Comparison of High Contact Ratio and Low Contact Ratio Gears

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ABSTRACT

In this work, comparison of high contact ratio gears and low contact ratio gears has been done for the same load. We have considered the bending strength, shearing at contact region, normal stress and some contact parameters like contact pressure, penetration and contact stiffness to compare HCR gears and LCR gears. in the results we have found that HCR gears r deforming less when compared to the later type for the same load transmitted. Bending strength in HCR gear set less when compared to the later one because of the load distribution among the teeth.
Keywords: HCR, LCR, ANSYS, FEA, CAD

I. INTRODUCTION

Gears are one of the oldest of humanity's inventions. Nearly all the devices we think of as a machines utilize gearing of one type or another. Gear technology has been developed and expanded throughout the centuries. In many cases, gear design is considered as a specialty. Nevertheless, the design or specification of a gear is only part of the overall system design picture. From industry's standpoint, gear transmission systems are considered one of the critical aspects of vibration analysis. The understanding of the behavior when gears are in mesh is extremely important if one wants to perform system monitoring and control of the gear transmission system. Although there are large amount of research studies about various topics of gear transmission, the basic understanding of gears in mesh still needs to be confirmed.

1.1 PROBLEM STATEMENT:

When a pair of gears mesh, localized Hertzian contact stress are produced. This is a non-linear problem, and it can be solved by applying different types of contact elements and algorithms in finite element codes. However, due to the complicated contact conditions, acquiring results in the meshing cycle can be challenging since some solutions may not converge. In any case, using quadrilateral elements seem to be useful in solving gear contact problems with finite element analysis. Furthermore, meshing stiffness is often being discussed when a pair of gears are in mesh. Meshing stiffness can be separated into Torsional Mesh Stiffness and Linear Tooth Mesh Stiffness. The torsional mesh stiffness is defined as the ratio between the input torque load and the angular displacement of the input gear. Once in mesh, the gears' pitch circles roll on each other without slipping. With a constant torque load, the torsional mesh stiffness changes through the rotation of the gears. These changes are due to the contact ratio between the pinion and gear. Depending on the contact ratio; the contact region would change and alternate from single tooth contact to double tooth contact or even a higher number of contacting pairs. This change of contact regions is referred to as a mesh cycle. Through the mesh cycle, the torsional mesh stiffness can be utilized as a tool to investigate gear transmission errors. Furthermore, the torsional mesh stiffness is related to the linear tooth mesh stiffness by the normal contact force that acts along the line of action. Basically, the linear tooth mesh stiffness of the gears is an easy approach to understand the coupling between the torsional and transverse motions of the system. The linear tooth mesh stiffness has been chosen as the primary parameter to be studied in this work. This work is mainly focusing on, but not limited to, the gear modeling and analysis using the finite element method. Large amounts of FEA calculations were made using the finite element code-ANSYS. Comparisons between predicted linear tooth mesh stiffness are presented with different type of elements, integration methods, meshing quality, plane stress vs. plane strain, sensitivity of model tolerance, and crack modeling. In addition, small amount of experiments are performed in the aim of validating gearbox diagnostic methodologies. The objective of the

experiments is to monitor and identify vibration frequencies associated with the gears and bearings in a gearbox.

1.2 OBJECTIVE:

In present generation vehicles demand a very compact transmission to meet mobility requirements. A compact transmission with low operating noise and vibration is desirable in military tracked vehicles to reduce weight and improve power-to-weight ratio. It is also necessary to increase the rating of existing transmissions in military tracked vehicles, like prime movers, to accommodate the additional weight required for ballistic protection. Hence, it was decided to apply a high-contact-ratio (HCR) spur gearing concept that will reduce noise and vibration and enhance load carrying capacity. In HCR gearing, the load is shared by a minimum two pairs of teeth, as in helical gears. It was decided to analyze the load sharing of the low-contact-ratio (LCR) gearing used in the spur gear mesh of the existing final drive; and, to analyze the load sharing of the HCR gearing that will be used to replace the LCR gears without any change in the existing final drive assembly. This paper deals with analysis of different parameters of gear mesh throughout the profile for both gear and pinion gears of LCR/HCR gearing—using finite element analysis (FEA), contact stress and deflection along the profile of both LCR and HCR gearing.

II. LITERATURE REVIEW

Gears are a critical component in the rotating machinery industry. Various research methods, such as theoretical, numerical, and experimental, have been done throughout the years regarding gears. One of the reasons why theoretical and numerical methods are preferred is because experimental testing can be particularly expensive. Thus, numerous mathematical models of gears have been developed for different purposes. This chapter presents a brief review of papers recently published in the areas of gear design, transmission errors, vibration analysis, etc., also including brief information about the models, approximations, and assumptions made.

Wyluda and Wolf [1] performed an elastic-plastic finite element analysis of the quasi-static loading of two acetal copolymer gears in contact. The applied load vs. gear set rotation is compared to actual experimental results. The geometry of the gear is modeled with variable thickness between the rim and web. Plane strain elements were used in the finite element model. Gear tooth failure is considered and modeled using methods of deactivating and separating elements when the tensile strength is exceeded. As a result, the mechanical behavior and prediction of copolymer acetal gears is quite complicated. Combination of computer simulations and component testing has merged a better understanding of copolymer acetal gear design. Also, the results indicate that a linear elastic approach is only suitable when the gears are under low loads and deformations. So, performing non-linear analysis is essential in order to optimize a gear set.

In 2003, Barone *et al*. [2] aimed at investigating the behavior of a face gear transmission considering contact path under load, and load sharing and stresses, for an unmodified gear set including shaft misalignment and modification on pinion profile. The investigation is carried out by integrating a 3D CAD system and a FEA code, and by simulating the meshing of pinion and gear sectors with three teeth, using contact elements and an automated contact algorithm. The results show the influence of load on theoretically calculated contact paths, contact areas, contact length and load sharing. Also, it shows that the effectiveness of the numerical approach to the meshing problem in its complexity and that commonly adopted approaches are not suitable for non conventional, highly loaded gears in which rim and tooth deformations are not negligible. Overloads due to pinion misalignments and shift of contact areas are also being considered.

