolling mills, hosting and transmitting and transporting machinery, massive engines etc.

Sameer Chakravarthy N C

This paper fully focused on fatigue analysis and fatigue life is determined by FE package ANSYS 11.0. In this paper gear is fixed in the gearbox this transmission unit of Armored tracked vehicle is which will be finding considerable fatigue damage over its life period due to the dynamic excitations occur by the terrain undulations, the rotating wheel and track assemblies. in this paper first static analysis of the model was done and validate the model and the boundary conditions correctness. The spectrum of stress variation is obtained and given input to the fatigue analysis and fatigue life is determined by FE package ANSYS 11.0.

Anders Flodin

This proposed work on the wear in gear flanks. At this point results regarding the consignment of wear calculation and their distribution. In this paper the distribution of mild wear is observed using some existing wear models and numerical methods. Spur and helical gears are treated. Finally, FZG machine used to assessment both the wear development of a gear wheel and its distribution. This test differentiates simulated results for assay of the simulation. Hence small changes of the shape of a surface can lead to increased surface pressures, mild wear on a gear flank can increase to surface pressures above fatigue limits. Calculating the wear of spur gears. To determine Surface behavior with the help of modified Archard's equation to determine the wear. Two other types of models were tested namely an oxidation Model and adsorption model. The modified Archard's model describes the wear. The helical gear was modeled with the same parameters as the FZG gears but with a helical angle. The tooth of helical gear thin uncoupled spur gears, which are allowed to deform individually. In this simulation, the surfaces are considered as Hertz surfaces. In The transmission error load was investigated and found to be minimized by the wear. The replicas and the teeth of gear was analyzed using optical microscopy and SEM, stylus instrument. Attention may be brought to rapid initial wear at the starting point of the active flank of the pinion. The change in value from the original involute tooth profile found and the number of teeth in mesh changes was also observed. Analyzing a tooth flank as a whole. Find the wear simulations, which give the properties of mild wear of a tooth surface.

V. Rajaprabakaran

This Research Paper constitutes the study Analysis shows that aero-fin shaped hole introduced along the stress flow direction yielded better results; Gears transmit the power. The stress concentration localized at the root and the point of contact. The continuous stress on the fillets arrise the fatigue failure of gear tooth. The main importance of this paper is to create different sizes of holes to reduce stress concentration factor. The FEA of Spur gear with a piece of three teeth is taken consideration for analysis and stress concentration minimizes hole size are included on gear teeth at different locations.

Abhijit Mahadev Sankpal

In This paper the Contact stress that refers to the localized stresses which develops as two curved surfaces come in contact and deform lightly under the given loads. Due to which contact stresses takes place at gear tooth. Wear is progressive unfasten of metal from the surface. Due to this tooth gets weakened. Gear Pitting is failure of the gear tooth. It happens due to wrong alignment; wrong selection of viscosity of the lubricant used, and contact stress increasing the surface fatigue strength of the given material. Due to separate of material a score is formed. At last stage contact stresses are find using FEM method and experimental method by utilize the Polariscope. Compare the FEM result with experimental result.

John M. Thompson

This paper discussed the wear analysis in gear is not done in FEA software. But, wear is important in many structure subjected to continuous loadings and it may be critical for some of the tribological applications which contain the prediction of the sealing potential of surfaces. The procedure is transfer the outcome of wear is calculated and applied in the overall analysis of the structure. For wear strain calculated by Archard equation which is used to alter the elastic strain of an element.

Archard Equation:

Spur gears are subjected to high contact and bending stresses at tooth mesh. These stresses develop wear surfaces during operation and are initiation point of crack propagation. Hence investigation of wear at this loading condition forms basis of this project work. Modeling spur gear in CAD software and analyzing it for induced structural stresses and deformation in CAE software. It is also tested experimentally for the structural stresses and results were correlated with analysis results.

The starting point for any discussion of wear on the macro scale is the Archard Equation, which states that:
W = K * s * P

Where W is the worn volume, s is the sliding distance, P is the applied load and K is the wear per unit load per sliding distance. Archard says "[K] may be described as the coefficient of wear and, in a series of experiments with the same combination of materials; changes in [K] denote changes in surface conditions." The Archard equation assumes that the wear rate is independent of apparent area of contact.

III OBJECTIVE

1) To find out the Von Mises Stresses generated are within yield limit or not at at different torque loading conditions.
2) To find out the sliding distances at different torque loading conditions with the help of ANSYS.
3) To find out the contact pressure at different torque loading conditions with the help of ANSYS.
4) To find out the deformation at different torque loading conditions with the help of ANSYS.
5) To find out the theoretical warn volume by using archard equation.
6) To find out the sliding distances with the help of experimentation.
7) To compare the sliding distances from ANSYS and Experimental.

