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Title: **A SCHEMTAIC DESIGN OF MULTI SPINDLE GEAR BOX AND ITS ANALYSIS UNDER STATIC LOADS**

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A SCHEMTAIC DESIGN OF MULTI SPINDLE GEAR BOX AND ITS ANALYSIS UNDER STATIC LOADS.

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ABSTRACT:
Now days Milling and drilling operations not uniquely did on single machine. To avoid this confusion of doing manufacturing operations a small difference in gear box design may give a single option solution to the manufacturers. A liver based shifter used as a differential to change gear systems to perform these two different operations on a single machine. On the proposed above operations torque generation will play an important role. From the survey taken before thesis base papers it is observed that low torque is good for milling and high torque is good for drilling. Gear box outline will be same for casing to avoid machine space and change in gear alignment by addition of a shifter to perform good results for drilling and milling. Angle drilling is one of the major time taking processes in large scale production industries. To overcome such deviation special purpose radial drilling with milling attachments introducing in market. Present paper is a partial fulfillment to the present market needs to understand the variations in machine design accessories.Gear design and gear ratios are theoretically calculated and the whole gear alignment - performed in design software CATIA V5 R20 Simulations and stress analysis and torque generation graphs have been generated by using ANSYS work bench 14.5 or 15.0. Simulations on each material will be submitted with torque applications and performance comparison with before designs of general purpose machines. The alignment will be in two stages with torque increasing for drilling and torque decreasing for milling operation, alignment of gear box will be changes by using lever.

Keywords: Multi Spindle, working of multi spindles Flange, CATIA V5 R20 ANYSIS 14.5 or 15.0

INTRODUCTION

INTRODUCTION OF GEARS

A radial load, or overhung load (OHL) as it is also called, is a bending force imposed on a shaft due to the torque transmitted by belt drives, chain drives, or gears. Radial forces can also be created by belt or chain tension and by a misaligned shaft coupling.

The purpose of a gear reduction system is to convert input an speed and torque into a different output speed and torque. The design at hand requires the use of two gears whose diameters are specified at 24 and 12 inches each. These gears are attached to a shaft whose diameter is specified at two inches, and the bearings, keys, gears, speeds, safety factors, etc need to be determined from statics, strengths, fatigue, and various other design considerations.Along with torque overloading and shock loading, excessive radial loading (overhang load) is one of the top reasons for gearboxes fail. It is also one of the least considered elements when integrating speed reducers with gear, belt, and sprocket drive systems for tools, rolling mills and transmitting machinery. Toothed gears are used to change the speed and power ratio as well as direction between an input and output shaft. Gears are the most common means of transmitting motion and power in the modern mechanical engineering world. They form vital elements of mechanisms in many machines such as automobiles, metal cutting machine

1.2 TYPES OF GEARS:

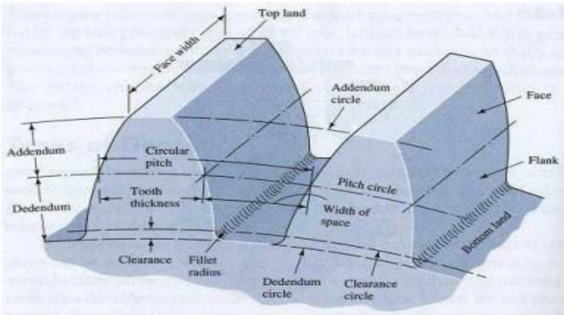
There are many different types of gears used by industry, but all these gears share the same purpose, which is to transmit motion from one shaft to another. Generally, gearing consists of a pair of gears with axes are either parallel or perpendicular. Among all the gears in the world, the four most commonly discussed gears are spur gear, helical gear, bevel gear, and worm gearing. Worm gearing consists of the worm and worm gear. Depend upon the rotation direction of the worm; the direction of rotation of the worm gear would be different. The direction of rotation also depends upon whether the worm teeth are cut left-hand or right-hand. In general, worm gear sets are more efficient when the speed ratios of the two shafts are high. Basically, in worm gearing, higher speed equals to better efficiency. The following figure demonstrates the four most common types of gears in industry.