In 2001, Howard *et al*. [3] used a simplified gear dynamic model to explore the effect of friction on the resultant gear case vibration. The model includes the effect of variations in gear tooth torsional mesh stiffness, developed using finite element analysis, as the gears mesh together. The frictional force between teeth is integrated into the dynamic equations. Single tooth crack effects are shown on the frequency spectrum. The effect of the tooth crack could be seen in the time waveforms of all the dynamic variables being simulated when friction was neglected. The diagnostic techniques worked clearly when friction was included in the model,

and in most cases friction gave a negligible change in the resulting values.

In 2005, Wang and Howard [4] presented the methods and results of the use of FEA high contact ratio gears in mesh. The numerical models were developed with gears in mesh under quasi-static conditions. The details of transmission error, combined torsional mesh stiffness, load-sharing ratio, contact stress and tooth root stress against various input loads over a complete mesh cycle are also taking into account. Thus, various tooth profile modifications are presented and comparisons between the results show evidence for the optimal profile modification expected to gain the maximum benefit of high contact ratio gears. Also, the optimal relief length is normally dependent on the gears' geometrical properties. The results of optimal relief length vs. the tooth addendum variations have shown that the relief length can be very small, and it suggests that the contact ratio or the module be increased in order to retain the natural benefits of high contact ratio gears.

One year later (2006), Wang and Howard [5] investigated a large number of 2D and 3D gear models using finite element analysis. The models included contact analysis between teeth in mesh, a gear body, and teeth with and without a crack at the tooth root. The model results were compared using parameters such as the torsional mesh stiffness, tooth stresses and the stress intensity factors that are obtained under assumptions of plane stress, plane strain, and 3D analysis. Also, the models considered variations of face width of the gear. As a result, the finite element solution has been shown to produce acceptable results for stresses within a limited range. The 2D modeling errors can be significant when the gear is subject to a fracture such as a tooth root fatigue crack. Thus, 2D solutions may only apply in a very narrow range. Also, ignoring these errors (fatigue analysis) can lead to significantly erroneous results. The actual parameters used in the investigations demonstrate that caution must be taken where 2D assumptions are applied in the modeling.

In 2007, Carmignani *et al*. [6] have simulated the dynamic behavior of a faulted gear transmission. The meshing stiffness was evaluated statically as a function of the gear angular position using finite element gear meshing models. The deformation of the teeth under load and the faulted gears such as tooth cracks of different lengths at different locations on the tooth flank were taking into account in the simulations. Also, the numerical simulations were carried out in a simulink environment with different applied torques and gear angular velocities. As a result, the fracture causes a variation in the meshing stiffness when the faulty tooth is engaged in meshing. The crack affects stiffness only if the cracked zone is loaded between the tooth root and the contact point. However, if there are more teeth in contact, the uncracked teeth would share the load, which unloads the cracked tooth and thus reduces the stiffness disturbance effect.

III. GEAR DESIGN AND CALCULATIONS

3.1 Overview
The main purpose of gearing is to transmit motion from one shaft to another. If there is any mistake or error on the gears, motion will not be transmitted correctly. Also, if the errors on the gears are crucial, it may destroy or heavily damage the components in a gearbox. Therefore, it becomes important to understand the subject of gearing. In order to gain better understanding of gearing, one should get some knowledge about the design of gear and the theory of gear tooth action.

3.2 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.

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.

Figure 1. Types of gears

3.3 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 shape. According to Drago [7], the same theoretical tooth forms can be produced by both forming and generating, but the actual profiles that result on the parts differ slightly. Generated profiles are actually a series of flats whose envelope is the desired

form, While the surface of a formed profile is usually a continuous curve. In general, gear teeth may be machined by milling, shaping, or hobbing. Also, they may be finished by shaving, burnishing, grinding, or lapping.

Milling-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. The only drawback for this method is the necessity to use a different cutter for each gear because different gears have different-shaped tooth spaces. Shaping-either a pinion cutter or a rack cutter will be used to generate the gear teeth. The cutter reciprocates with respect to the work and is fed into the gear bank. Since each tooth of the cutter is a cutting tool, the teeth are all cut after the blank has completed one rotation.

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. A single hob of a given normal pitch and pressure angle can be used to produce any standard external spur or helical gear with the same pitch and pressure angle.

Finishing-if there are errors in the tooth profiles, gears may be subjected to additional dynamic forces. A good finishing on tooth profiles would help to diminish these errors. Shaving machines offer to cut off a small amount of metal and improve the accuracy of the tooth profile. Burnishing utilizes hardened gears with slightly oversized teeth and run in mesh with the gear until the surfaces become smooth. Grinding employs the principle of generating and produces very accurate gear teeth.

Lapping is applied to heat treated gears to correct small errors, improve surface finish, and remove nicks and burrs

3.4 Theory of gear tooth action

3.4.1 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 is presented by Shigley *et al.* [8] and displays the nomenclature of spur gear teeth.
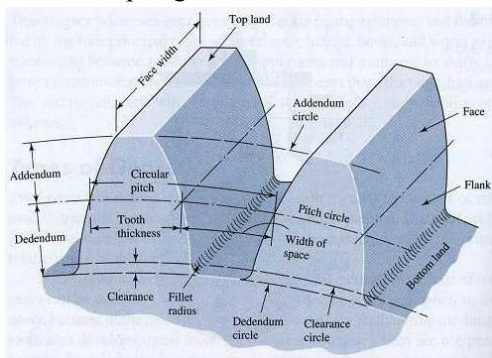
Figure 2. Nomenclature of spur gear teeth

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 the circumference of the pitch circle between the corresponding points of adjacent teeth. Therefore, the circular pitch is the sum of the tooth thickness and the width of a space. The addendum is the radial distance between the pitch circle and the top of the tooth (top land). The dedendum is the radial distance between the pitch circle and the bottom of the tooth space (bottom land). The clearance is the amount by which the dedendum in a given gear exceeds the addendum of its mating gear. The 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,

3.4.2 Line of action:
When gear teeth are meshing against each other, it will generate rotary motion. Also, when a curved surface pushes against another, the point of contact appears where the two surfaces are tangent to each other. Imagine a line pierces through this contact point with the characteristic of being common normal to the surfaces. Then, the forces at any instant are directed along this line, and this line represents the direction of the forces. This is called the line of action or pressure line. Furthermore, the line of action will intersect the line of centers which is formed by the gears' centers at point *P*. This point is referred as the pitch point. The pitch point

can be found by drawing the pitch circles of the gears since they are supposed to come in contact as soon as the gears are meshed together. The following figure shows the line of action and the tooth action of a pair of gears.