A) EXISTING CAD MODEL AND PROPERTIES

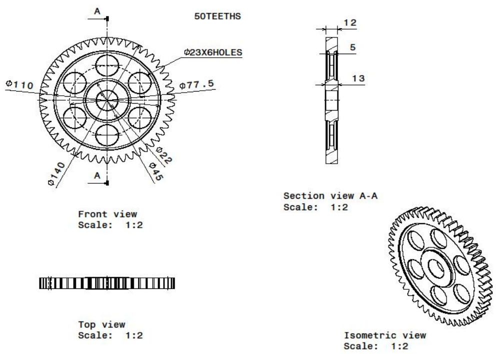

Fig.1 Drafted model of spur gear

Fig.2 Actual model of spur gear

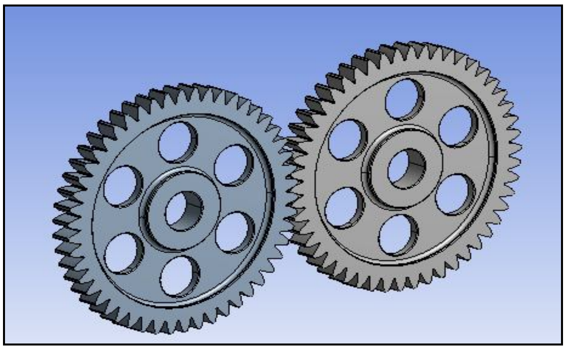

Fig.3 CAD model of spur gear

Gear parameter used:

Gear type: -	Standard involute teeth
Material: -	20MnCr5 alloy steel
Pressure angle: -	20 ^{0}Stub
Gear ratio: -	1

Table 1: Composition of alloy steel (20MnCr5)	
Constituent Material	**Percentage Content (%)**
Carbon (C)	0.17-0.22
Silicon (Si)	0.4 Max
Manganese (Mn)	1.10-1.40
Sulphur (S)	0.035 Max
Phosphors (P)	0.035 Max
Chromium (Cr)	1.00-1.30

Table 2: Mechanical Properties of alloy steel (20MnCr5)		
Properties	**Values**	**Unit**
Young's Modulus	2 x10^5	N/MM^2
Tensile Strength	650-880	Mpa
Elongation	8-25	%
Fatigue	275	Mpa
Yield Strength	520	Mpa
Poisons Ratio	0.3	-
Density	7850	Kg/M^3

Table 3: Wear Classifications and Mechanisms		
Classification	**Wear Mechanisms**	**Wear coefficient K (range)**
Wear dominated by mechanical behavior of materials	1. Asperity deformation and removal 2. Wear caused by plowing 3. Delamination wear 4. Adhesive wear 5. Abrasive wear 6. Fretting wear 7. Wear by solid particle impingement	10-4 10-4 10-4 10-4 10-2 to 10-1 10-6 to 10-4 -
Wear dominated by chemical behavior of Materials	1. Solution wear 2. Oxidation wear 3. Diffusion wear 4. Wear by melting of the surface layer 5. Adhesive wear at high temperatures	

IV. FINITE ELEMENT ANALYSIS

It is a numerical technique for finding approximate solutions to boundary value problems for partial differential equations. It is also referred to as finite element analysis (FEA). FEM subdivides a large problem into smaller, simpler, parts, called finite elements. The simple equations that model these finite elements are then assembled into a larger system of equations that models the entire problem. FEM then uses vibrational methods from the calculus of variations to approximate a solution by minimizing an associated error function.

1) Discretization and mesh generation of model

Table 4: Discretization of existing gear model	
Title	**Details**
Element Type	Tetrahedron
Element Order	Second Order
Mesh Method	Solid
Node Population count	97132
Element Population count	52867

