Stress Analysis of Bevel Gear Tool, Using FEA Tool ANSYS V.14

Ms. Shweta Khaire[1] Mr. Shachindra Sheode[2]

Abstract—*Bevel gears are used widely at different applications in Industrial Machinery, especially in the automotive industry. Since bevel gears have such a great range of applications. This investigation gives a detailed approach to bevel gear design and analysis. Key design parameters are investigated in accordance with industry standards. Potential gear materials are described leading to the selection of SAE 2800 steel as the proper material for this application, finished with carburization and case hardening processes. A final gear design is proposed and analyzed to show that proper margins of safety have been included in the design. Fatigue analysis is conducted at the two most critical sections of the gear shaft resulting in margins of safety using FEA Tool ANSYS V.14. Further analysis is conducted on Total Deformation, Directional Deformation, Equivalent Stress, Shear Stress, Maximum Shear Stress, Equivalent Elastic Strain, Shear Elastic Strain, Maximum Shear Elastic Strain, Stress Intensity, and Elastic Strain Intensity using FEA Tool ANSYS V.14. Results are compared to the recommended allowable stresses as published by the American Gear Manufacturing Association.*

Keywords: -Rotational frequency: - N Measured in rotation over time, such as RPM.Angular frequency: - ω Measured in radians per second.

Root diameter: - Diameter of the gear, measured at the base of the tooth.

Addendum: - a, Radial distance from the pitch surface to the outermost point of the tooth.

Dedendum: - b, Radial distance from the depth of the tooth trough to the pitch surface.

I. INTRODUCTION

A gear is a mechanical device often used in transmission systems that allows rotational force to be transferred to another gear or device. The gear teeth, or cogs, allow force to be fully transmitted without slippage and depending on their configuration, can transmit forces at different speeds, torques, and even in a different direction. Throughout the mechanical industry, many types of gears exist with each type of gear possessing specific benefits for its intended applications. Bevel gears are widely used because of their suitability towards transferring power between nonparallel shafts at almost any angle or speed. Spiral bevel gears have curved and sloped gear teeth in relation to the surface of the pitch cone. As a result, an oblique surface is formed during gear mesh which allows contact to begin at one end of the tooth (toe) and smoothly progress to the other end of the tooth (heel), as shown below in Figure 1.1 Spiral bevel gears, in comparison to straight or zero bevel gears, have additional overlapping tooth action which creates a smoother gear mesh.

This smooth transmission of power along the gear teeth helps to reduce noise and vibration that increases exponentially at higher speeds. Therefore, the ability of a spiral bevel gear to change the direction of the mechanical load, coupled with their ability to aid in noise and vibration reduction, make them a prime candidate for use in the helicopter industry