3.4.3 Fundamentals:
When two gears are meshed with each other, their pitch circles roll on one another without slipping. Thus the pitch line velocity can be defined as:

$$V = r_p \omega_p = r_g \omega_g \qquad (3.4)$$
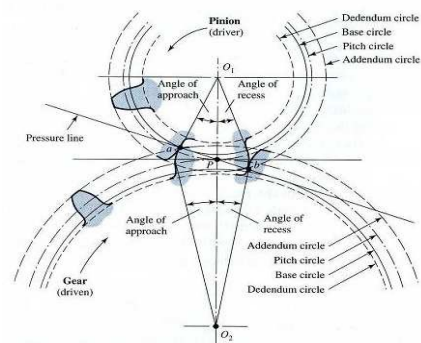
Figure 3. Tooth action

where $p\ r$ and $g\ r$ are the pitch radii of the pinion and gear; p and g are the angular velocities of the pinion and gear respectively. As shown in Figure 3.4.2, the pressure line is tangent to the base circles of the pinion and gear, and it pierces through the contact point. The horizontal line which is tangent to the pitch circles of the pinion and gear also pierces through the contact point. The angle between this horizontal line and the pressure line is identified as the pressure angle. The pressure angle usually has values of 141/2, 20, or 25 degrees. Furthermore, since the base circles are tangent to the pressure line, using basic geometry, the base circle radius can be determined through the pitch radius and the pressure angle.

$$r_b = r. \cos\phi \qquad (3.5)$$

For standard gear teeth, the addendum and dedendum distances are 1/*P* and 1.25/*P* respectively. The clearance, as previous described, is equal to the dedendum distance minus the addendum distance. In order to draw a tooth, one must know the tooth thickness. The tooth thickness is measured on the pitch circle and can be calculated as:

$$t = \frac{p}{2} \qquad (3.6)$$

3.4.4 Contact ratio:

The contact between the gears begins and ends at the intersections of the two addendum circles with the line of action. Depending on the design of the gears and the contact ratio, sometimes there will be two or more teeth in contact. According to Machinery's Handbook [9], the contact ratio is the ratio of the arc of action in the plane of rotation to the circular pitch. Often, it is considered as a number that indicates the average number of pairs of teeth in contact. Furthermore, the contact ratio is obtained most directly as the ratio of the length of action to the base pitch.

$$m_c = \frac{Lab}{pb} = \frac{Lab}{p.cos\phi}$$

3.5 Involute properties

The involute curve of a tooth plays an important role in gear design and analysis. For instance, friction and wear between two gears is dependent on the profile of the teeth; the uniform velocity ratio is also dependent on the tooth profile. The involute tooth allows the center distance or spacing of the gears to vary over some range without affecting the velocity ratio. Therefore, an accurate gear tooth profile will lead to high quality results. Even though the formation of the involute tooth profile has been described in [7] and [8], it is still challenging to construct the correct gear tooth profile in CAD and FEA code environment. However, with the up-to-dated numerical programs, one is able to develop a reliable code to create the gear tooth profile [10]. With some adjustments [11], the code can accurately generate the profile of a gear tooth. The details of the code can be found in the Appendix at the end of this work. The code only requires users to provide few parameters of the gear: diametral pitch, pressure angle, and number of teeth. 3D model was developed in PTC creo 3.0 software.

	Low Contact Ratio		High Contact Ratio	
	Pinion	Gear	Pinion	Gear
Diametral Pitch, P (teeth/inch)	4.064	4.064	2.674	2.674
Pressure Angle (degree)	20	20	17.5	17.5
Number of teeth	25	89	38	135
Contact ratio	1.74	1.74	2.24	2.24
Face width (mm)	73.66	73.66	73.66	73.66
Pitch diameter(mm)	101.6	360.946	101.6	360.946
Steady transmitted load(KN)	30	30	30	30
E (For steel) (GPa)	210	210	210	210
μ (For steel)	0.30	0.30	0.30	0.30