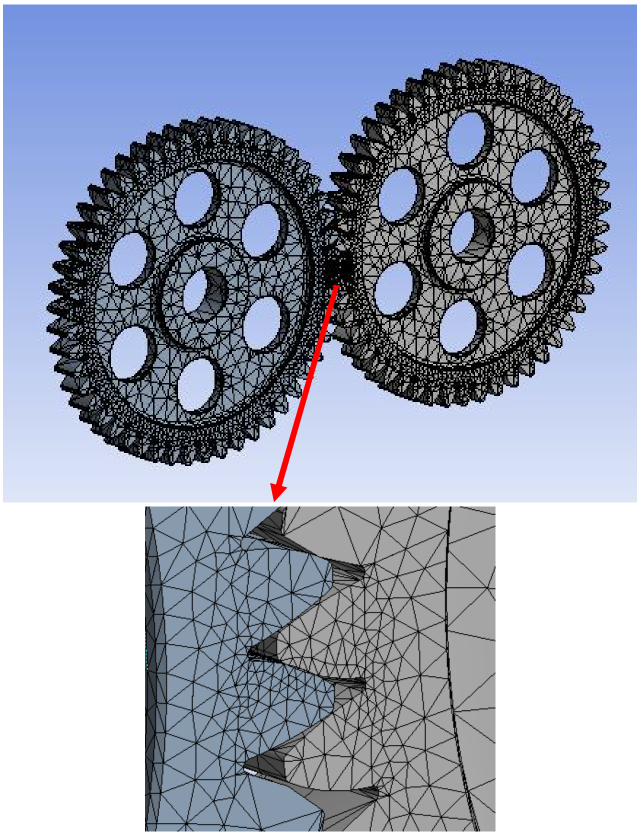

Fig.4 Discretization of CAD model and mesh generation

2) Frictional wear to gear

Fig.5 Frictional wear to gear

3) Gear Analysis with Torque 1.24 N.m

a) Boundary Conditions

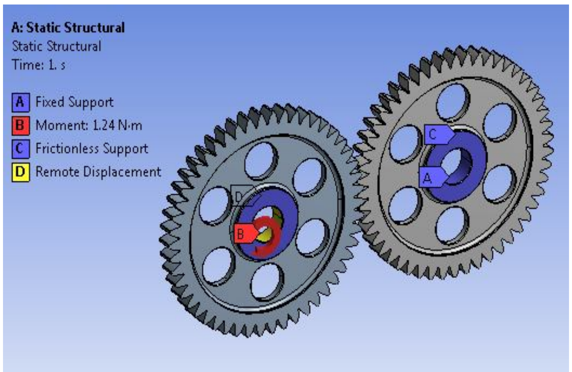

Fig.6 Boundary condition for gear model with torque 1.24 N-m

b) Von-Mises Stress

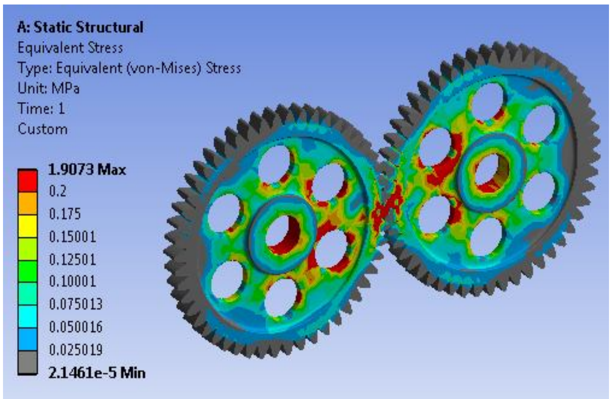

Fig.7 Von-Mises Stress at torque 1.24 N-m

c) Total deformation

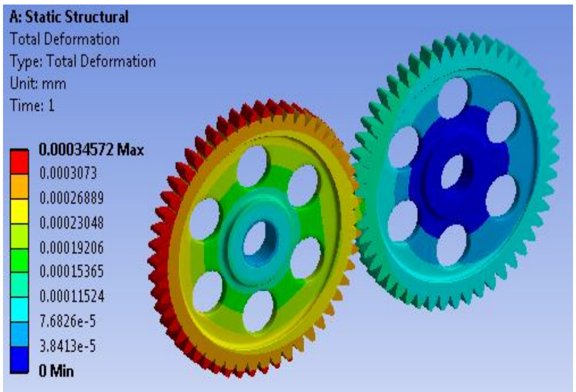

Fig.8 Total deformation at torque 1.24 N-m

d) Sliding status

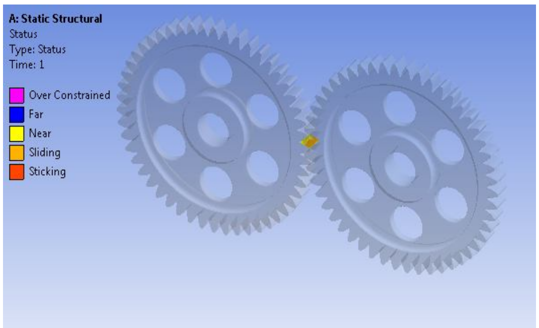

Fig.9 Sliding status at torque 1.24 N-m

e) Pressure

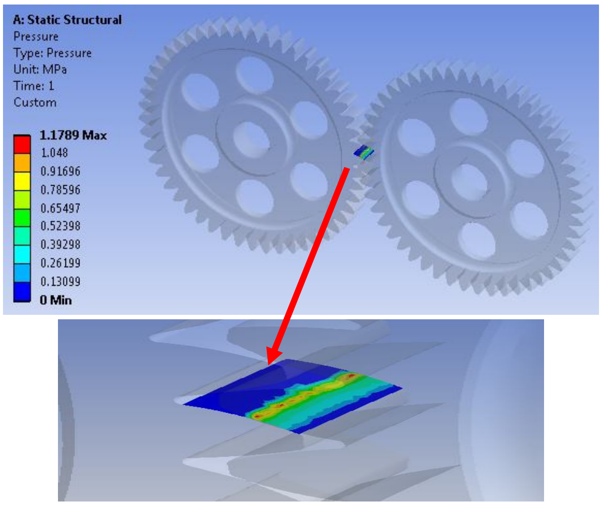

Fig.10 Pressure at torque 1.24 N-m

f) Sliding distance

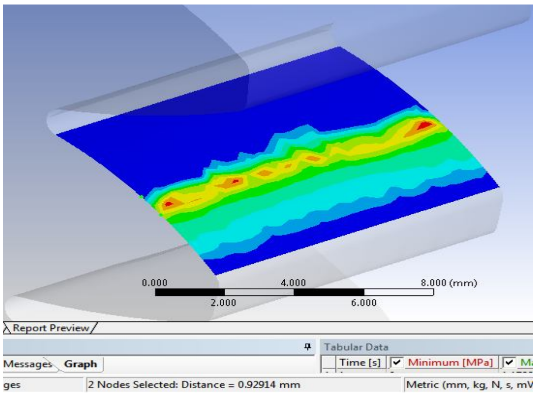

Fig.11 Sliding distance at torque 1.24 N-m

Table 5: Gear Analysis with Torque 1.24 N-m		
Properties	**Maximum**	**Minimum**
Von-misses stress (Mpa)	1.9073	2.1461e-5
Total deformation (mm)	0.00034572	0
Pressure (Mpa)	1.1789	0
Sliding distance (mm)	0.92914	0