Fig 1.1 types of gears

Manufacturing processes A number of ways can be used to manufacture the shape of the gear teeth; however, they can be classified into two categories – Forming and Generating. In forming processes, the tooth space takes the exact form of the cutter. On the other hand, generating is a process that uses a tool having a shape different from the tooth profile which is moved relative to the gear blank as to obtain the proper tooth shapeMilling – a form milling cutter will be used to conform the tooth space. The tooth form is produced by passing the milling cutter with the appropriate shape through the blank. Hobbing – one of the fastest ways of cutting gears. The hob basically is a cutting tool that is shaped like a worm. As the hob rotates and feeds along the gear axis, the gear rotates about its axis in a carefully controlled environment. Lapping is applied to heat treated gears to correct small errors, improve surface finish, and remove nicks and burrs.

TERMINOLOGY:

The first step of learning gear design is to know the basic terminology of the gear. Since spur gears are the most common form of gearing, it will be used to illustrate the nomenclature of gear teeth. The following figure displays the nomenclature of spur gear teeth.



Nomenclature of spur gear teeth

One of the most important parameters on the gear teeth is the pitch circle since all calculations are based on this theoretical circle. The diameter of the pitch circle is called the pitch diameter d. When a pair of gears is mated together, the pitch circles of the gears are tangent to each other. The circular pitch p is the distance on theThe diametric pitch P is the ratio of the number of gear teeth to each inch of the pitch diameter. The module m is the ratio of the pitch diameter to the number of teeth, and the unit of module is usually millimeter. Hence,

$$P = \frac{N}{d}$$

$$m = \frac{d}{N}$$

$$p = \frac{\pi d}{N} = \pi m$$

Where,

N = Number of teeth

p = Circular pitch

P = Diametral pitch,

teeth per inch d = Pitch diameter, inch

m = Module, mm

d = Pitch diameter, mm

LITERATURE SURVEY

In 2001, Howard [34] simplified the dynamic gear model to explore the effect of friction on the resultant gear case vibration. The model which incorporates the effect of variation in gear tooth torsional mesh stiffness, was developed using finite element analysis, as the gears mesh together. The method of introducing the frictional force between teeth into the dynamic equations is given in his paper. The comparison between the results with friction and without friction was investigated using Matlab and Simulink models developed from the differential equations

In 2003, Wang [1] surveyed the nonlinear vibration of gear transmission systems. The progress in nonlinear dynamics of gear driven system is reviewed, especially the gear dynamic behavior by considering the backlash and time-varying mesh stiffness of teeth. The basic concepts, the mathematical models and the solution methods for non-linear dynamics of geared systems were all reviewed in his paper.

In 1991, Lim and Singh [2] presented study of the vibration analysis for complete gearboxes. Three example cases were given there: a single-stage rotor system with a rigid casing and flexible mounts, a spur gear drive system with a rigid casing and flexible mounts, and a high-precision spur gear drive system with a flexible casing and rigid mounts. In 1994, Sabot and **Perret-Liaudet [3]** presented another study for noise analysis of gearboxes. A troublesome part of the noise within the car or truck cab could be attributed by the transmission error which gives rise to dynamic loads on teeth, shafts, bearings and the casing. During the same year, a simulation method by integrating finite element vibration analysis was developed by others. Each shaft was modeled as a lumped mass and added to the shaft in their model. Each of the rolling element bearings was represented as a spring and damper. The casing of the gearbox was modeled by a thin shell element in the finite element package program.

In 1999, Kelenz [4] investigated a spur gear set using FEM. The contact stresses were examined using a two dimensional FEM model. The bending stress analysis was performed on different thin rimmed gears. The contact stress and bending stress comparisons were given in his studies.

Randall and Kelley [5] modifications have been made to Sweeney's basic model to extend it to higher quality gears where the tooth deflection component is more 15 important. The tooth deflection compliance matrix and the contact compliance vector have been derived using finite element models. The effects on the transmission error of the variation of the tooth body stiffness with the load application point have been investigated, and a simulation program for transmission error (TE) computation with varying stiffness has been developed. In order to study the case where the tooth deflection component is the dominant source of the transmission error nylon gears were used. All the simulation results have been compared with the measured transmission errors from a single-stage gearbox.