IV. CREO PROCEDURE

1. Open sketch module and draw four circles and modify the dimensions as per the design ans hit okay mark.
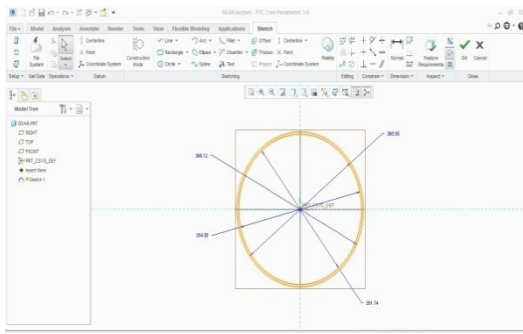
2. now draw select the option "draw a datum curve through equation" to draw the involute profile
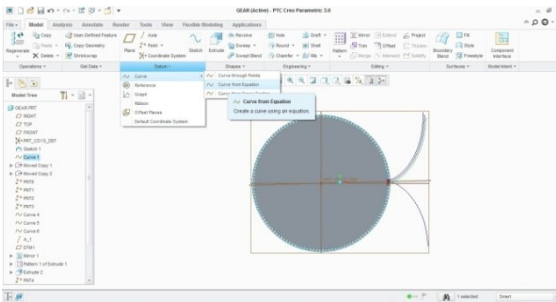
We have to use the following equation to draw the involute profile
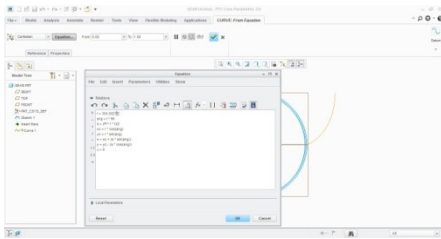
Take some datum points and datum curves for reference as follows
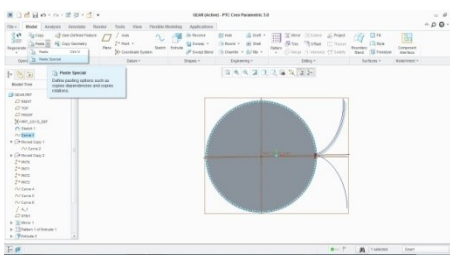
Now select the involute curve and select copy and paste special at angle of 2.66 and 1.33 degrees respectively
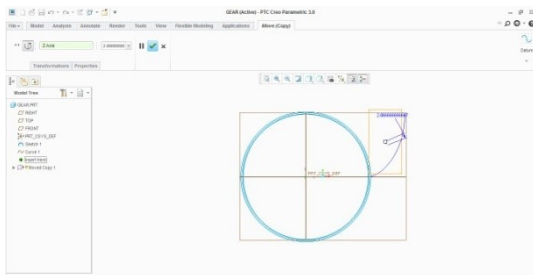
Take the mirror image of the middle curve about the plane passing through the origin and point of intersection of middle curve and the pitch circle
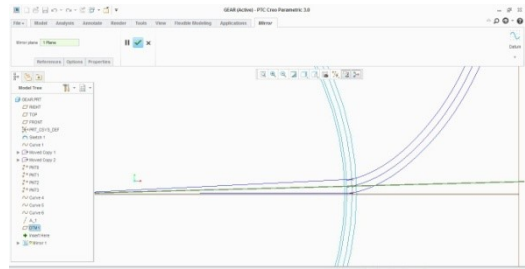
Now we get the closed area to draw the tooth profile and now we can make tooth using extrude option
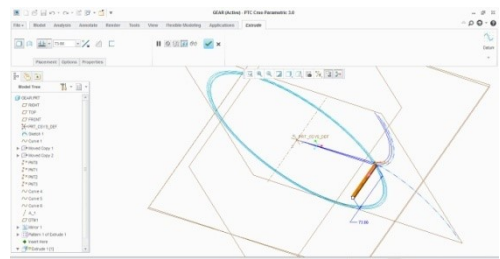
Now make the pattern to get the required gear train
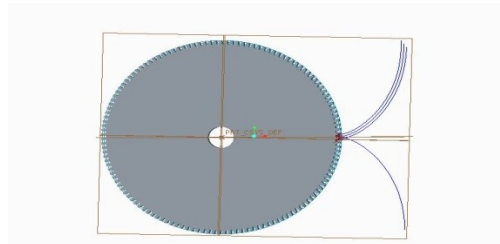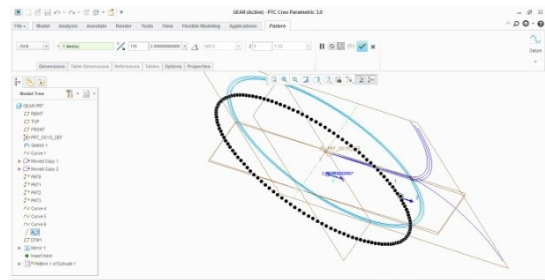
Dinal gear train with high contact ratio is
Final assembly of hcr gears
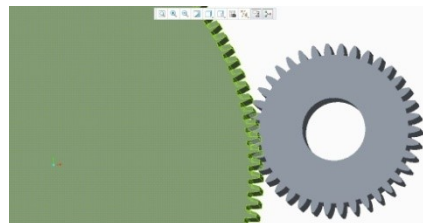
Low contact ratio gears
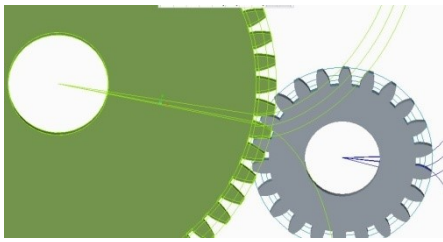
V. ANSYS PROCEDURE

1. go to start button and open ansys workbench 15.0
2. open geometry and import the required geometry in '.igs' format and hit generate.