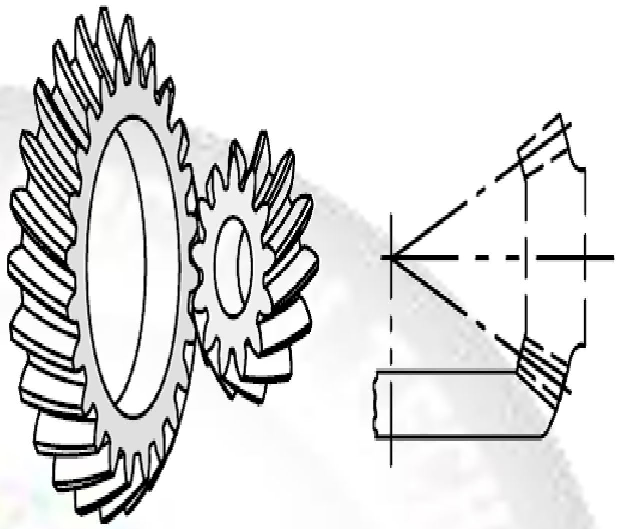

Fig.1: Spiral Bevel Gear Mesh

A. Geometry & Terminology

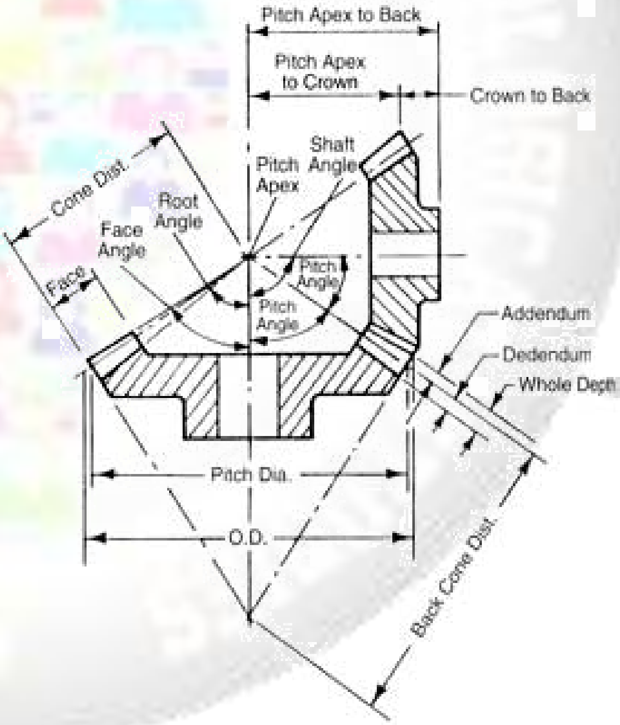

Fig. 2: Bevel Gears in Mesh

When intersecting shafts are connected by gears, the cones are tangent along an element, with their apexes at the intersection of the shafts as in Figure 1.7 where two bevel gears are in mesh. The size and shape of the teeth are defined at the large end, where they intersect the back cones. Pitch cone and back cone elements are perpendicular to each other. The tooth profiles resemble those of spur gears having pitch radii equal to the developed back cone radii.

$$Z_{v1} = (2\tau\, r_{b1}/p) = Z_1/\cos\gamma$$
$$Z_{v2} = (2\tau\, r_{b2}/p) = Z_2/\cos\gamma$$

Where Z_v is called the virtual number of teeth, p is the circular pitch of both the imaginary spur gears and the bevel gears. Z_1 and Z_2 are the number of teeth on the pinion and gear, γ_1 and γ_2 are the pitch cone angles of pinion and gears. It is a practice to characterize the size and shape of bevel gear teeth as those of an imaginary spur gear appearing on the developed back cone corresponding to Tredgold's approximation.

1. Bevel gear teeth are inherently non - interchangeable.
2. The working depth of the teeth is usually 2m, the same as for standard spur and helical gears, but the bevel pinion is designed with the larger addendum (0.7 working depth).
3. This avoids interference and results in stronger pinion teeth. It also increases the contact ratio.
4. The gear addendum varies from 1m for a gear ratio of 1 to 6.7 and 0.54m for ratios of 6.8 and greater.

B. Bevel Gear Applications

Application requirements should be considered with the workload and environment of the gear set in mind.

1. Power, velocity and torque consistency and output peaks of the gear drive so the gear meets mechanical requirements.
2. Inertia of the gear through acceleration and deceleration. Heavier gears can be harder to stop or reverse.
3. Precision requirement of gear, including gear pitch, shaft diameter, pressure angle and tooth layout.
4. Handedness (left or right teeth angles) for spiral and hypoid bevel gears.
5. Gear lubrication requirements. Some gears require lubrication for smooth, temperate operation.
6. Mounting requirements. Application may limit the gear's shaft positioning.
7. Noise limitation. Commercial applications may value a smooth, quietly meshing gear.
8. Corrosive environments. Gears exposed to weather or chemicals should be especially hardened or protected.
9. Temperature exposure. Some gears may warp or become brittle in the face of extreme temperatures.
10. Vibration and shock resistance. Heavy machine loads or backlash, the deliberate surplus space in the circular pitch, may jostle gearing.
11. Operation disruption resistance. It may be necessary for some gear sets to function despite missing teeth or misalignment.