In 1996, Sweeney [6] developed a systematic method of calculating the static transmission error of a gear set, based on the effects of geometric parameter variation on the transmission error. He assumed that the tooth (pair) stiffness is constant along the line of action (thin-slice model) and that the contact radius for calculation of Hertzian deformation is the average radius of the two profiles in contact. Sweeney's model is applicable to cases where the dominant source of transmission error is geometric imperfections, and is particularly suited to automotive quality gear analysis. The results of his model gave very good agreement with measurements on automotive quality gears.

In 1979 Mark [7] analyzed the vibration excitation of gear systems. In his papers, formulation of the equations of motion of a generic gear system in the frequency domain is shown to require the Fourier-series coefficients of the components of vibration excitation. These components are the static transmission errors of the individual pairs in the system. A general expression for the static transmission error is derived and decomposed into components attributable to elastic tooth deformations and to deviations of tooth faces from perfect involute surfaces with uniform lead and spacing.

METHODOLOGY

METHOD OF GEAR DESIGN

New advances in computer technology have made finite element stress analysis a routine tool in design process has given rise to computer-aided design (CAD) using solid-body modeling. Some benefits of CAD are productivity improvement in design, shorter lead times in design, more logical design process & analysis, fewer design errors, greater accuracy in design calculations, standardization of design, more understandability and improved procedures for engineering changes.

FINITE ELEMENT ANALYSIS (FEA)

It is widely accepted method of accessing product performance without the need for physical building and testing. It also shortens prototype development cycle times & facilitates quicker product launch. FEA consists of a computer model of a material or design that is loaded and analyzed for specific results. It is used in new product design, and existing product refinement.

Advantages of FEA

1. The inherent advantages of finite element analysis are as under:
2. Easy to model irregular shapes
3. Possible to evaluate different materials
4. Can apply general load conditions
5. Large numbers and kinds of boundary conditions are possible in FEA
6. Different sizes of elements can be used where necessary
7. Cheap and easy
8. Dynamic effects, nonlinear behaviors and nonlinear materials can be examined
9. Reduce the number of prototypes required in the design process

STEPSREQUIREDFOR DEVELOPMENT OF FEA MODEL

Steps required for development of finite element model are as under:

1. Assigning material and its properties to various parts.

2. Discretize and choose element types.

3. Choose a displacement function.

4. Derive the element stiffness matrix and equations.

5. Generate global or total equations from the element equations and introduce loads and boundary conditions.

6. Solving for elemental strains and stresses and interpretation of the model

Role of CAD & Solid Modeling in Gear

CONTACTPROBLEM CLASSIFICATION

There are many types of contact problems that may be encountered, including contact stress, dynamic impacts, metal forming, bolted joints, crash dynamics, assemblies of components with interference fits, etc. All of these contact problems, as well as other types of contact analysis, can be split into two general classes (ANSYS), Rigid - to - flexible bodies in contact, Flexible - to - flexible bodies in contact. In rigid - to - flexible contact problems, one or more of the contacting surfaces are treated as being rigid material, which has a much higher stiffness relative to the deformable body it contacts. Many metal forming problems fall into this category. Flexible-to-flexible is where both contacting bodies are deformable. Examples of a flexible-to-flexible analysis include gears in mesh, bolted joints, and interference fits

MODELING IN UG:

Siemens NX/Solidedge, and Solidworks..

