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A Comprehensive Study Optimization and Analysis of Gear Alignments for Special Purpose Radial Milling Machine

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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. Gear design and gear ratios are theoretically calculated and the whole gear alignment performed in design software creo 3.0. Simulations and stress analysis and torque generation graphs have been generated by using ANSYS work bench 15.0. L18 taguchi orthogonal array on torque generated and spindle RPM on these cases has been submitted. 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.

Total design will be analyses by using finite element analysis in terms of torque applicable at each point with different speed ratios on different materials. Simulations on each material will be submitted with torque applications and performance comparison with before designs of general purpose machines.

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

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.

Spur gears considered as the simplest form of gearing, and they consist of teeth parallel to the axis of rotation. The common pressure angles used for spur gears are 141 /2, 20, and 25 degrees. One of the advantages of a low pressure angle is smoother and quieter tooth action. In contrast, larger pressure angles have the advantages of better load carrying capacity.

Helical gears consist of teeth that are cut at an angle and inclined with the axis of rotation. Helical gears essentially have the same applications as spur gears. However, because of their gradual engagement of the teeth during meshing, helical gears tend to be less noisy. In addition, the inclined tooth develops thrust loads and bending couples, which are not present in the spur gear.

Bevel gears teeth are formed on conical surfaces and unlike spur and helical gears, bevel gears are used for transmitting motion between intersecting shafts not parallel shafts. There are different types of bevel gears, but all of them establish thrust, radial, and tangential loads on their support bearings.

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.

Objectives of present work

1. Conversion of normal gear box to multiple using radial gear box
2. Gear data of each gear depend on co axiality of different gear alignments
3. Gear data taken for spur gears with matching of ratios for torque generation.
4. Comparisions from normal torque to radial torque appearances for drilling and milling applications.

LITERATURE REVIEW:

1. Kelenz [1] 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.
2. Randall and Kelley [2] 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.

3. Sweeney [3] 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.
4. Mark [4] 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.
5. Kasuba [5] determined dynamic load factors for gears that were heavily loaded based on one and two degree of freedom models. Using a torsional vibratory model, he considered the torsional stiffness of the shaft. In 1981, he published another paper [48]. An interactive method was developed to calculate directly a variable gear mesh stiffness as a function of transmitted load, gear profile errors, gear tooth deflections and gear hub torsional deformation, and position of contacting profile points. These methods are applicable to both normal and high contact ratio gearing. Certain types of simulated sinusoidal profile errors and pitting can cause interruptions of the normal gear mesh stiffness function, and thus, increase the dynamic loads. In his research, the gear mesh stiffness is the key element in the analysis of gear train dynamics. The gear mesh stiffness and the contact ratio are affected by many factors such as the transmitted loads, load sharing, gear tooth errors, profile modifications, gear tooth deflections, and the position of contacting points.
6. Sirichai [6] has developed a finite element analysis and given a definition for torsional mesh stiffness of gear teeth in mesh. The combined torsional mesh stiffness is defined as the ratio between the torsional load and the angular rotation of the gear body. The development of a torsional mesh stiffness model of gears in mesh can be used to determine the transmission error throughout the mesh cycle.
7. Chong and Bar [7] demonstrated a multiobjective optimal design of cylindrical gear pairs for the reduction of gear size and meshing vibration. The results of the relation between the geometrical volume and the vibration of a gear pair were analyzed, in addition a design method for cylindrical gear pairs to balance the conflicting objectives by using a goal programming formulation was proposed. The design method reduces both the geometrical volume and the meshing vibration of cylindrical gear pairs while satisfying strength and geometric constraints.
8. Tuplin [8] suggested that the number of stress cycles causing failure of a given material under any particular stress is dependent of the time-rate of repetition of stress. High-speed gears

have failed under stresses lower than the fatigue limit so it becomes necessary to consider whether the actual stress was as low as had been assumed. Pitch errors and profile variation in gear teeth cause actual stresses to be higher than nominal stresses. The nominal permissible stress (corresponding to the mean transmitted torque) should therefore consider probable errors in the teeth. So spring mass model of mating gears is developed. Equivalent stiffness was calculated by considering individual stiffnesses. Dynamic loads were approximated by considering various types of errors.