C. Bevel Gear Materials

Gear composition is determined by application, including the gear's service, rotation speed, accuracy and more.

1. Cast iron provides durability and ease of manufacture.
2. Alloy steel provides superior durability and corrosion resistance. Minerals may be added to the alloy to further harden the gear.
3. Cast steel provides easier fabrication, strong working loads and vibration resistance.
4. Carbon steels are inexpensive and strong, but are susceptible to corrosion.
5. Aluminum is used when low gear inertia with some resiliency is required.
6. Brass is inexpensive, easy to mold and corrosion resistant.Copper is easily shaped, conductive and corrosion resistant. The gear's strength would increase if bronzed.
7. Plastic is inexpensive, corrosion resistant, quiet operationally and can overcome missing teeth or misalignment. Plastic is less robust than metal and is vulnerable to temperature changes and chemicals. Acetyl, Delran, nylon, and polycarbonate plastics are common.
8. Other material types like wood may be suitable for individual applications.

D. Gear Failures Bending Fatigue

This common type of failure is a slow, progressive failure caused by repeated loading. It occurs in three stages:

Crack initiation. Plastic deformation occurs in areas of stress concentration or discontinuities, such as notches or inclusions, leading to microscopic cracks.

Crack propagation. A smooth crack grows perpendicular to the maximum tensile stress.

Fracture. When the crack grows large enough, it causes sudden fracture.

As a fatigue crack propagates, it leaves a series of "beach marks" (visible to the naked eye) that correspond to positions where the crack stopped the origin of the crack is usually surrounded by several concentric curved beach marks.

Most gear tooth fatigue failures occur in the tooth root fillet where cyclic stress is less than the yield strength of the material and the number of cycles is more than 10,000. This condition is called high-cycle fatigue. A large part of the fatigue life is spent initiating cracks, whereas a shorter time is required for the cracks to propagate.

Stress concentrations in the fillet often cause multiple crack origins, each producing separate cracks. In such cases, cracks propagate on different planes and may join to form a step, called a ratchet mark

II. CONTACT FATIGUE

In another failure mode, repeated stresses cause surface cracks and detachment of metal fragments from the tooth contact surface called contact or Hertz fatigue. The most common types of surface fatigue are macro pitting (visible to the naked eye) and micro pitting. Macro pitting occurs when fatigue cracks start either at or below the surface. As the cracks grow, they cause a piece of surface material to break out, forming a pit with sharp edges. Based on the type of damage, macro pitting is categorized as no progressive, progressive, spall or flake. The no progressive type consists of pits less than 1 mm diameter in localized areas. These pits distribute load more evenly by removing high points on the surface, after which pitting stops. Progressive macro pitting consists of pits larger than 1 mm diameter that cover a significant portion of the tooth surface.

In one type, called spelling, the pits coalesce and form irregular craters over a large area. In flake macro pitting, thin flakes of material break out and form triangular pits that are relatively shallow, but large in area. Micro pitting has a frosted, matte or gray stained appearance.

Under magnification, the surface is shown to be covered by very fine pits (< 20 mm deep). Metallurgical sections through these pits show fatigue cracks that may extend deeper than the pits.

A. Wear

Gear tooth surface wear involves removal or displacement of material due to mechanical, chemical or electrical action. The three major types of wear are adhesion, abrasion and polishing. Adhesion is the transfer of material from the surface of one tooth to that of another due to welding and tearing. It is confined to oxide layers on the tooth surface. Adhesion is categorized as mild or moderate, whereas severe adhesion is termed scuffing (described later).Typically, mild adhesion occurs during gear set run-in and subsides after it wears local imperfections from the surface. To the unaided eye, the surface appears undamaged and machining marks are still visible. Moderate adhesion removes some or all of the machining marks from the contact surface. Under certain conditions, it can lead to excessive wear. Abrasion is caused by contaminants in the lubricant such as sand, scale, rust, machining chips, grinding dust, weld splatter and wear debris. It appears as smooth, parallel scratches or gouges.

