Design and analysis of a multi spindle drill gearbox

Torque divider

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Abstract—Gearboxes play an important role in many applications. One of the important applications is in multi spindle drilling machine. So, it is necessary to study and analyze the gearbox. In general, gear analysis includes calculations of the tooth stresses, failures due to bending, wear or scoring by considering both the beam strength and fatigue strength. In the multi spindle drilling machine, smooth functioning of the gearbox is of utmost importance. This paper gives the design and analysis of the gearbox of a special purpose machine for drilling application. In this paper, bending stress analysis will be performed on different size spur gears to investigate the weakest portion against bending. Then bending stresses calculated using ANSYS 15 workbench, are compared to the results obtained from existing or theoretical methods.

Keywords- *Beam strength,multi spindle ,Bending stress, fatigue strength, ANSYS 15.*

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I. INTRODUCTION

Gearbox, speed reducer or a torque divider as it is variedly called is a major part of various machines. A gearbox needs to be given prime focus when it comes to a special purpose machine. SPM gives the highest productivity and is considered as the most economic production method. These advantages include mass production of parts in shorter time, high accuracy of products, uniformity and repeatability of production, elimination of some quality control steps, simultaneous machining of a number of parts, and reduced labour and overhead costs. Hence, for continuous operations we need to design the gearbox for its strength and safety.

There are several failure mechanisms for spur gears. Bending failure and pitting of the tooth are the two main failure modes in a transmission gearbox. Pitting of the teeth is usually called a surface failure. This was already discussed in the last section. The bending stresses in a spur gear are another interesting problem. When loads are too large, bending failure will occur. Bending failure in gears is predicted by comparing the calculated bending stress to experimentally-determined allowable fatigue values for the given material. This bending stress equation was derived from the Lewis formula. Wilfred Lewis (1892) was the first person to give the formula for bending stress in gear tooth using the bending of a cantilevered beam to simulate stresses. On the basis of the Hertz and the Lewis theory of stresses, the analytical mechanics was studied by Buckingham, E.(1949). Later, the effect of contact ratio on bending and dynamic loading was analysed by Chuen-Huei Liou and Hsiang Hsi Lin (1992).He then proved that a good contact ratio results in better transmission. Later, Stresses and deformations in involute spur gears by finite element method was done by Zeping wei (2004).His work focussed on the generation of involute profile of gears and its analysis .He concluded that involute profile reduces the contact stresses and has greater transmission efficiency. Design and Structural Analysis of High Speed Helical Gear Using ANSYS was done by J. Venkatesh (2014). In this work bending stresses were calculated by using modified Lewis beam strength equation and Ansys software. It concluded that Error percentage is around 6% in bending stresses and around 1 % in contact stress analysis. Mrs.Shinde S.P did the Static Analysis of Spur Gear Using Finite Element Analysis. In that work, a 3D deformable-body (model) of spur gears was developed. The simulation results had good agreement with the theoretical results, The model was applied onto commercial FEA software ANSYS. It was found out that the numerically obtained values of stress distributions were in good agreement with the theoretical results. Recently, generation of the involute profile for proper meshing and its analysis was done by Vishwjeet V.Ambade (2013). Attainment of predetermined centre distance to increase the strength at the root and flank of the tooth was analysed in ANSYS.

Thus from the previous researches, it is understood that the analaysis of two gears is done with existing model. The Hertzian stresses were verified with analytical results. There has never been an analysis done on different sizes gears of the same gear box and more number of gears in mesh. In this paper, modeling of different sizes gears and the analysis on ANSYS 15 workbench with as much as five gears is done. The torque divider in this paper has a gear ratio of 1:1 which simply divides the torque with a constant speed of 800rpm. The main steps involved in this work are as follow:

1. Modelling the gear in CATIA V5R21 software.
2. Calculating the beam strength and checking the gear for safety with Lewis theory.

3. Develop and determine models of contact elements, to analyze contact stresses using ANSYS 15 software.

4. Comparing the theoretical results and results obtained from ANSYS 15.

II. GEARBOX MODELLING

The drilling is to be done on a component to produce simultaneously eight holes. Considering the space constraints, the diameter and other parameters of the gears were found. The gear train used for transmission is a compound gear train.

From the prime mover, the power is transmitted to the spindles through gearbox. It is thus very important to analyze the stresses on the gears. The three larger gears are identical and then the torque gets equally divided to the eight gears. Hence for the small values of the torque and the smaller dimensions, speed is comparatively high. So, separate analysis is to be done of the larger as well as the smaller gears.