3.now go to model and click edit
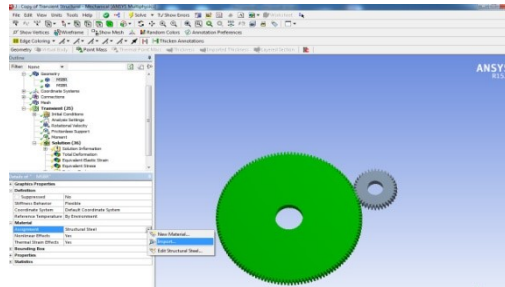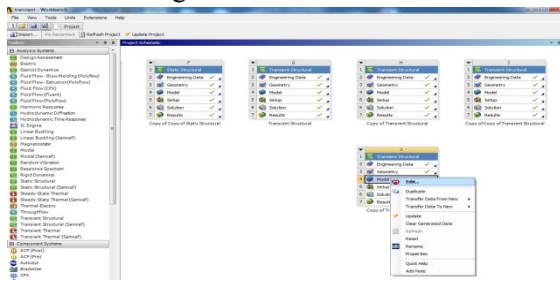
now assign the materials for the geometries meshing
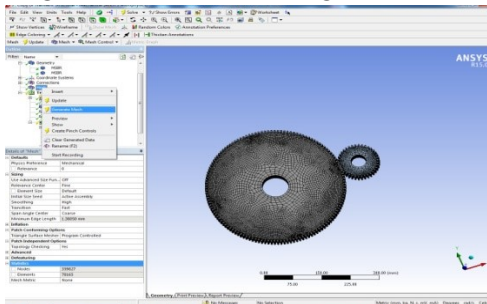
5. go to transient structural and set the boundary conditions for model like supports and loads
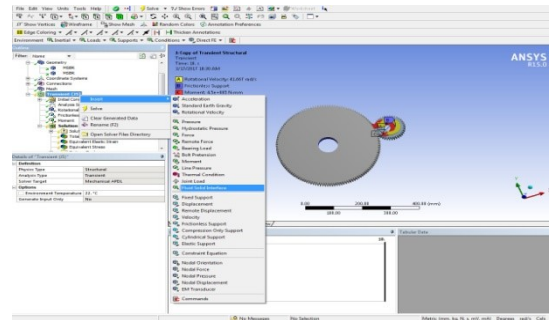
6. go to solution and add the required entities for results
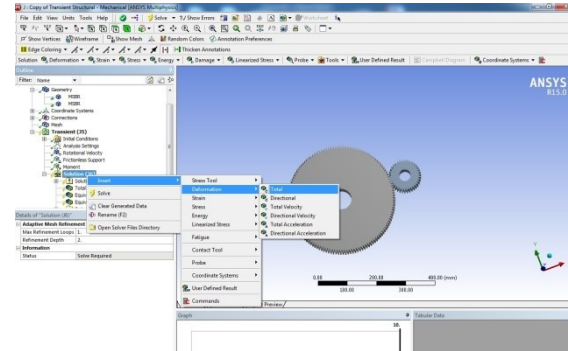
VI. RESULTS

Total deformation-lcr gears at 2500rpm
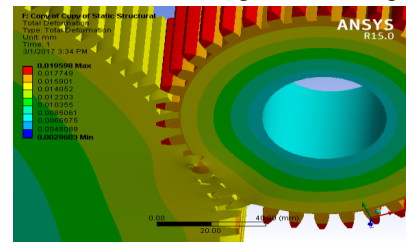
Total deformation-lcr gears at 5000rpm
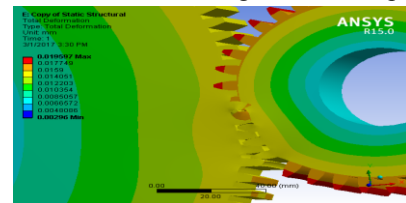
Total deformation-lcr gears at 10000rpm
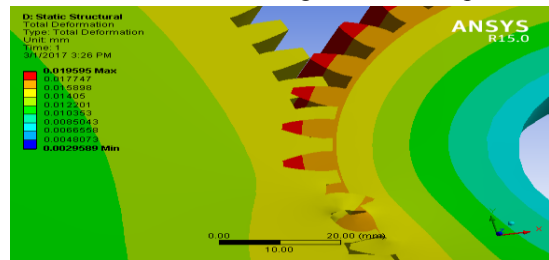
Total Deformation-HCR at 10000rpm
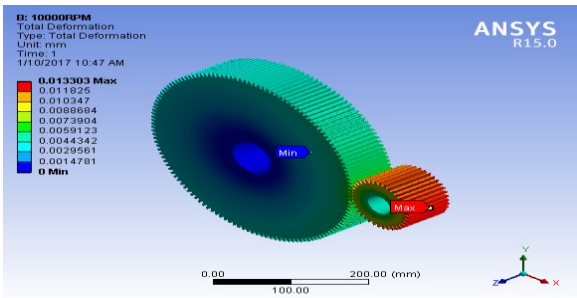
Total Deformation-HCR at 5000 rpm
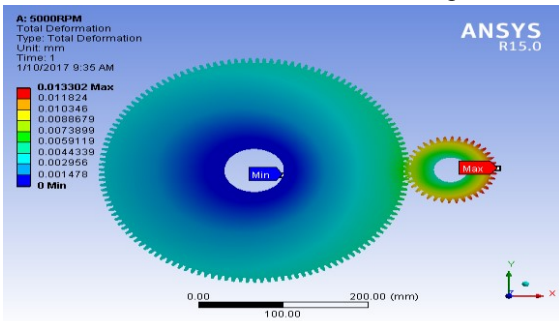
Total Deformation-HCR at 2500 rpm

TOTAL DEFORMATION (mm)