Abrasion ranges from mild to severe. Mild abrasion consists of fine scratches that don't remove a significant amount of material from the tooth contact surface, whereas moderate abrasion removes most of the machining marks. Severe abrasion, which removes all machining marks, can cause wear steps at the ends of the contact surface and in the dedendum. Tooth thickness may be reduced significantly, and in some cases, the tooth tip is reduced to a sharp edge. Finally, polishing is fine-scale abrasion that imparts a mirror-like finish to gear teeth. Magnification shows the surface to be covered by fine scratches in the direction of sliding. Polishing is promoted by chemically active lubricants that are contaminated with a fine abrasive. Polishing ranges from mild to severe. Its mild form, which is confined to high points on the surface, typically occurs during run-in and ceases before machining marks are removed. Moderate polishing removes most of the machining marks. Severe polishing removes all machining marks from the tooth contact surface. The surface may be wavy or it may have wear steps at the ends of the contact area and in the dedendum.

Scuffing. Severe adhesion or scuffing transfers metal from the surface of one tooth to that of another. Typically, it occurs in the addendum or dedendum in bands along the direction of sliding, though load concentrations can cause localized scuffing. Surfaces have a rough or matte texture that, under magnification, appear to be torn and plastically deformed. Scuffing ranges from mild to severe. Mild scuffing occurs on small areas of a tooth and is confined to surface peaks. Generally, it is no progressive moderate scuffing occurs in patches that cover significant portions of the teeth. If operating conditions do not change, it can be progressive. Severe scuffing occurs on significant portions of a gear tooth (for example, the entire addendum or dedendum). In some cases, surface material is plastically deformed and displaced over the tooth tip or into the tooth root. Unless corrected, it is usually progressive.

B. Modeling Involute Gears in Pro/Engineer

The Pro/Engineer using desire to design a gear should start with the basics: the involute curve. An involute is described as the path of a point on a straight line called the generatix, as it rolls along the convex base curve (the evolutes). The involute curve is most often used as the basis for the profile of a gear tooth.

While several techniques can be used to create the involute tooth profile in Pro/Engineer, this article focuses on using datum curves by equation. The benefit of this method are that the involute curve profile is based on the exact geometric equations, it is highly flexible in terms of the types of gears and curves that can be created .In addition the datum curve equation method allow the user to use either Cartesian or cylindrical coordinate systems to create the involute profile. Finally, the curves generated by this approach are automatically truncated at the major diameter of the gear without any additional operation.

*C. Fem Package Ansys*is the name commonly used for ANSYS mechanical, general-purpose finite element analysis (FEA) computer aided engineering software tools developed by ANSYS Inc. ANSYS mechanical is a self-contained analysis tool incorporating pre-processing such as creation of geometry and meshing, solver and post processing modules in a unified graphical user interface. ANSYS is a general-purpose finite element-modeling package for numerically solving a wide variety of mechanical and other engineering problems. These problems include linear structural and contact analysis that is non-linear.

Among the various FEM packages, in this work ANSYS is used to perform the analysis.

D. General Procedures to Create an Involute Curve

The sequence of procedures employed to generate the involute curve are illustrated as follows: -

1) Set up the geometric parameters

– Number of teeth
– Diametral Pitch
– Pressure angle
– Pitch diameter
– Face width
– Helix angle

2) Create the basic geometry such as addendum, dedendum and pitch circles in Support of the gear tooth.

3) Define the involute tooth profile with datum curve by equation using cylindrical coordinate system.

4) Create the tooth solid feature with a cut and extrusion. Additional helical datum curves are also required in this step to sweep helical gear teeth.

5) Pattern the tooth around the centre line axis.