4) Gear Analysis with Torque 3.74 N.m

a) Boundary Conditions

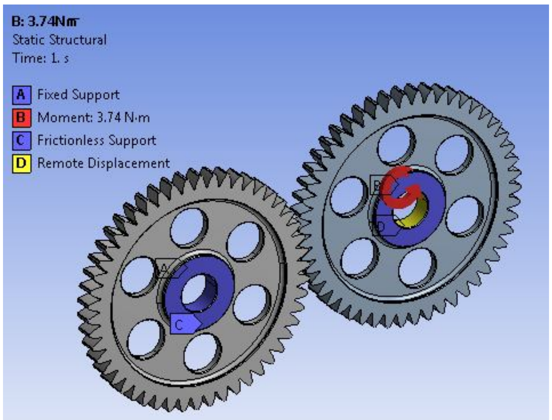

Fig.12 Boundary condition for gear model with torque 3.74 N-m

b) Von-Mises Stress

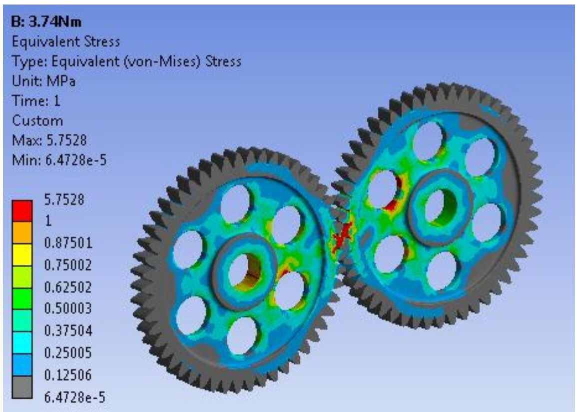

Fig.13 Von-Mises Stress at torque 3.74 N-m

c) Total deformation

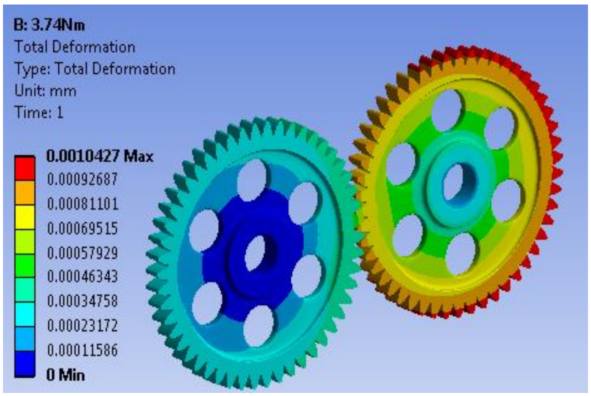

Fig.14 Total deformation at torque 3.74 N-m

d) Sliding status

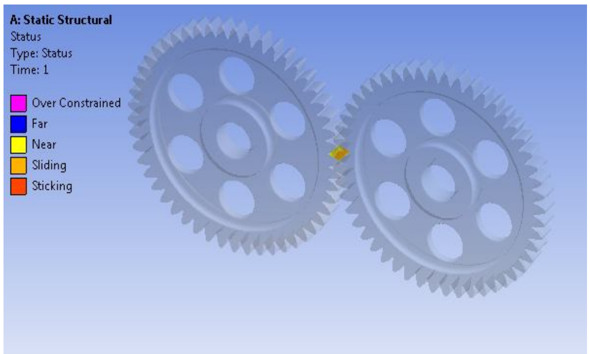

Fig.15 Sliding status at torque 3.74 N-m

e) Pressure

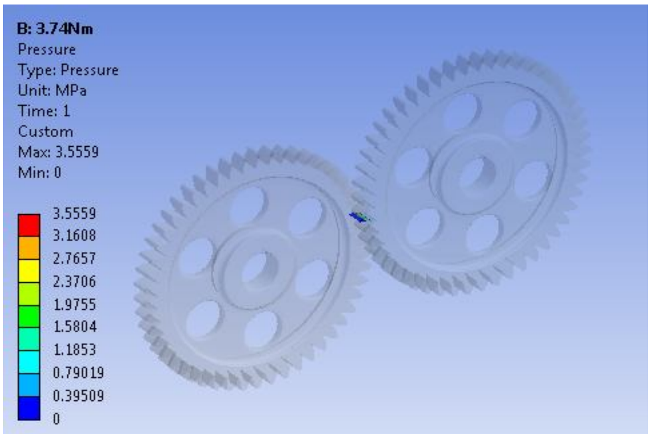

Fig.16 Pressure at torque 3.74 N-m

f) Sliding distance

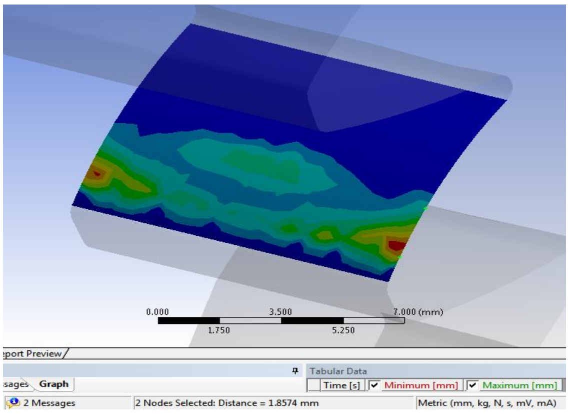

Fig.17 Sliding distance at torque 3.74 N-m

Table 6: Gear Analysis with Torque 3.74 N-m		
Properties	**Maximum**	**Minimum**
Von-misses stress (Mpa)	5.7528	6.4728e-5
Total deformation (mm)	0.0010427	0
Pressure (Mpa)	3.5559	0
Sliding distance (mm)	1.8574	0