Both the gears are been modeled on the CATIA V5R21 software. The material of the gear is gray cast iron. The part drawing is separately made with the help of parameters of gears. The profile is drawn in the software with parametric dimensions as the pitch diameter, face width, module. The addendum, dedundum and the backlash are obtained from the empherical relations. These gears are assembled in the assembly workbench of CATIA software. Each gear pair is given the constraints of tangency and surface coincidence. The small gears are mounted on the spindle shafts.

PARAMETERS	LARGER GEAR	SMALLER GEAR
T =Torque	13 Nm	3.25 Nm
d = Diameter	60 mm	24 mm
F = Face width	20mm	20mm
m = Module	2mm	2mm
Z = No. Of teeth on the gear	32	14
K_m=Load distribution factor	2	2
Ka= Application Factor	1.5	1.5
K= Velocity Factor	$\frac{3}{3+V}$	$\frac{3}{3+V}$

Table.1 Parameters for modelling

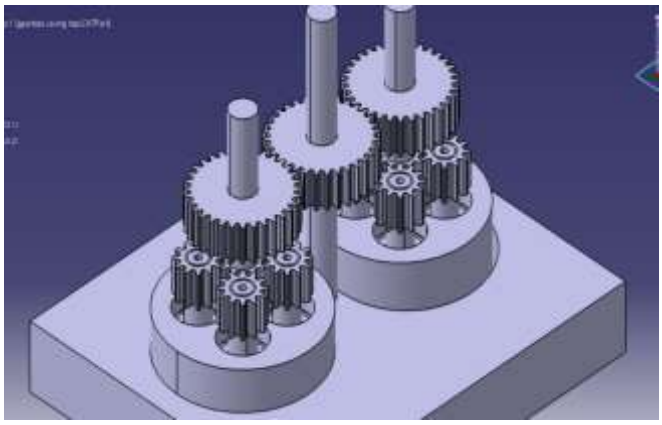


Fig.1 Model of the gearbox on CATIA.

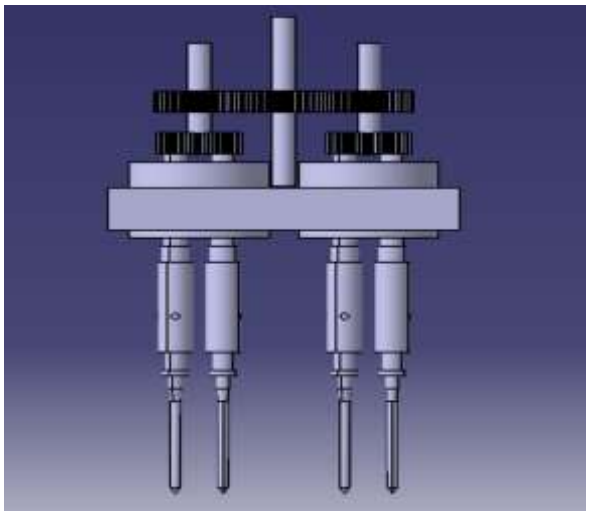


Fig.2 Spindle attachment

III. BENDING STRESS CALCULATION

Beam strength

The gears are checked to resist bending stresses of their bending stresses.There is a problem of failures at the root of the teeth because of the inadequate bending strength and surface pitting. This can be avoided or minimized by proper method and modification of the different gear parameters. There are two theoretical formulas, which deal with these two fatigue mechanisms. One is the Hertz equation, which can be used to calculate the contact stresses and other is the Lewis formula, which can be used to calculate the bending stresses.

1. Lewis form factor

$$Y = 0.484 - \frac{2.865}{Z}$$

2. Beam strength

$$F_b = \sigma_b y.m.b$$

3. Buckingham's equation

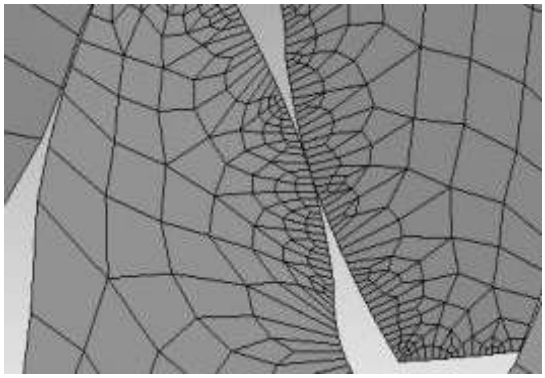
$$F = \frac{21V(bC+F_{t\max})}{21V+\sqrt{bC+F_{t\max}}}$$

From the calculations, we get the values as
$F_b = 993.6$ N
$F_{eff} = 734.1$N
As $F_b > F_{eff}$
Also $\sigma_b = 66.67$ mpa for Z= 32.