S. No	Geometry Name	Gear 1	Gear 2
1	No. of teeth	18	28
2	RPM	10000rpm	10000 rpm
3	Rotational velocity	200 radian per sec	200 radian per sec
4	Diametric pitch	12.25 mm	20 mm
5	Addendum	200 mm	300 mm
6	Dedendum	160 mm	258 mm
7	Face width	6.67 mm	7.104 mm
8	Diameter	21 mm	31 mm
9	Material type	SAE 9310	SAE 9310

Table. 1: Key geometrical Parameters of Bevel Gear used

III. FEM Package

ANSYS is the name commonly used for ANSYS mechanical, general-purpose finite element analysis (FEA) computer aided engineering software tools developed by ANSYS Inc. ANSYS mechanical is a self-contained analysis tool incorporating pre-processing such as creation of geometry and meshing, solver and post processing modules in a unified graphical user interface. ANSYS is a general-purpose finite element-modeling package for numerically solving a wide variety of mechanical and other engineering problems. These problems include linear structural and contact analysis that is non-linear.

Among the various FEM packages, in this work ANSYS is used to perform the analysis.

The following steps are used in the solution procedure using ANSYS

1. The geometry of the gear to be analyzed is imported from solid modeler Pro/Engineer in IGES format this is compatible with the ANSYS.
2. The element type and materials properties such as Young's modulus and Poisson's ratio are specified
3. Meshing the three-dimensional gear model. Figure (5.2) shows the meshed 3D solid model of gear
4. The boundary conditions and external loads are applied
5. The solution is generated based on the previous input parameters.
6. Finally, the solution is viewed in a variety of displays.

IV. Totals And Directional Deformation Analysis

Deformation in continuum mechanics is the transformation of a body from a reference configuration to a current configuration. A configuration is a set containing the positions of all particles of the body. Contrary to the common definition of deformation, which implies distortion or change in shape, the continuum mechanics definition includes rigid body motions where shape changes do not take place.

Deformation is the change in the metric properties of a continuous body, meaning that a curve drawn in the initial body placement changes its length when displaced to a curve in the final placement. If none of the curves changes length, it is said that a rigid body displacement occurred.

It is convenient to identify a reference configuration or initial geometric state of the continuum body which all subsequent configurations are referenced from. The reference configuration need not be one the body actually will ever occupy. Often, the configuration at t = 0 is considered the reference configuration, $K_0(\mathbf{B})$. The configuration at the current time t is the current configuration.

Physical deformations can be calculated on an inside a part or an assembly. Fixed supports prevent deformation; locations without a fixed support usually experience deformation relative to the original location. Deformations are calculated relative to the part or assembly world coordinate system.

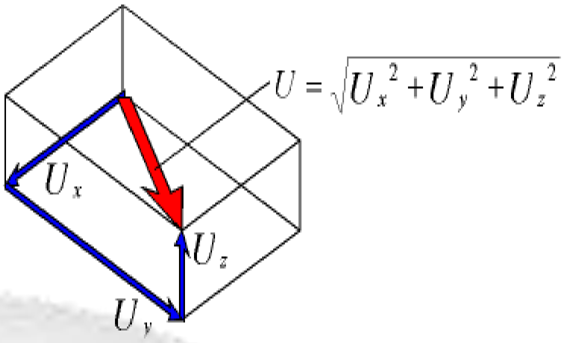

Fig. 3:
Component deformations (Directional Deformation)
Deformed shape (Total Deformation vector)

The three component deformations U_x, U_y, and U_z, and the deformed shape U are available

"Stress" measures the average force per unit area of a surface within a deformable body on which internal forces act, specifically the intensity of the internal forces acting between particles of a deformable body across imaginary internal surfaces.[3] These internal forces are produced between the particles in the body asa reaction to external forces. External forces are either surface forces or body forces. Because the loaded deformable body is assumed to behave as a continuum, these internal forces are distributed continuously within the volume of the material body, i.e. the stress distribution in the body is expressed as a piecewise continuous function of space and time.