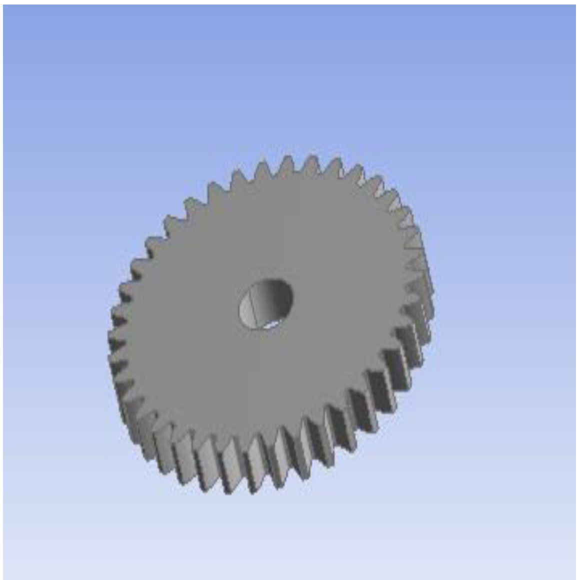


Fig 3.1 Shifter gear 3

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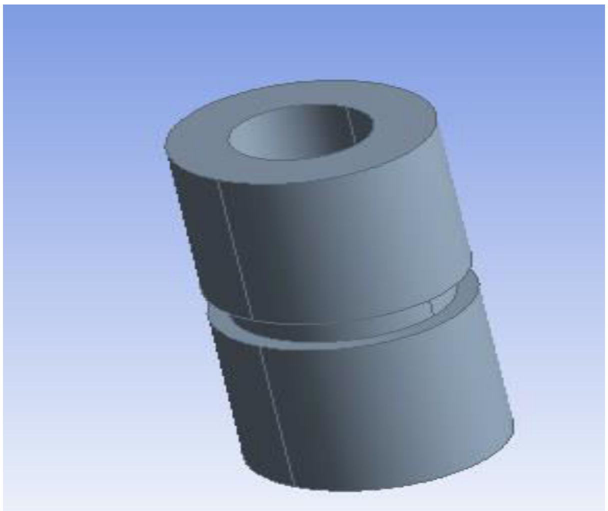