- LCR GEARS-2500
- LCR GEARS-5000
- LCR GEARS-10000
- HCR GEARS-2500
- HCR GEARS-5000
- HCR GEARS-10000
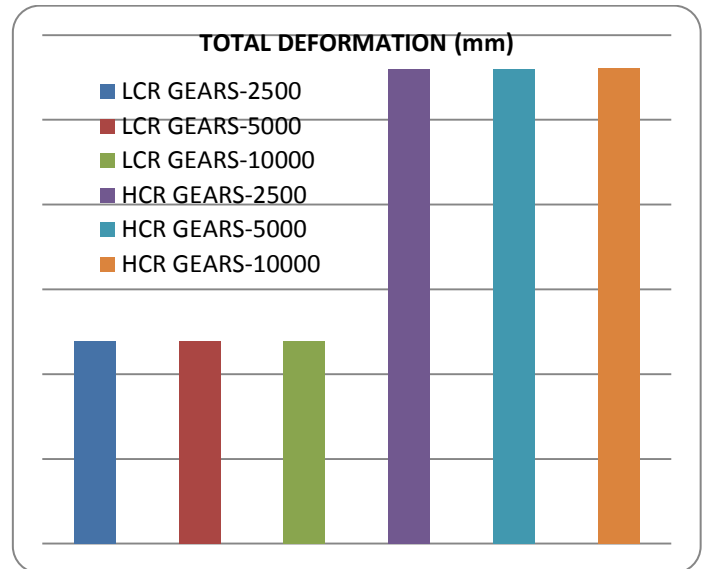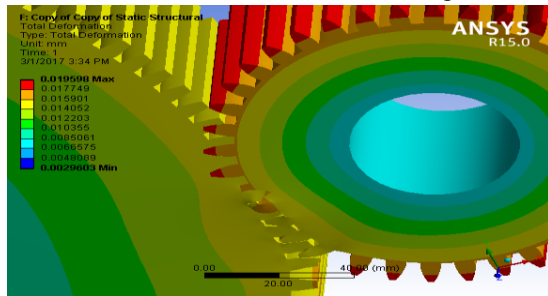
Equivalent Stress (VON-MISES) -HCR at 10000rpm
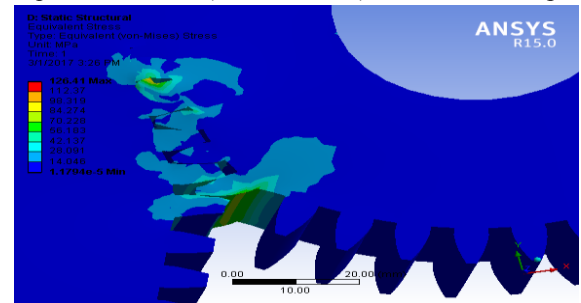
Equivalent Stress (VON-MISES) -HCR at 5000rpm
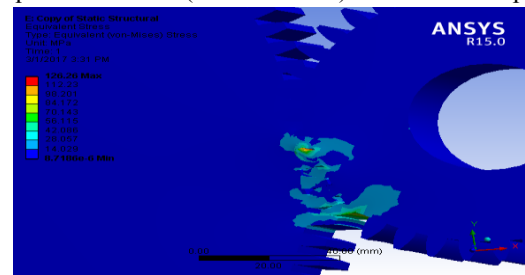
Equivalent Stress (VON-MISES) -HCR at 2500rpm
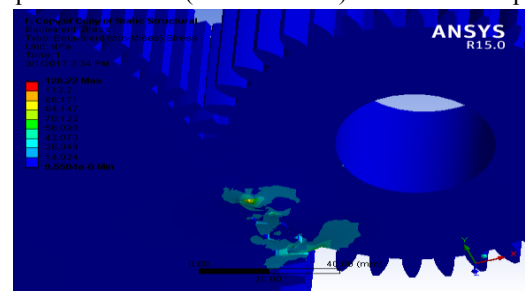
Equivalent Stress (VON-MISES) -LCR at 10000rpm

From the below graph we can observe that deformation is less in low contact ratio gears when compared to high contact ratio gears. It is because of the variation in the thickness of gear. in HCR and LCR we have taken the same pitch circle diameter but we have varied the number of teeth. so thickness of gear varies and hence the total deformation. but the difference in the deformation is in microns. so, this deformation doesn't really influence the performance of the gears.
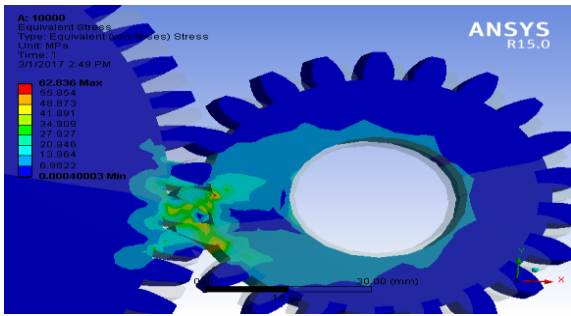
Equivalent Stress (VON-MISES) -LCR at 5000rpm
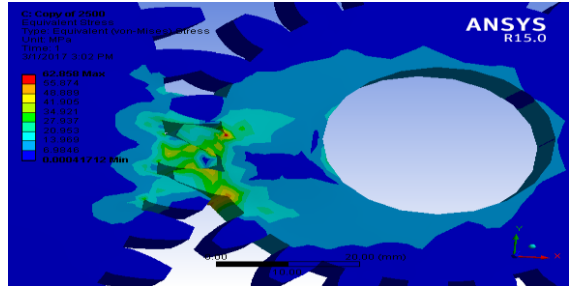
Equivalent Stress (VON-MISES) -LCR at 2500rpm

Equivalent (von-Mises) Stress (Mpa)

■ LCR GEARS-2500 ■ LCR GEARS-5000
■ LCR GEARS-10000 ■ HCR GEARS-2500
■ HCR GEARS-5000 ■ HCR GEARS-10000

In the above graph of Equivalent Stress (VON-MISES) for HCR and LCR gears at different rotational velocities we can see the very slight variation in the stress with the rotational speed for same set of gear train. But we can see a tremendous
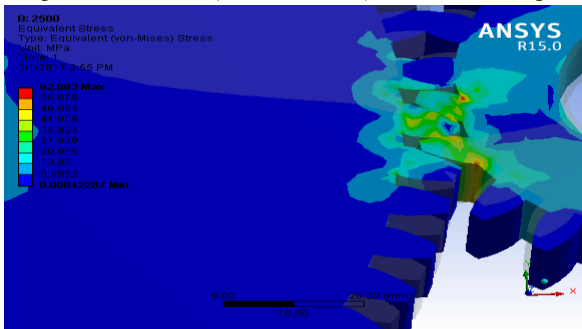
decrement in the stress values for HCR gears when compared to LCR gears. this trend occurred due to the difference in the load on each tooth. we can more stress in case of LCR because the load has been shared by 2 pairs of teeth but where as in HCR gears , the total applied load has been shared by 4 pairs of teeth.