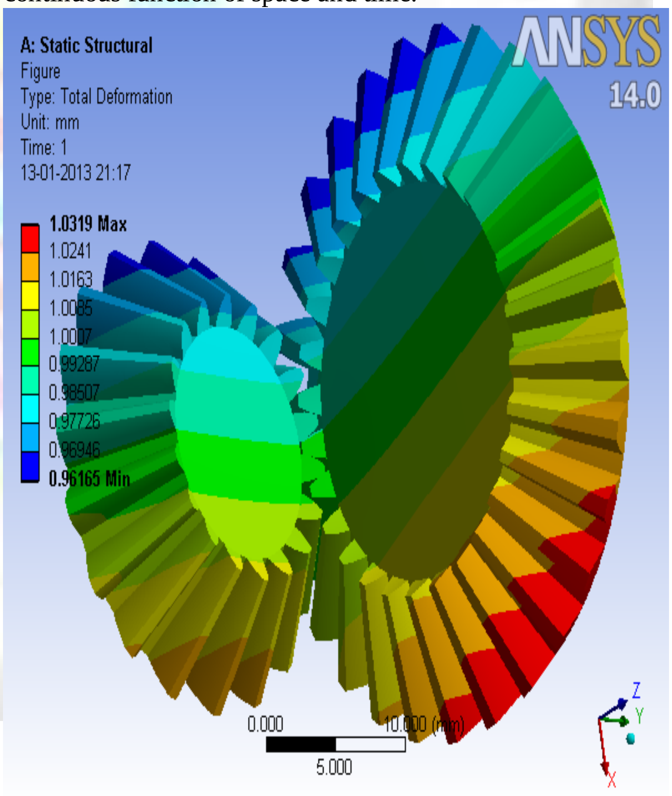

Fig. 4: Total Deformations of Two Bevel Gears

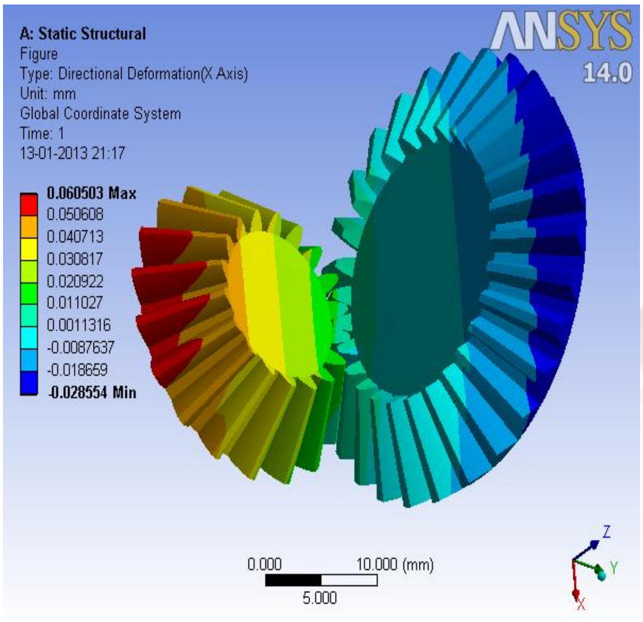

Fig. 5: Directional Deformations of Two Bevel Gears

A. *Normal stress*

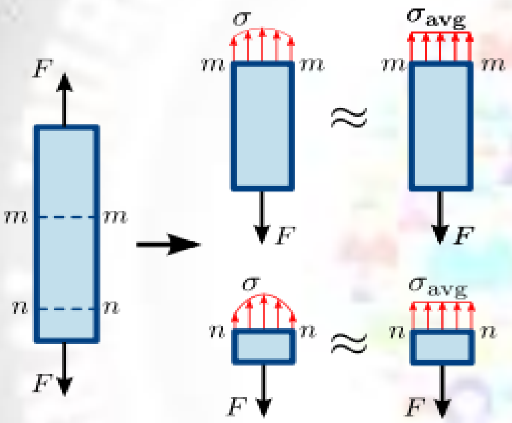

Fig. 6: Normal Stress

B. *Shear Stress*

A different type of stress occurs when the force occurs in shear, as shown in Figure 5.7 is called the shear force. Dividing the sheer force by the cross-sectional area we obtain the shear stress (tau).

$$\tau_{\text{avg}} = \frac{F_s}{A} \approx \tau$$

Shear stress can also be caused by various loading methods, including direct shear, torsion, and can be significant in bending. A shaft loaded in torsion experiences shear stress in the direction tangential to its axis. I-beams see significant shear in the web under bending loads; this is due to the web constraining the flanges.