Fig 3.2 Shifter spacer

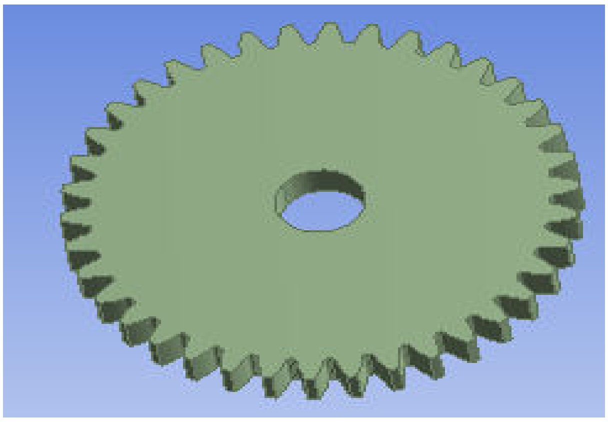

Fig 3.3 Shifter spacer

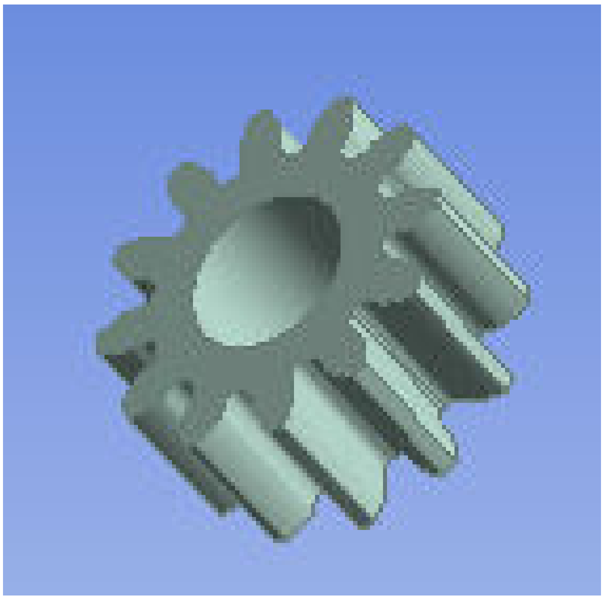

Fig 3.4 Shifter spacer

Fig 3.5 Shifter spacer

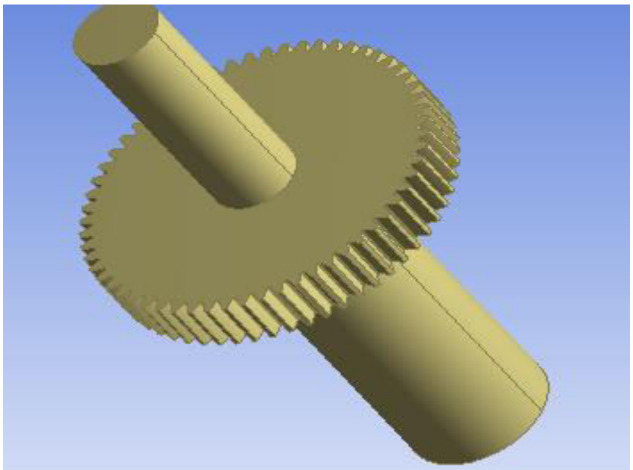

Fig 3.6 Driver gear

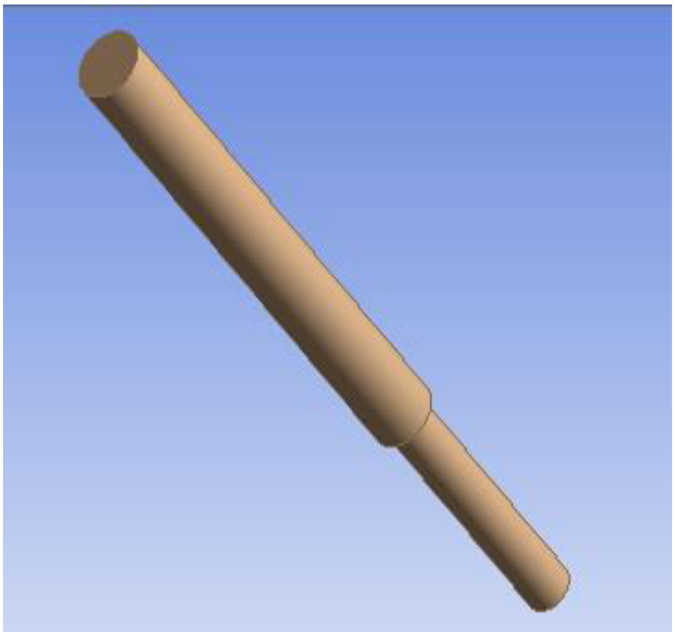

Fig 3.7 Shifter

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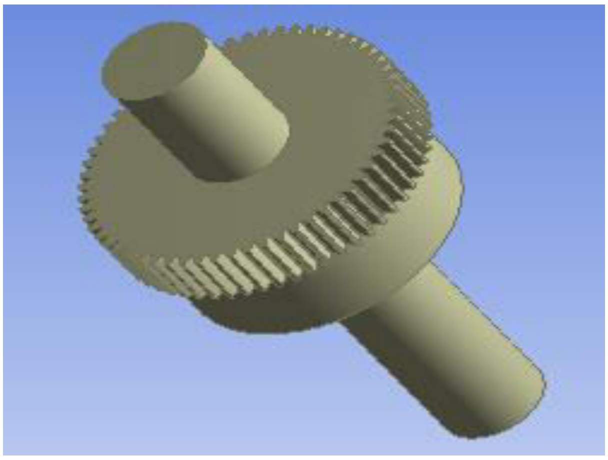
Fig 3.8 Drum gear