It is concluded that the design is safe for bending stresses of beam strength.

IV. ANALYSIS OF THE GEARBOX

Fatigue or yielding of a gear tooth due to excessive bending stresses are two important gear design considerations. In order to predict fatigue and yielding, the maximum stresses on the tensile and compressive sides of the tooth, respectively, are required. In the past, the bending stress sensitivity of a gear tooth has been calculated using photo elasticity or relatively coarse FEM meshes. However, with present computer developments we can make significant improvements for more accurate FEM simulations.



A. *Larger gears*

In the present gearbox problem, we export the modelled drawing from CATIA software with the .igs extension and do its analysis in the ANSYS 15 software. Initially geometry and parts are defined. The boundary condition and loading condition should be specified. At the starting condition the load (F_t) is applied on the gear teeth and the amount of load transfer from one gear to other gear. At the starting condition the amount of torque is high than at the running condition.

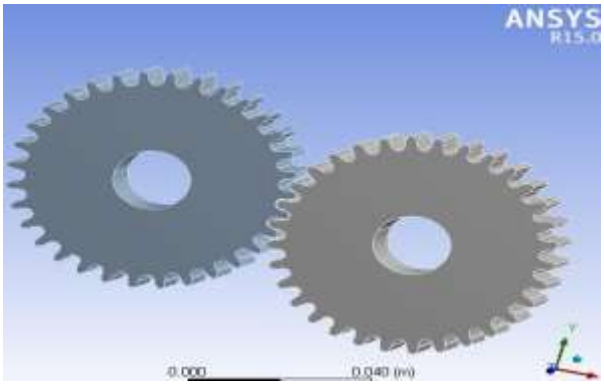


Fig.3 Meshing gears in ANSYS 15

The area defining as the contact region of the gears is taken as the area or the surface where the tangential force is going to act. The contacting surface is taken as frictionless surface or support. Again the torque is applied to the gears and the results are obtained.

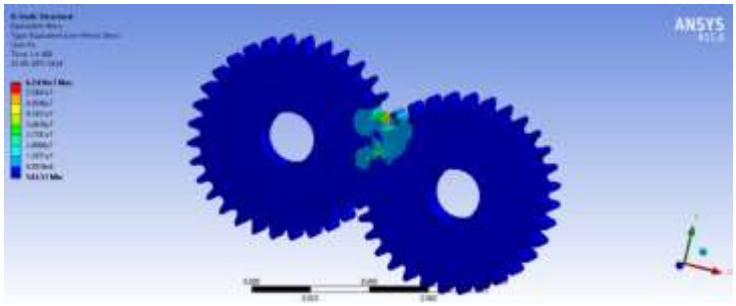


Fig.4 Contact stresses on the gears

From the figure it is clear that the minimum stress on the gear surface is 0.00543mpa. Minimum stress at the root is 6.9mpa and the maximum stress goes to 62.4mpa. The following figure reflects that the stress distribution is within the range and is permitted.

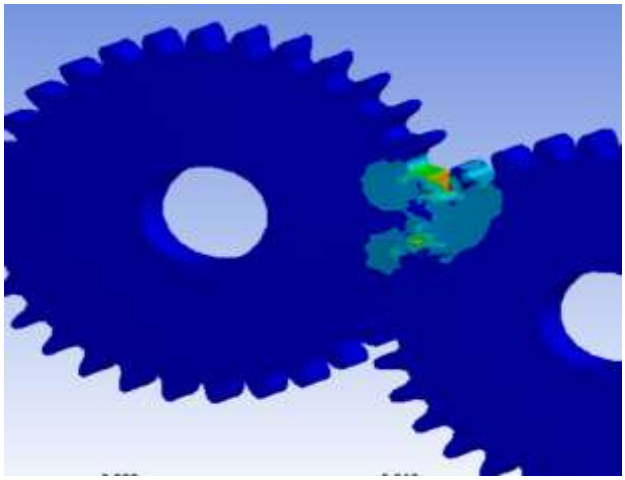


Fig.5 Stress distribution on the gears

The deformation analysis is also done of the gears to know the maximum deformation.