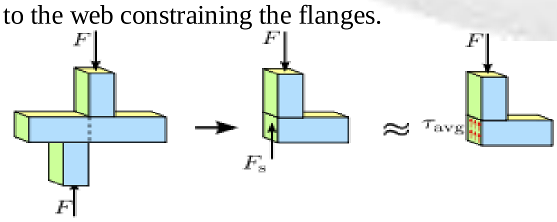

Fig. 7: Shear Stress

V. Stress Analysis

Stress analysis is the determination of the internal distribution of stresses in a structure. It is needed in engineering for the study and design of structures such as tunnels, dams, mechanical parts, and structural frames, under prescribed or expected loads. To determine the distribution of stress in a structure, the engineer needs to solve a boundary-value problem by specifying the boundary conditions. These are displacements and forces on the boundary of the structure.

Constitutive equations, such as Hooke's law for linear elastic materials, describe the stress-strain relationship in these calculations. When a structure is expected to deform elastically (and resume its original shape), a boundary-value problem based on the theory of elasticity is applied, with infinitesimal strains, under design loads.

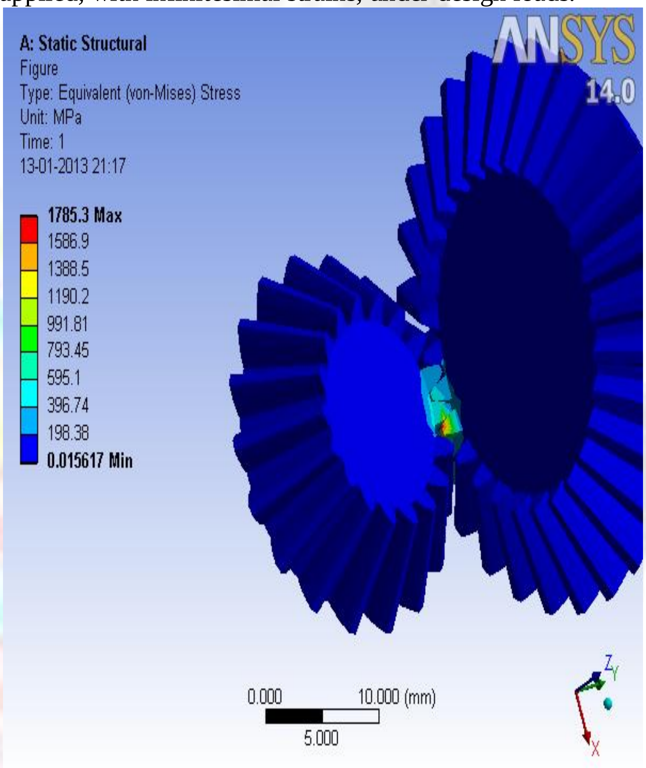

Fig. 8: Equivalent Stress of two Bevel Gears

When the applied loads permanently deform the structure, the theory of plasticity applies. Stress analysis is simplified when the physical dimensions and the distribution of loads allow the structure to be treated as one- or two-dimensional. For a two-dimensional analysis a plane stress or a plane strain condition can be assumed. Alternatively, stresses can be experimentally determined.