5) Gear Analysis with Torque 6.24 N-m

a) Boundary Conditions

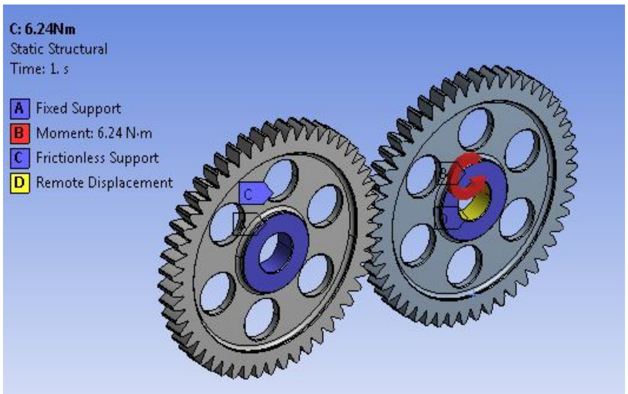

Fig.18 Boundary condition for gear model with torque 6.24 N-m

b) Von-Mises Stress

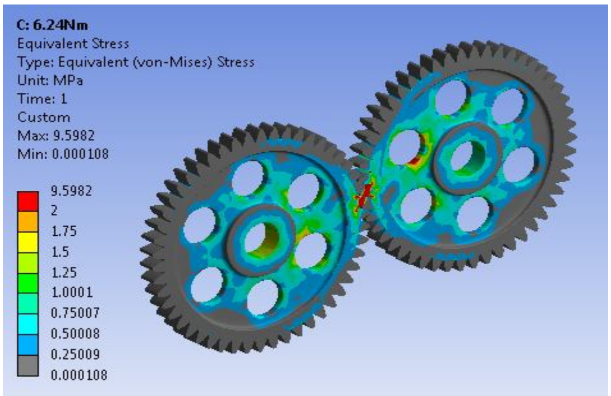

Fig.19 Von-Mises Stress at torque 6.24 N-m

c) Total deformation

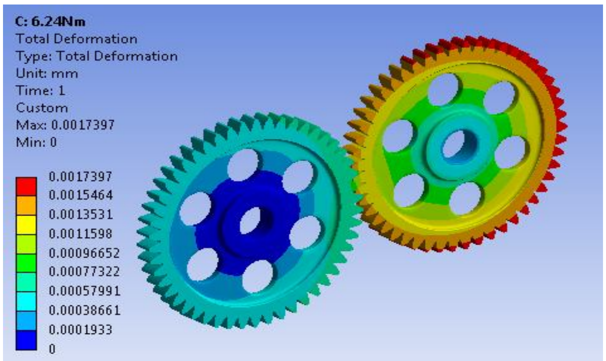

Fig.20 Total deformation at torque 6.24 N-m

d) Sliding status

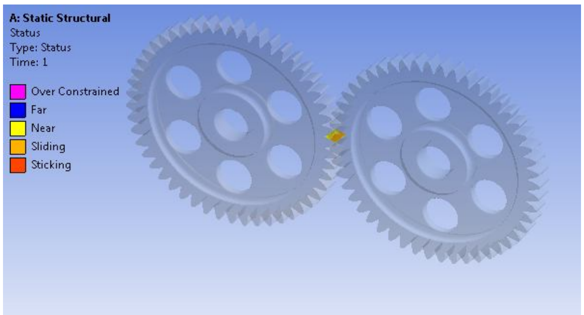

Fig.21 Sliding status at torque 6.24 N-m

e) Pressure

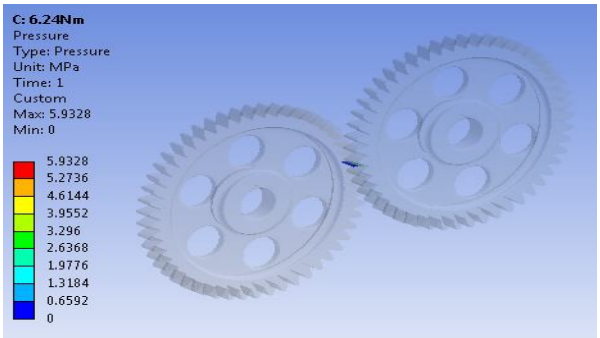

Fig.22 Pressure at torque 6.24 N-m

f) Sliding distance

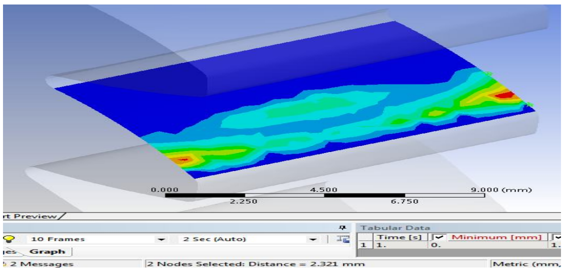

Fig.23 Sliding distance at torque 6.24 N-m

Table 7: Gear Analysis with Torque 6.24 N-m

Properties	Maximum	Minimum
Von-misses stress (Mpa)	9.5982	0.000108
Total deformation (mm)	0.0017397	0
Pressure (Mpa)	5.9328	0
Sliding distance (mm)	2.321	0

