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Comparision Of High Contact Ratio And Low Contact Ratio Gears

Bhukya Balu
[1]*Department of Mechanical Engineering Mandava Institute of Engg & Tech.Jaggayapeta Andhrapradesh India*

ABSTRACT: *In this paper, 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.*

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 gearbo.

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.

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.

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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

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teeth are all cut after the blank has completed one rotation.

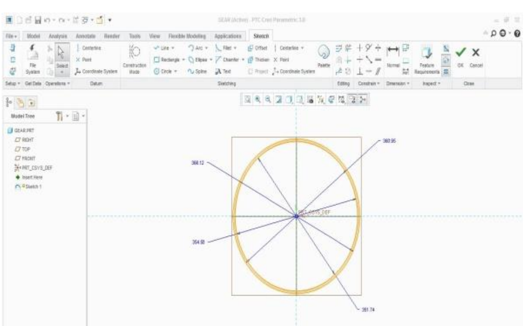
Hobbing – one of the fastest ways of cutting gears. The hob basically is a cutting tool that is shaped like a worm. As the hob rotates and feeds along the gear axis, the gear rotates about its axis in a carefully controlled environment. A single hob of a given normal pitch and pressure angle can be used 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.

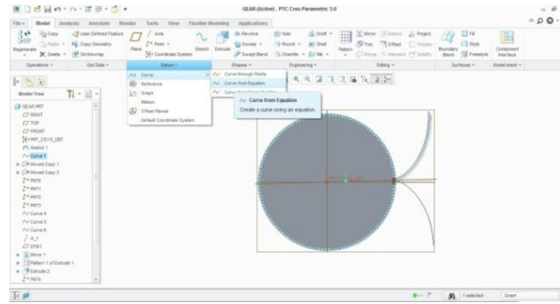
Lapping is applied to heat treated gears to correct small errors, improve surface finish, and remove nicks and burrs

IV. CREO PROCEDURE

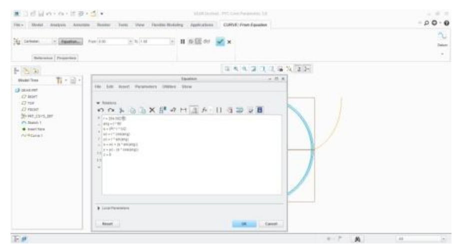
1. Open sketch module and draw four circles and modify the dimensions as per the design ans hit okay mark.



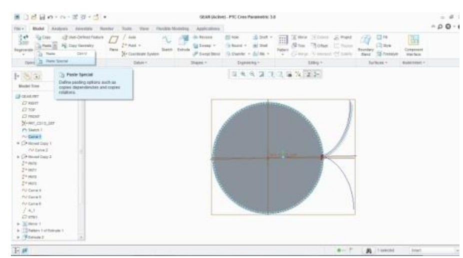
2. now draw select the option "draw a datum curve through equation" to draw the involute profile



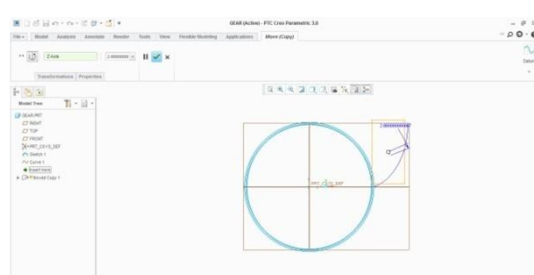
We have to use the following equation to draw the involute profile