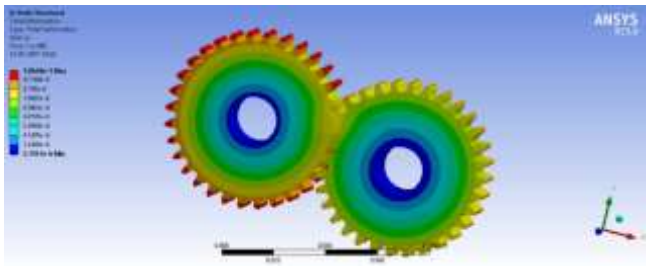


Fig.6 Deformation of the gears

It has been found that deformation is maximum at the top land of face width. This indicates that top land area of spur gear is under high temperature and pressure due to which deformation is maximum at its top land. Here the deformation is 0.0106mm we can say that Analytical deformation is in permissible limits.

B. Smaller gears

The torque is then transmitted to the smaller gears. The stress distribution is in similar manner as that of larger gear.

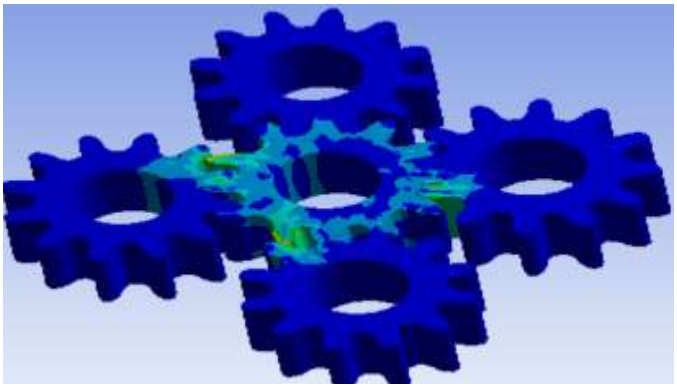


Fig.7 Stress distribution on the five gears

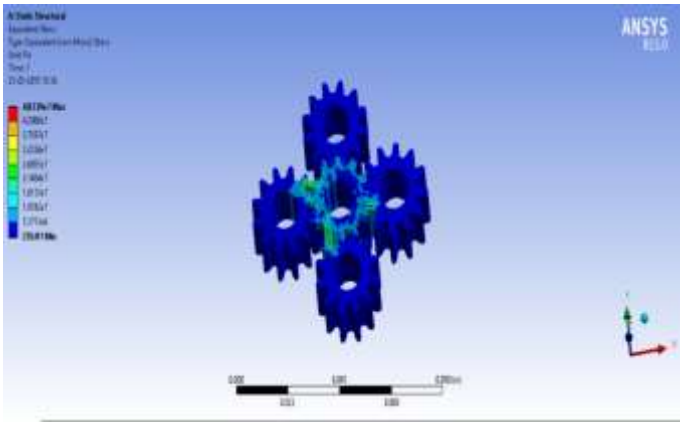


Fig.8 Contact stresses on the gears

From the figure it is clear that the minimum stress on the gear surface is 0.002508 mpa. The maximum stress goes to 62.4mpa. The figure shows that the stress distribution is within the range and is permitted and apart from the root and tooth, the stresses are negligible.

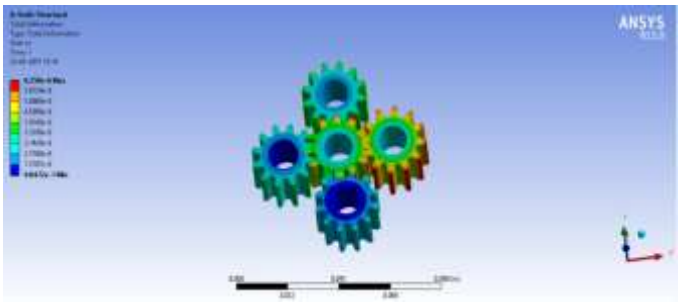


Fig.9 Deformation of the gears

V. RESULTS

The results obtained from the ANSYS 15 workbench are compared with the theoretical results. The first comparison is done between the stresses obtained therotically and by using ANSYS. The graph is plotted between stress vs the no. of tooth. The following graph shows the deformation of gears. The graph represents the nature of deformation along the distance from the base circle. The deformation goes on increasing towards the tip of the tooth