9. Houser et al. [9] investigated dynamic factors for spur and helical gears. Comprehensive program was developed for spur and helical to investigate the influence of errors and variation in mesh stiffnesses on peak stresses. Four sets of specially designed gears were tested. Tooth loads and system shaft torques at various operating conditions were compared.

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

Steps Required For 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 Design and Analysis

CAD techniques give the design engineer a powerful tool for graphical and analytical tasks. Modern CAD systems are based on ICG in which the computer is employed to create, transform and display geometric data. CAD helps in :-

Creating conceptual product models

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Editing, refining the model to improve aesthetics, ergonomics and performance
Analyze stress, static deflection and dynamic behavior for different mechanical and thermal loading configurations and carry out quickly any necessary design modifications to rectify deficiencies in the design. ÿ Study the product from various aspects such as material requirements, cost, value, value engineering, manufacturing processes, standardization, simplification, variety reduction, service life, servicing and maintenance aspects

Contact Problem Classification

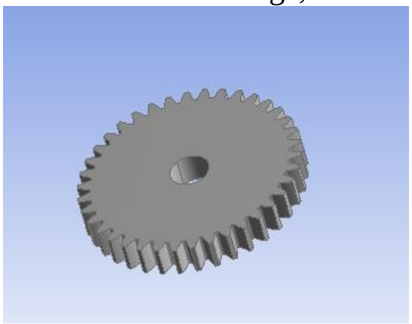
There are many types of contact problems that may be encountered, including contact stress, dynamic impacts, metal forming, bolted joints, crash dynamics, assemblies of components with interference fits, etc. All of these contact problems, as well as other types of contact analysis, can be split into 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.

Types of Contact Models

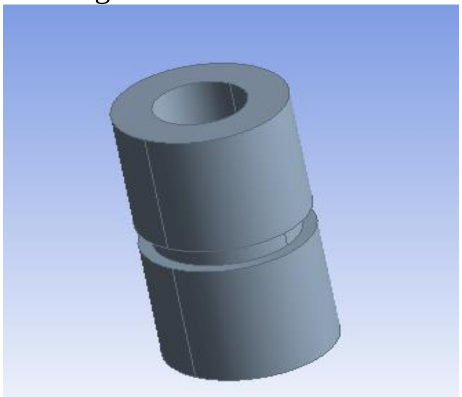
In general, there are three basic types of contact modeling application as far as ANSYS use is concerned. Point-to-point contact: the exact location of contact should be known beforehand. These types of contact problems usually only allow small amounts of relative sliding deformation between contact surfaces. Point-to-surface contact: the exact location of the contacting area may not be known beforehand. These types of contact problems allow large amounts of deformation and relative sliding. Also, opposing meshes do not have to have the same discretisation or a compatible mesh. Point to surface contact was used in this chapter. Surface-to-surface contact is typically used to model surface-to-surface contact applications of the rigid-to-flexible classification.

Flowchart for Structural (Static) Analysis Procedure

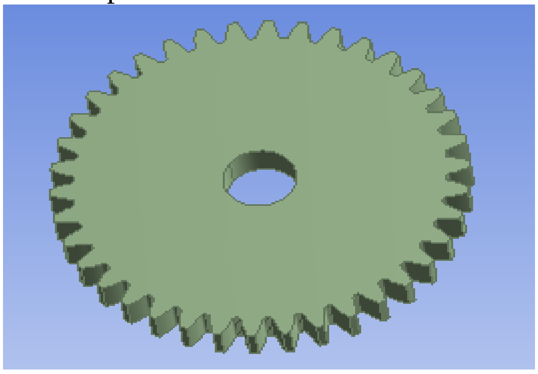
UG runs on Microsoft Windows and provides apps for 3D CAD parametric feature solid modeling, 3D direct modeling, 2D orthographic views, Finite Element Analysis and simulation, schematic design, technical illustrations, and viewing and visualization.
Siemens NX/Solidedge, and Solidworks..