Equivalent elastic Strain-HCR at 10000rpm
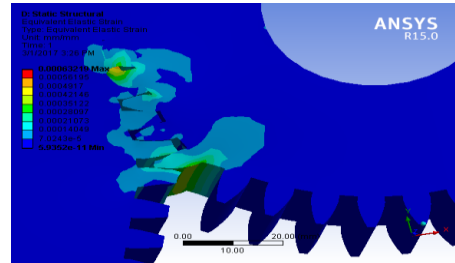
Equivalent elastic Strain-HCR at 5000rpm
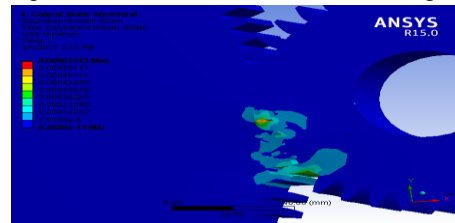
Equivalent elastic Strain-HCR at 2500rpm
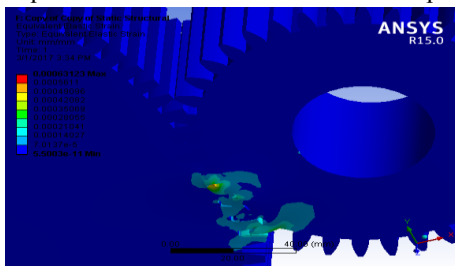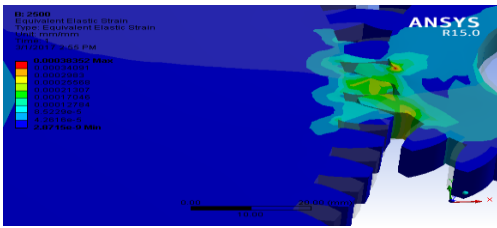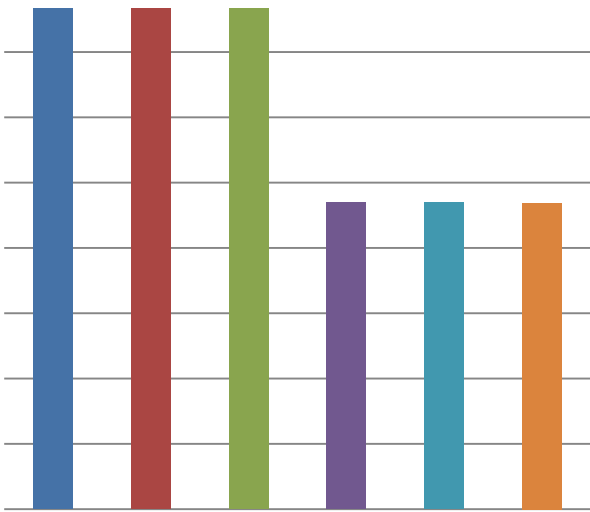

Equivalent elastic Strain-LCR at 10000rpm
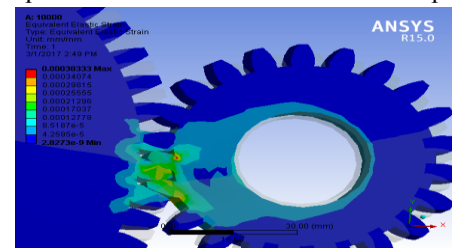
Equivalent elastic Strain-LCR at 5000rpm
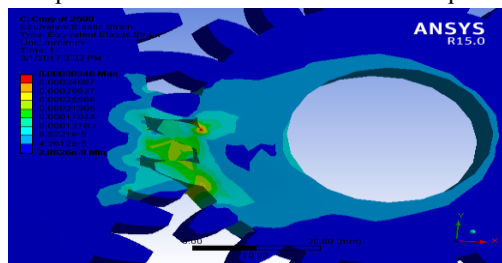
Equivalent elastic Strain-LCR at 2500rpm

## Equivalent Elastic Strain

- LCR GEARS-2500
- LCR GEARS-5000
- LCR GEARS-10000
- HCR GEARS-2500
- HCR GEARS-5000
- HCR GEARS-10000

In the above graph we can see the variation of strain between HCR- gears and LCR-gears at different rotational velocities. we can see more than 40% difference in strain in HCR-gear set when compared to LCR-gear set.
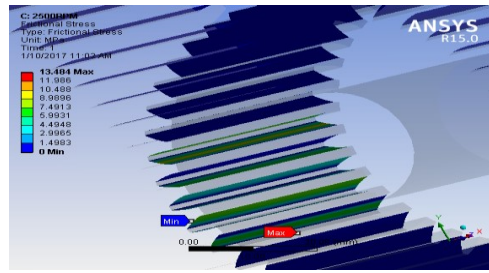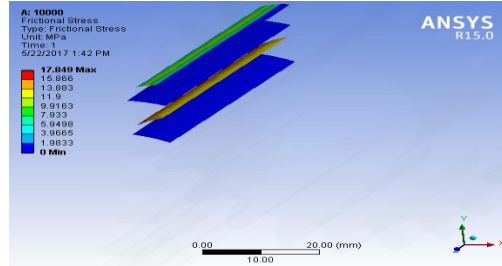
Frictional Stress