6) Gear Analysis with Torque 11.24 N-m

a) Boundary Conditions

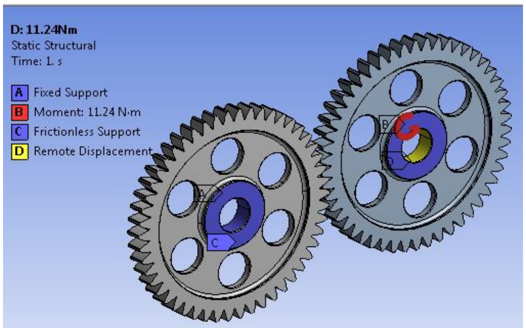

Fig.24 Boundary condition for gear model with torque 11.24 N-m

b) Von-Mises Stress

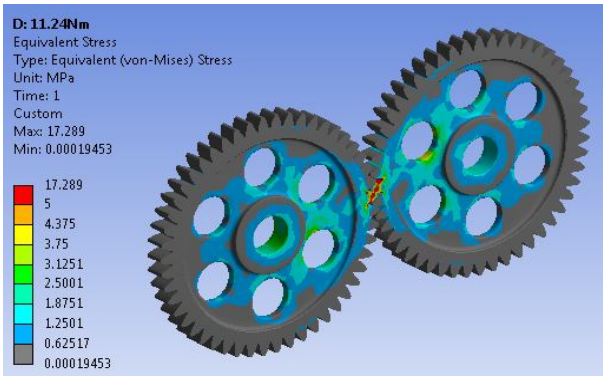

Fig.25 Von-Mises Stress at torque 11.24 N-m

c) Total deformation

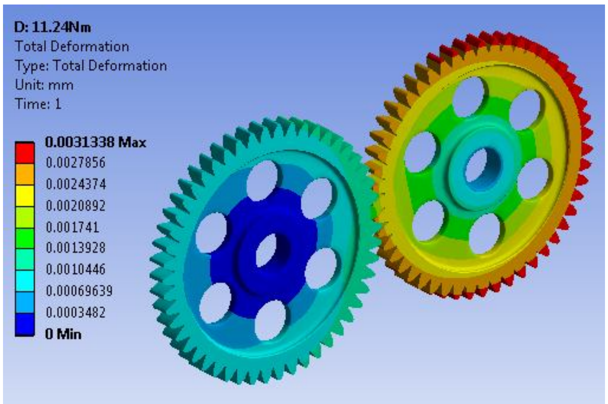

Fig.26 Total deformation at torque 11.24 N-m

d) Sliding status

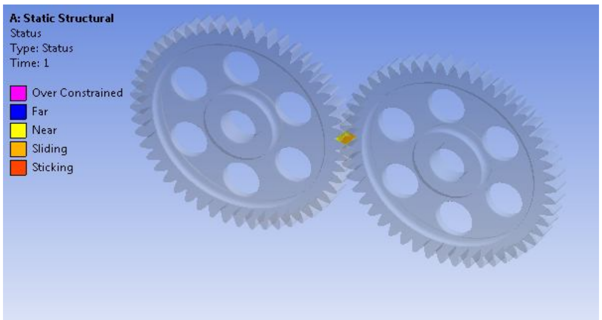

Fig.27 Sliding status at torque 11.24 N-m

e) Pressure

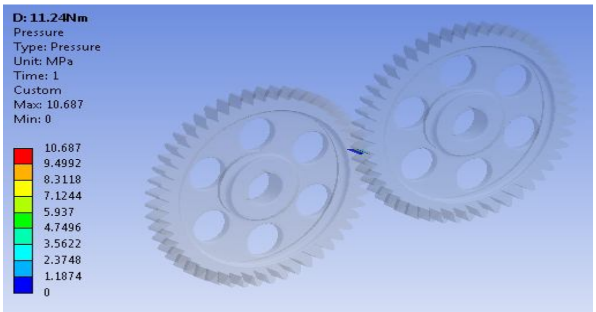

Fig.28 Pressure at torque 11.24 N-m

f) Sliding distance

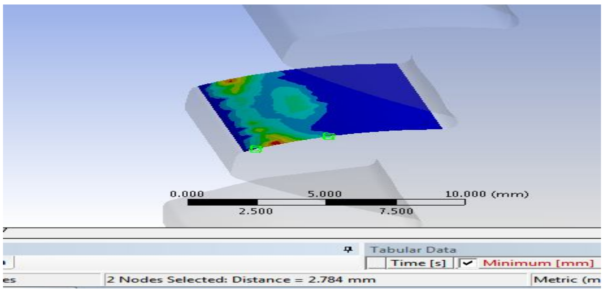

Fig.29 Sliding distance at torque 11.24 N-m

Table 8: Gear Analysis with Torque 11.24 N-m		
Properties	**Maximum**	**Minimum**
Von-misses stress (Mpa)	17.289	0.00019453
Total deformation (mm)	0.0031338	0
Pressure (Mpa)	10.687	0
Sliding distance (mm)	2.784	0

Wear calculation

By the Archard Equation

$$W = K * s * P$$

Where

W is the worn volume in mm^3.
s is the sliding distance in mm.
P is the contact pressure in Mpa.
K is the wear per unit load per sliding distance selected from table No.3 Wear caused by plowing.

Table 9: Sliding distances and contact pressure from FEA result		
Torque (N-m)	**Sliding distance (mm)**	**Contact pressure (Mpa)**
1.24	0.92914	1.1789
3.74	1.8574	3.5559
6.24	2.321	5.9328
11.24	2.784	10.687

Theoretical wear calculation from above table no.9 given below table No.10 result by using Archard equation above mentioned

Table 10: Theoretical wear calculation result	
Torque (N-m)	**Wear or Warn volume (mm3)**
1.24	1.0954×10^{-3}
3.74	6.6048×10^{-4}
6.24	1.3770×10^{-3}
11.24	2.9753×10^{-3}

V. EXPERIMENTAL ANALYSIS

Gear manufacturers have so far been occupied with failure due to high root stress, misalignments of gear shaft, chemical interactions between two bodies, high surface pressure and hardening cracks; they have neglected investigation of mild wear and its connection to more types of damages. Surface fatigue is known problem and the cure for it has been better and purer materials, smoother surface, heat-treated etc.