A. Larger gears

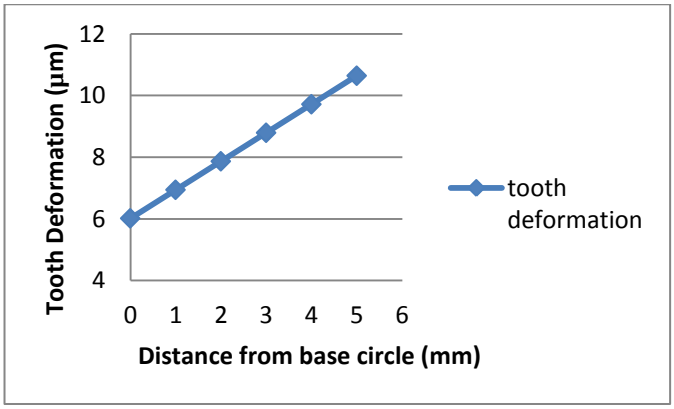


Fig.10 Deformation v/s distance from base circle

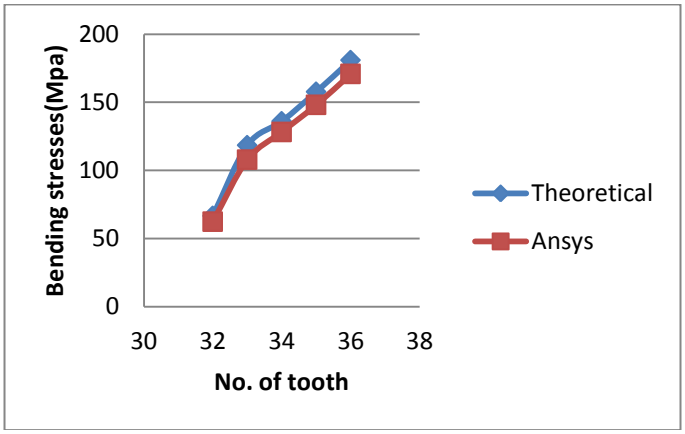


Fig.11 Contact stress v/s no. of tooth

B. Smaller gears

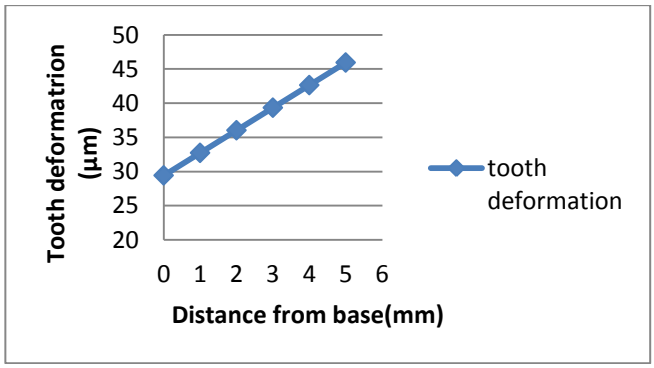


Fig.12 Deformation v/s distance from base circle.

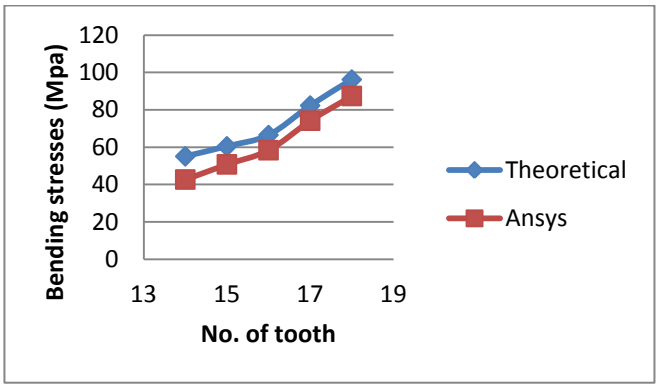


Fig.13 Contact stress v/s no. of tooth.

From the graphs it is clear that the deformation values are more for the smaller gears. The following table represents the theoretical and analytical stresses. The percent change is also given for both the gear pairs

GEARS	THEORETICAL STRESSES (MPA)	ANALYTICAL STRESSES (MPA)	%CHANGE
LARGER GEAR	66.67	62.41	6.8%
SMALLER GEAR	55.1	48.4	13.6%

Table.2 Comparing stresses on both the gears.

For the larger gear, the percent change in the bending stress value is 6.8 and that for smaller gear is 13.6.The deviation in the results are neglible.

CONCLUSION

In this paper, we analyzed larger gears as well as five smaller gears in mesh. We have concluded that the bending stress is a function of tooth geometry. Keeping minimum contact ratio alteast one, the stresses inreases with increase in number of tooths considering other operating parmameters such as torque and speed as constant.

Deformation of the gears gradually increases towards the tooth tip due to increase in the value of tangential force acting at the tip The analytical and theoretical results sre validated. The analytical stresses closely resemble the stresses calculated from equations.

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