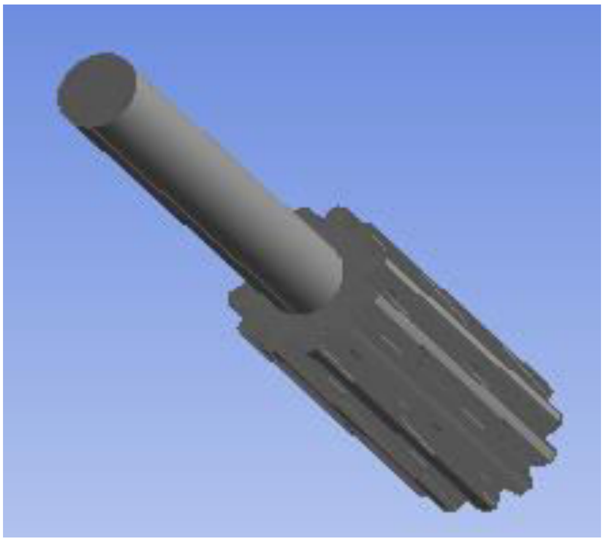
Fig 3.9 Splane 2

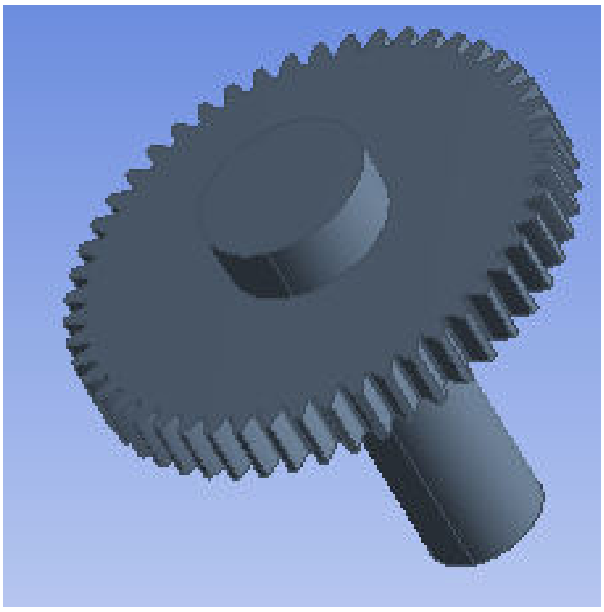
Fig 3.10 Motor shaft gear

RESULTS AND DISCUSSIONS

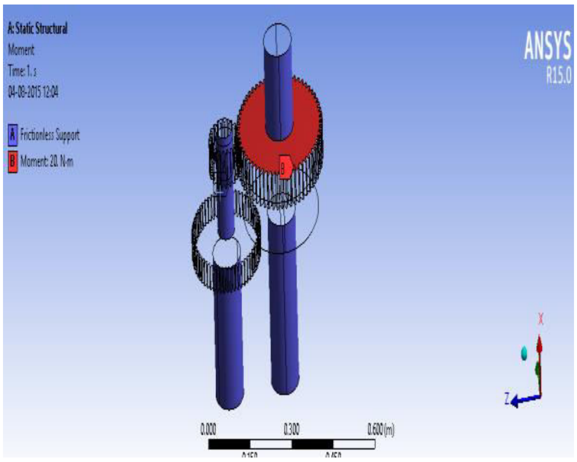

Fig 4.1 Boundary conditions for single gear system

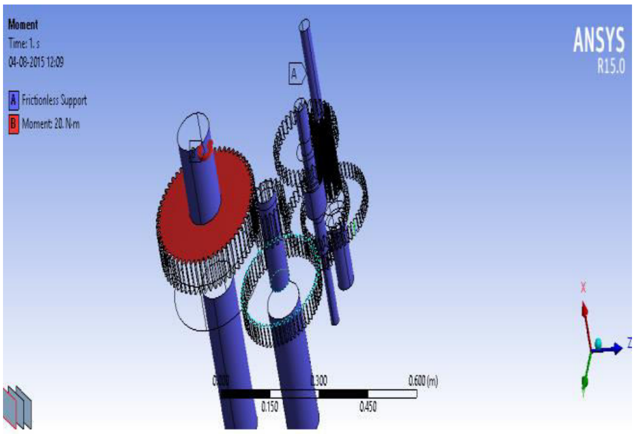

Fig 4.2 Boundary conditions for assembly gear system

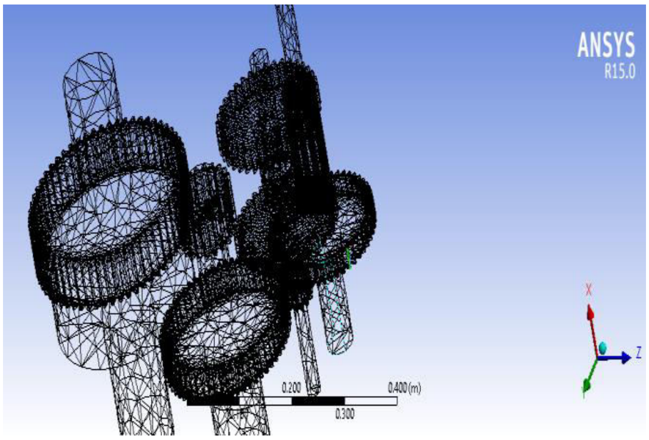

Fig 4.3 Meshed Model of gear assembly

Since a ratio is just another way to express a fraction, we can also write the gear ratio as:

GR=Output/Input = Driven/driver

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ANALYSIS OF STRESSES

The below figure shows the equivalent stresses acting on the contact edge of the gears maximum of 2.425e-6Pa was found on the edge.

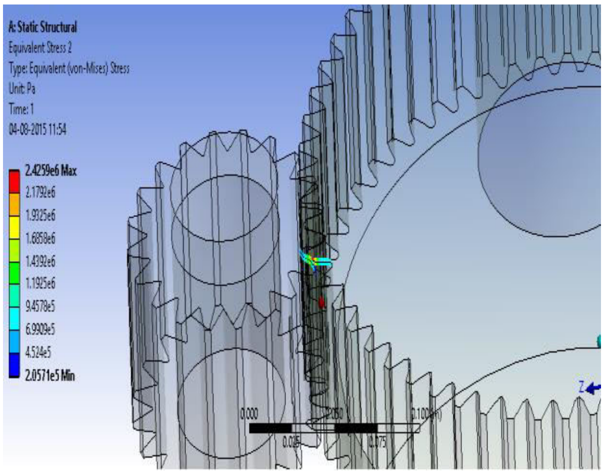

Figure 4.4 Vonmisses stress using problem

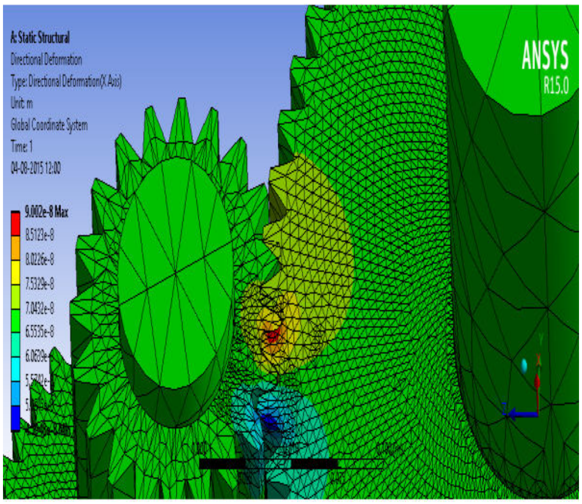

Figure 4.5 Directional deformations along x axis

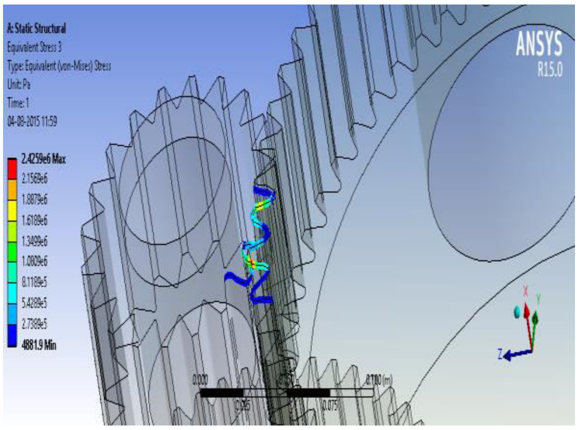

Figure 4.6 Equivalent Con misses stress

5.0 CONCLUSIONS AND FUTURE SCOPE

Torque calculations shows the different torques generates after aligning different gears .Operations of milling and drilling both are operated at the time of changing gear alignment. For the torque changing requirements for drilling and milling shifter shaft has been introduced to change the torque for both operations.

Torque is analyzed by finite element analysis byusing ANSYS work bench 15.0. The values and graphs obtained by analysis shows that the deformation with the shifter is very less and the torque generated with attachment of shifter gear is high and when it is disconnected torque is low for milling operation. This project shows an edge for both the operations which is useful for production.

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