In this experimentation, only dry (non-lubricated) sliding wear considered. Actual wear mechanism for dry wear depends on a number of variables includes surface finish, surface geometry, orientation, sliding speed, relative hardness, material microstructure, and more. From this variables, it can be seen that wear rate is not pure material property and does not always occur uniformly, this experimental set-up like FZG machine (Forschungsstelle fur Zahnrader und Getriebebau) but wear investigation done with the help traditional optical microscopy, Result taken with the help of Stereo Microscope and Image Analysis System visualization method.

In this experimentation, we fabricated the test rig, first up all we taken mild steel ERW square tube 25x25x1.5 mm thk and cutting would be done in a specified manner to achieve final specification. we taken standard EN8 round bar ø25 mm and get finished in to OD ø21.996 mm for the fitment with standard bearing sizes. In this set-up v-belt pulley and v-belt fenner make for transmitting power from source to destination, for performing the mechanical advantages. For drive continuous rating C.SIR type motor is used having specification below mentioned.

Table 11: Drive Specification

Parameters	Values	Unit
Make	American Universal Electric	
Type	C.SIR Type	-
Power	1/4	H.P.
	0.180	Kw
Voltage	220/240	Volt
Frequency	50	Hz
Current	2.4	Amp

In this set-up, Bearing is used for alignment and stability.

Gears is the main constituent in that project to analyses the wear or scoring at the time of loading condition, specification of gear mentioned above in the existing CAD model and properties bit. In this set-up or test rig preparation with the help of fabricate the body by Co2 and welding rod or filler material is used E6013 size is ø2.4 and fabrication will be done and final assembly would be done, set-up or test rig is ready for experimentation.

Fig.30 Actual experimental set-up

Experimental wear analysis:-

in this experimentation not calculate the wear or worn volume due to actual contact pressure but sliding distance can be calculate and comparing with theoretical (FEA) and decide how much wear in gear at different torque loading conditions.