Computer-based approximations for boundary-value problems can be obtained through numerical methods such as the finite element method, the finite difference method, and the boundary element method. Analytical or closed-form solutions can be obtained for simple geometries, constitutive relations, and boundary conditions

A. *Experimental Results and Conclusions*

In continuum mechanics, stress is a measure of the internal forces acting within a deformable body. Quantitatively, it is a measure of the average force per unit area of a surface within the body on which internal forces act. These internal forces arise as a reaction to external forces applied to the body. Because the loaded deformable body is assumed to behave as a continuum, these internal forces are distributed continuously within the volume of the material body, and result in deformation of the body's shape. Beyond certain limits of material strength, this can lead to a permanent shape change or structural failure.

The stresses considered in continuum mechanics are only those produced during the application of external forces and the consequent deformation of the body; that is to say, relative changes in deformation are considered rather than absolute values. A body is considered stress-free if the only forces present are those inter-atomic forces (ionic, metallic, and van der Waals forces) required to hold the body together and to keep its shape in the absence of all external influences, including gravitational attraction. Stresses generated during manufacture of the body to a specific configuration are also excluded.

The dimension of stress is that of pressure, and therefore the SI unit for stress is the pascal (symbol Pa), which is equivalent to one Newton (force) per square meter (unit area), that is N/m^2. In Imperial units, stress is measured in pound-force per square inch, which is abbreviated as psi.

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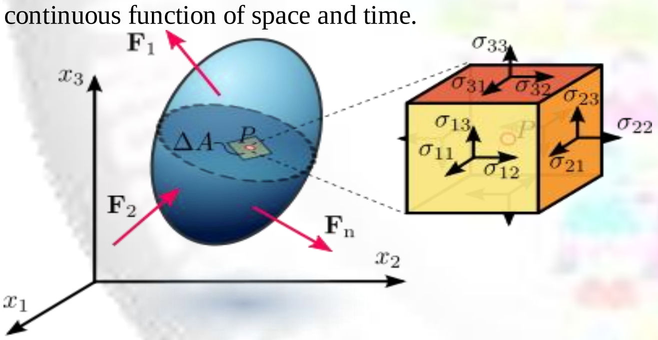

Fig. 9: Shows Stress in a loaded Deformable Material Body assumed as a Continuum

The stresses calculated were within the working limit, which prevents the gear from failure

Sr No	Object Name	Result	
		Minimum	**Maximum**
1	Total Deformation	0.96165mm	1.0319 mm
2	Directional deformation	-0.028554 mm	0.060503 mm
3	Equivalent Stress	1.5617e-002 MPa	1785.3MPa
4	Shear Stress	-295.78 MPa	536.18 MPa
5	Maximum Shear Stress	8.799e-003 MPa	928.41 MPa

Table. 2: Deformation and stress analysis

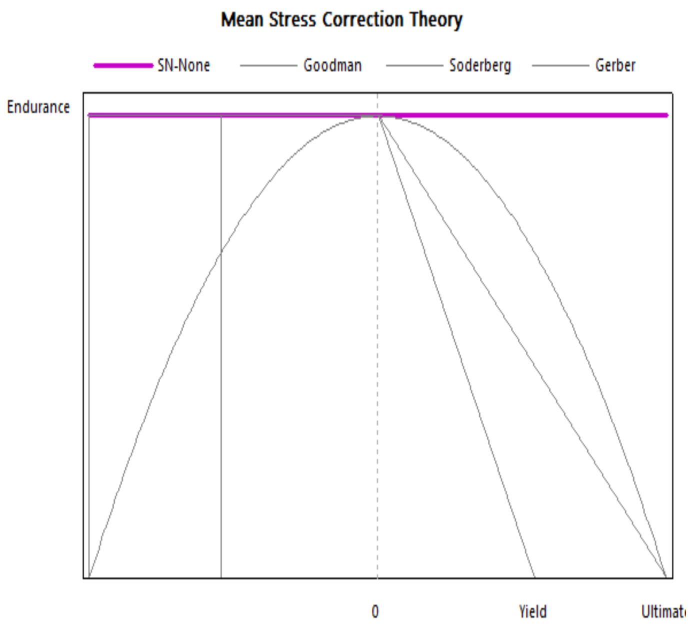

Fig. 10: Mean Stress Correction Theory

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