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



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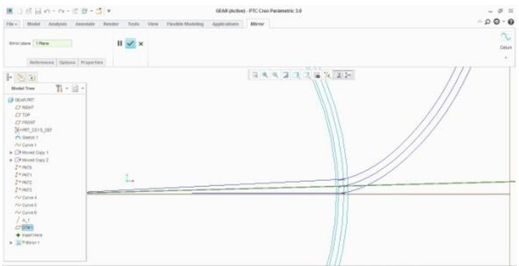
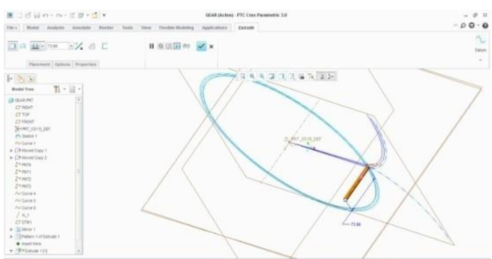
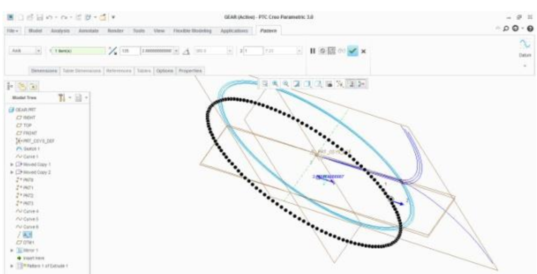
take the mirror image of the middle curve about the plane passing through the origin and point of intersection of middle curve and the pitch circle



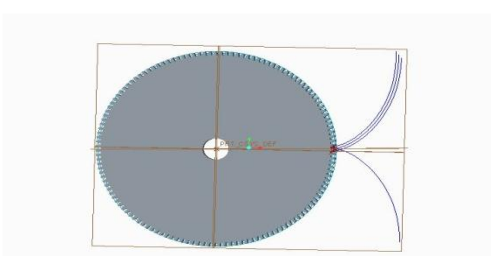
now we get the closed area to draw the tooth profile and now we can make tooth using extrude option



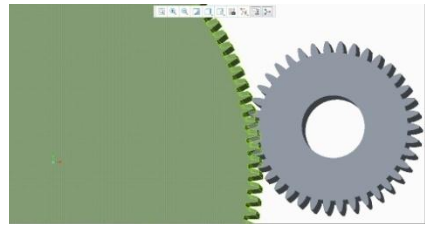
now make the pattern to get the required gear train



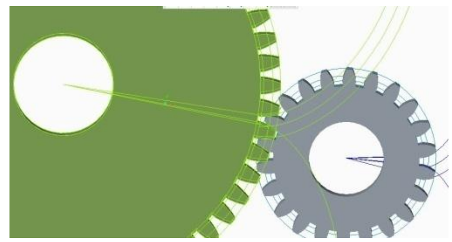
final gear train with high contact ratio is



final assembly of hcr gears



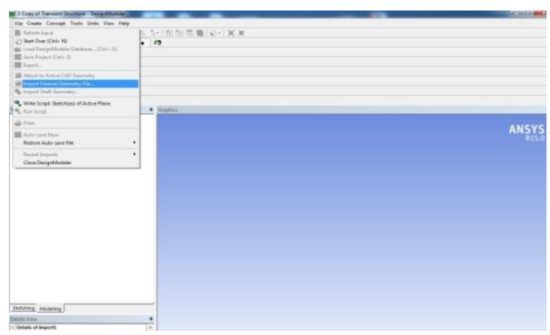
low contact ratio gears



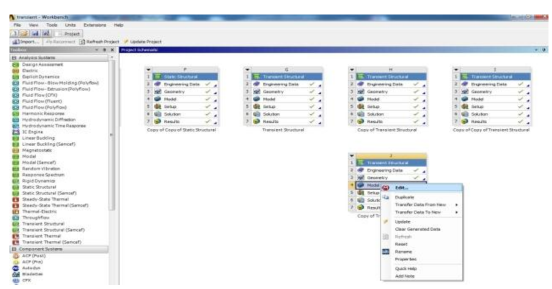
ANSYS PROCEDURE

1. go to start button and open ansys workbench 15.0

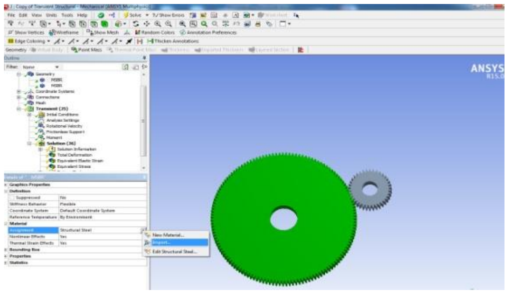
2. open geometry and import the required geometry in '.igs' format and hit generate.