Machine specifications:

i) **Stereo Microscope**
Make: Wuzhou New Found Instrument Co.Ltd.,China
Mode: XTL 3400E, Magnification: 10 X

ii) **Image Analysis System**
Make: Chroma Systems Pvt. Ltd., India
Model: MVIG 2005

Fig 31 Stereo Microscope and Image Analysis

a) Gear Analysis with Torque 1.24 N-m

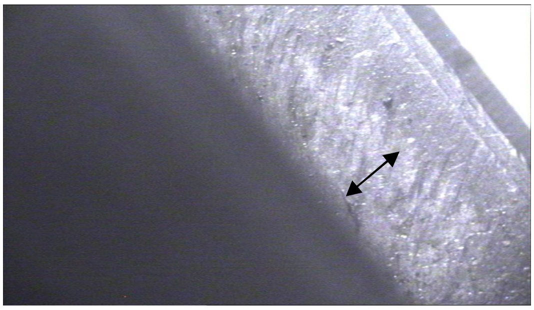

Fig.32 Stereo image, sliding distance is 1.30 mm

b) Gear Analysis with Torque 3.74 N-m

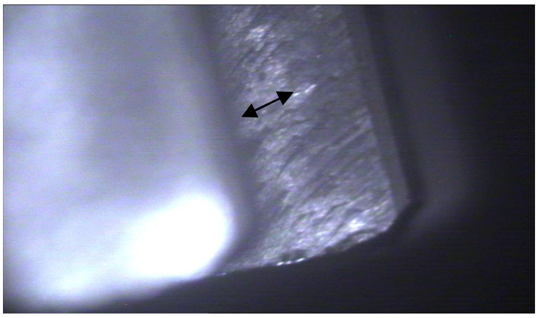

Fig.33 Stereo image, sliding distance is 1.80 mm

c) Gear Analysis with Torque 6.24 N-m

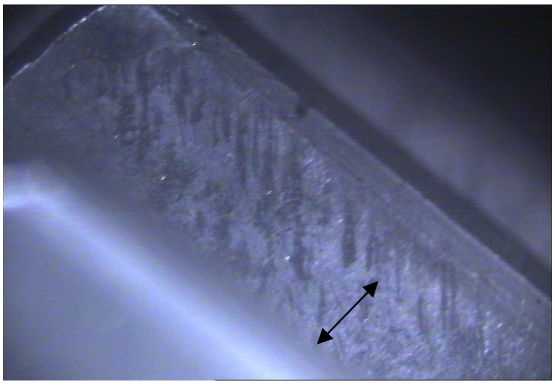

Fig.34 Stereo image, sliding distance is 2.780 mm

d) Gear Analysis with Torque 11.24 N-m

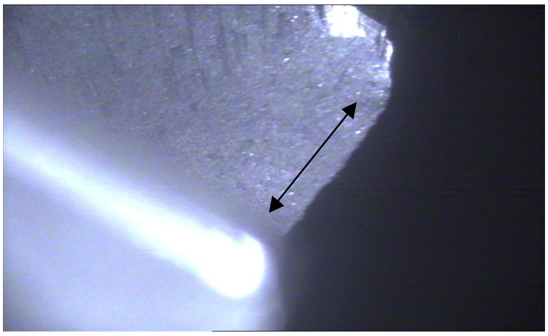

Fig.35 Stereo image, sliding distance is 3.280 mm

VI. RESULT

Comparing the results of FEA and Experimental is observed below table No.12

Table 12: Sliding distances by FEA and Experimental result

Torque (N-m)	Sliding distance by FEA (mm)	Sliding distance by Experimental (mm)
1.24	0.92914	1.300
3.74	1.8574	1.800
6.24	2.321	2.780
11.24	2.784	3.820

From above table No. 12 I analyzed that as torque increases sliding distance also increases hence due to increase in sliding distance wear or warn volume also increases.

Contact pressure by using FEA are given in the below table No.13

Table 13: Contact pressure by FEA

Torque (N-m)	Contact pressure (Mpa)
1.24	1.1789
3.74	3.5559
6.24	5.9328
11.24	10.687

VII. CONCLUSION

1) Archard Equation is the one of the method to calculate wear or warn volume.
2) Von Mises Stresses generated are within yield limit of the material i.e 17.289 Mpa at torque 11.24 N-m
3) Contact pressure is maximum i.e 10.687 Mpa at 11.24 N-m torque.
4) Sliding distance is maximum i.e 2.784 mm (by FEA) and 3.280 mm (Experimental) at 11.24 N-m torque.
5) Sliding distance increases gear going to be an backlash, due to this gear tooth damage.
6) Using stereo microscope and image analysis system to find the best result.

Table 14: Sliding distances by FEA and Experimental result

Torque (N-m)	Sliding distance by FEA (mm)	Sliding distance by Experimental (mm)	Remark
1.24	0.92914	1.300	Torque increases sliding distances also increases and the gear going to be failure
3.74	1.8574	1.800	
6.24	2.3210	2.780	
11.24	2.7840	3.280	

Table 15: Contact pressure by FEA

Torque (N-m)	Contact pressure (Mpa)	Remark
1.24	1.1789	Torque increases contact pressure also increases and the gear going to be failure
3.74	3.5559	
6.24	5.9328	
11.24	10.687	

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