Shifter gear 3



Shifter spacer



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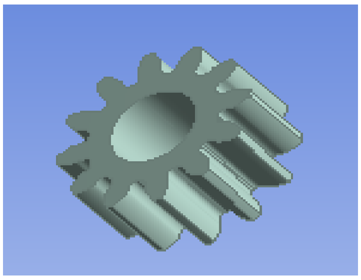
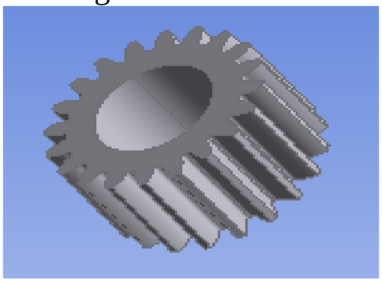
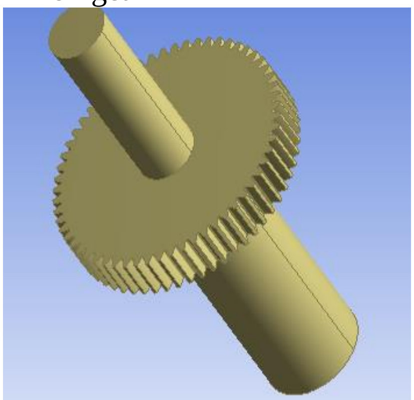
Shifter gear 2



Shifter gear 1



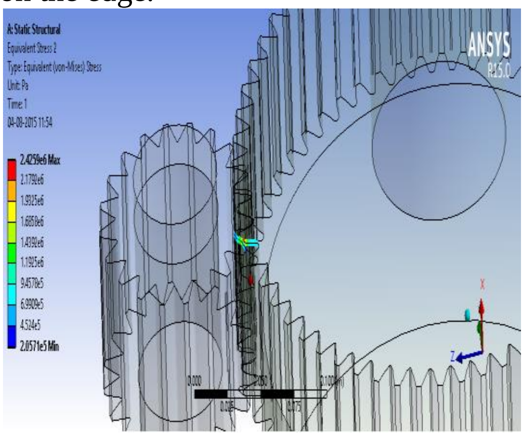
Pinion gear 1



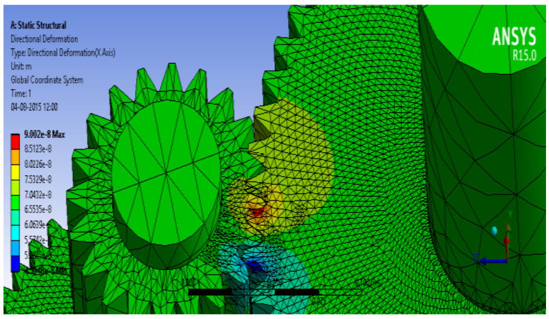
Driver gear

RESULTS AND DISCUSSIONS

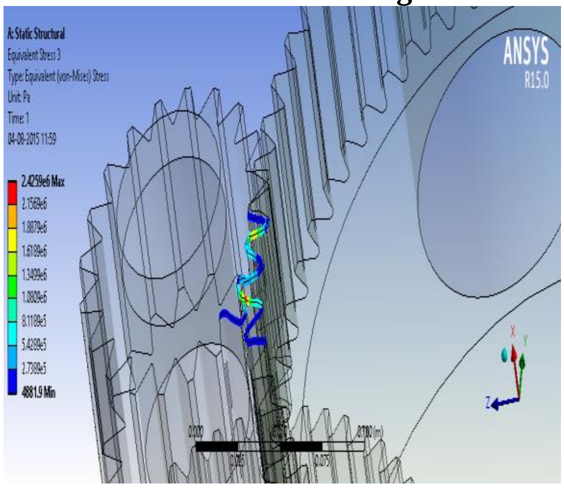
The below figure shows the equivalent stresses acting on the contact edge of the gears maximum of 2.425e-6Pa was found on the edge.