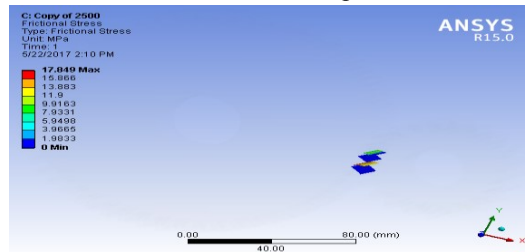Frictional Stress of LCR-10000rpm

Frictional Stress of LCR-5000rpm

Frictional Stress of LCR-2500rpm

Frictional Stress of HCR-10000rpm
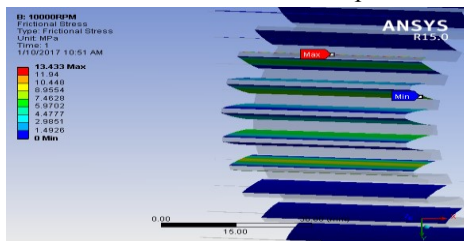
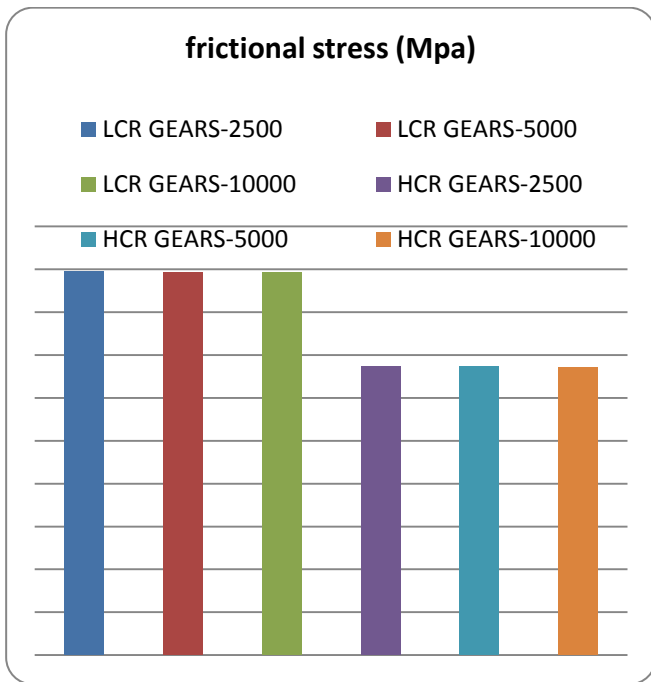Frictional Stress of HCR-5000rpm
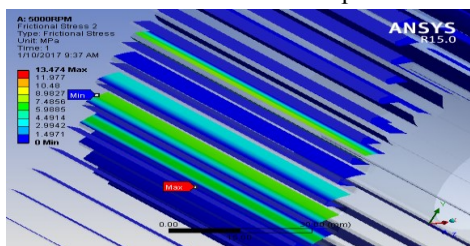
Frictional Stress of HCR-2500rpm

frictional stress (Mpa)

PRESSURE
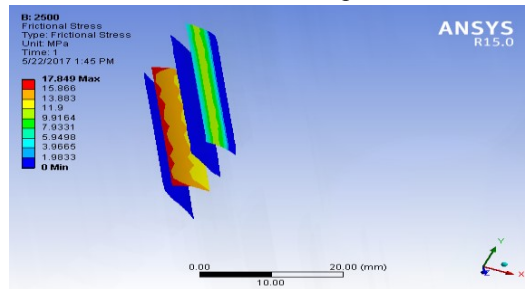
We can see more than 20% decrement in the frictional stress values between HCR and LCR- gear sets this is due the load shared by the frictional load among teeth. in case of lcr gear set the hole frictional load is shared by two pairs of teeth whereas the same has been shared by more than four pairs of teeth. the reduction in frictional stress will reduce the tooth wear and power consumption to run the machine also provides the smooth running.

PRESSURE:

Pressure is the ratio of load on each tooth to the face area of the tooth. in case of LCR gears the load on each tooth is very high when compared to the HCR gear set. So, the pressure value will less in HCR gear set when compared to HCR gear set and the same trend is shown in the graph shown below.
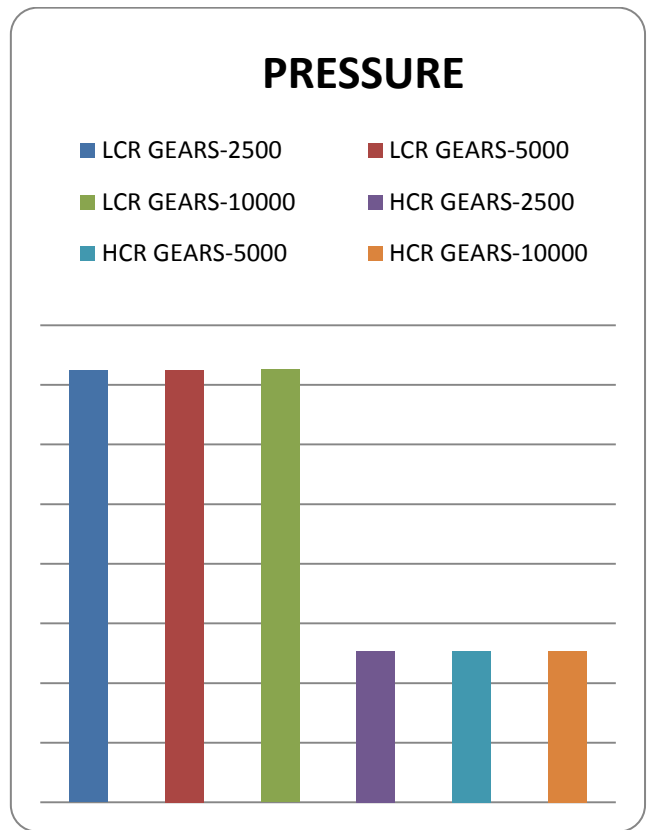
VII. CONCLUSION

- Maximum value of deformation 0.013303 mm is observed in HCR gear set at 10000rpm against 0.011698 mm in LCR gear set at the same rotational speed.

- Maximum value of equivalent elastic strain 0.00038352 is observed in LCR gear set at 2500rpm against 0.00023449 in HCR gear set at the same rotational speed.

- Minimum value of equivalent stress 45.398 Mpa is observed in HCR gear set at 10000rpm against 62.836 Mpa in LCR gear set at the same rotational speed.

- Minimum value of frictional stress 13.433 Mpa is observed in HCR gear set at 10000rpm against 17.849 Mpa in LCR gear set at the same rotational speed.

- Minimum value of pressure 25.348 Mpa is observed in HCR gear set at 10000rpm against 72.416 Mpa in LCR gear set 2500rpm.

VIII. FUTURE SCOPE

As the gears are finding prominent applications for transmission of power in different fields like automobile, marine, aerospace and more importantly in robotics it needs a vast research for using gears in different applications. in general gears are being used for transmission of loads we need to conduct research in dynamic load applications and we have to check the performance in dynamic load conditions.

As we all know that the performance of gears depends on the materials used for making of gears. so we have to conduct research in the material science to choose the right material in order to reduce the weight, contact friction that improves the performance of gars which in turn increase the efficiency.