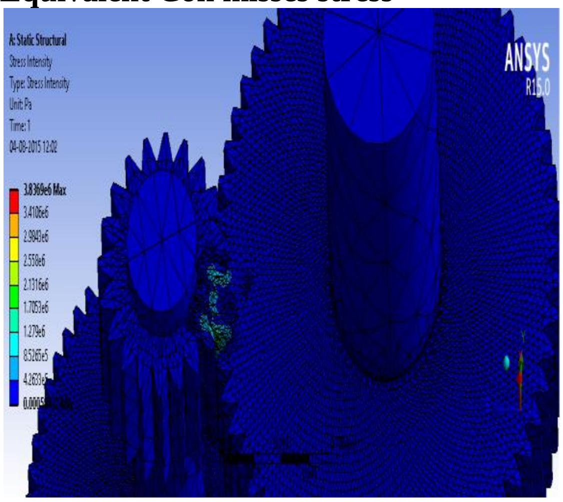
Vonmisses stress using probe



Directional deformation along x axis



Equivalent Con misses stress



Stress intensity on the contact region

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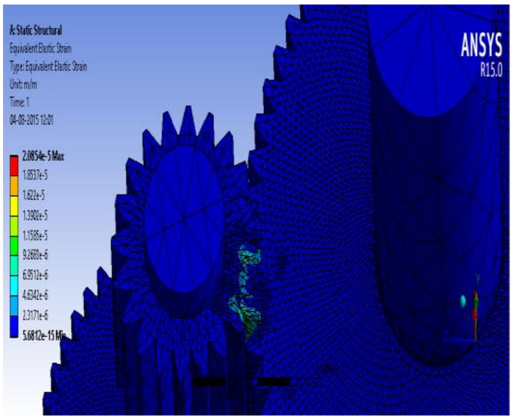
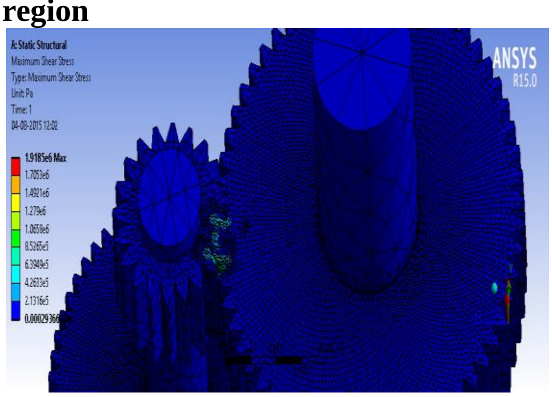
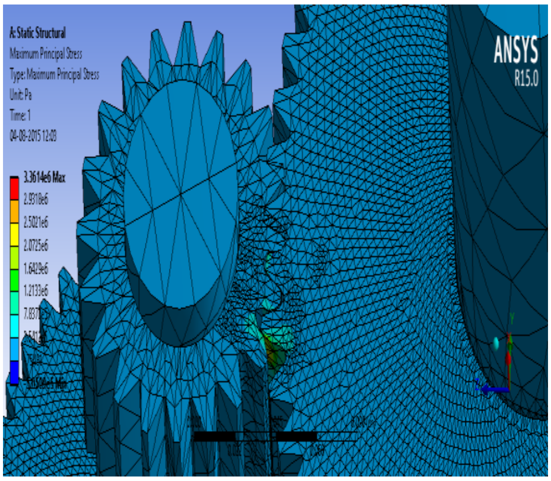


Equivalent elastic strain on the contact region



Maximum shear stress

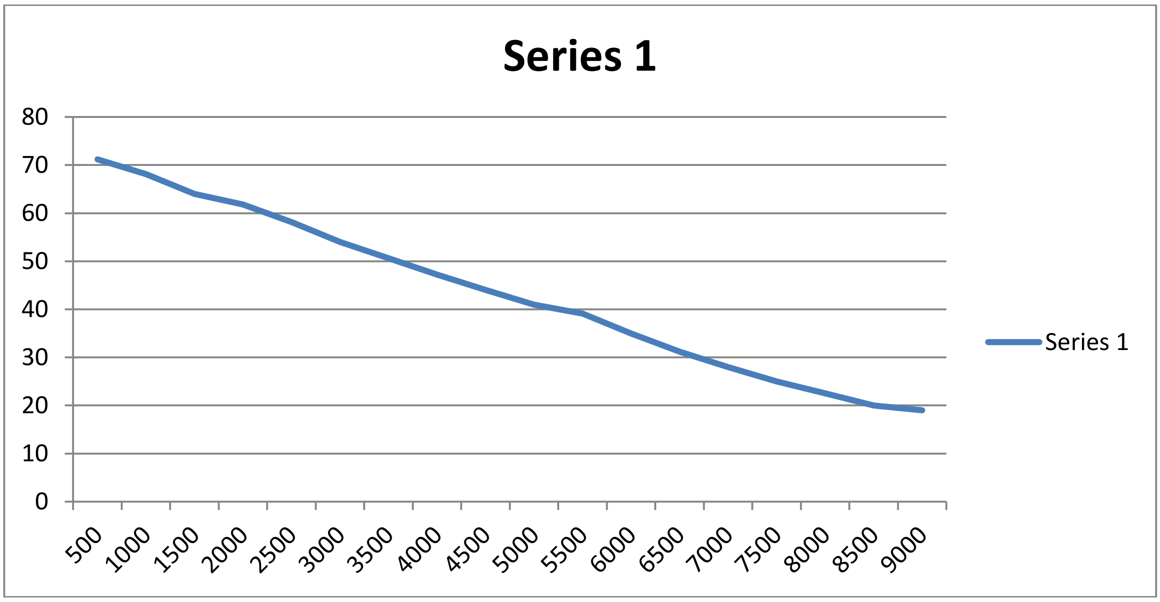


Maximum principal stress

TORQUE CALCULATION TABLE
VECTOR DUAL DRIVE 20HP & 10000RPM MAXIMUM

S.NO	SPEED RPM	TORQUE	TORQUE USED FOR	DRILLINGM/C ACTUALS	MILLINGM/C
1	500	71.2	DRILLING	74	61
2	1000	68.15	DRILLING	69.4	58.5
3	1500	64	DRILLING	65.2	56
4	2000	61.8	DRILLING/MILL	63.1	53
5	2500	58.15	DRILLING/MILL	59	51.2
6	3000	54	DRILLING/MILL	55	49
7	3500	50.6	MILL	NA	46.5
8	4000	47.2	MILL	NA	43

9	**4500**	**44**	**MILL**	**NA**	**41.5**
10	**5000**	**41**	**MILL**	**NA**	**38**
11	**5500**	**39.1**	**MILL**	**NA**	**36**
12	**6000**	**35**	**MILL**	**NA**	**32.3**
13	**6500**	**31.2**	**MILL**	**NA**	**28.6**
14	**7000**	**28**	**MILL**	**NA**	**25.1**
15	**7500**	**25**	**MILL**	**NA**	**22.6**
16	**8000**	**22.5**	**MILL**	**NA**	**21**
17	**8500**	**20**	**MILL**	**NA**	**19.2**
18	**9000**	**19**	**MILL**	**NA**	**18**



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 by using 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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