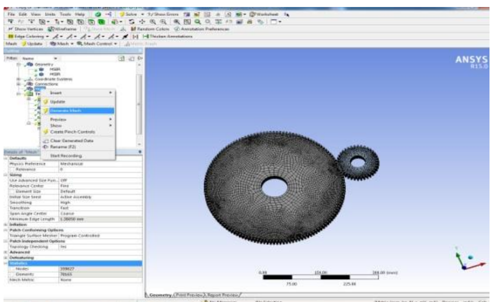
3.now go to model and click edit



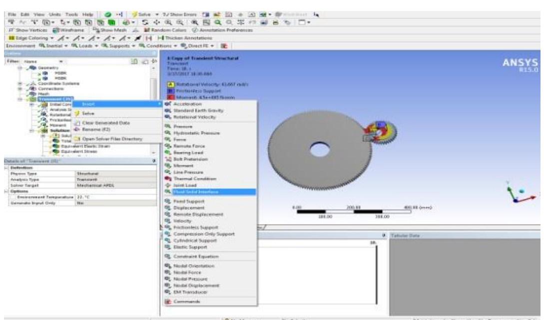
now assign the materials for the geometries



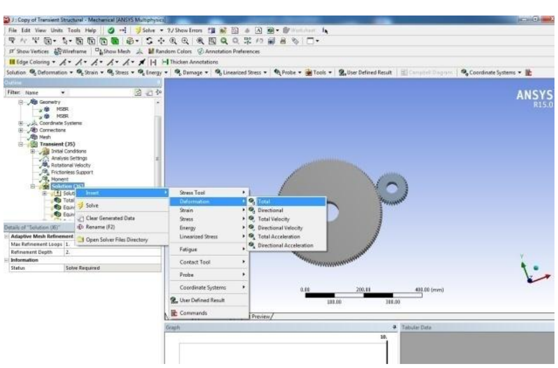
meshing



5. go to transient structural and set the boundary conditions for model like supports and loads

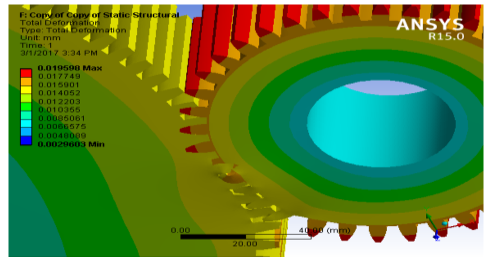


6. go to solution and add the required entities for results

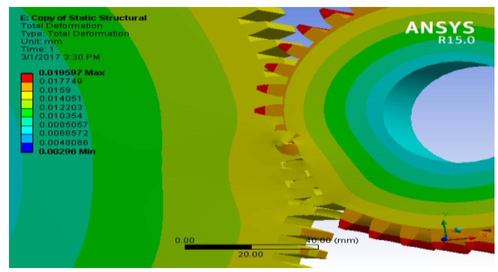


V. RESULTS

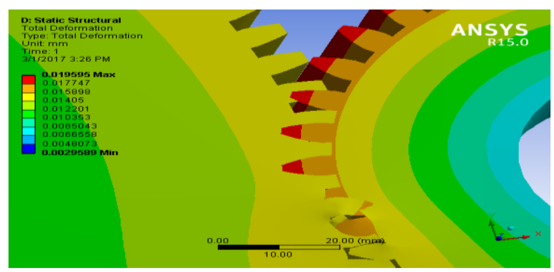
Total deformation-lcr gears at 2500rpm



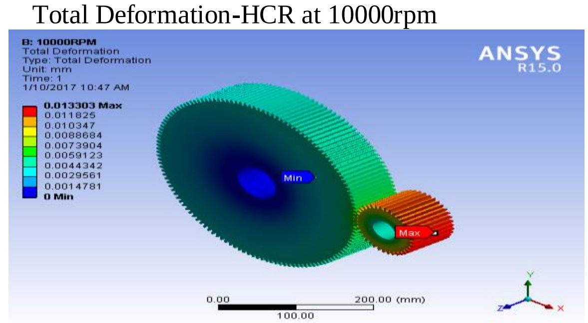
Total deformation-lcr gears at 5000rpm



Total deformation-lcr gears at 10000rpm



Total Deformation-HCR at 10000rpm



Total Deformation-HCR at 5000 rpm

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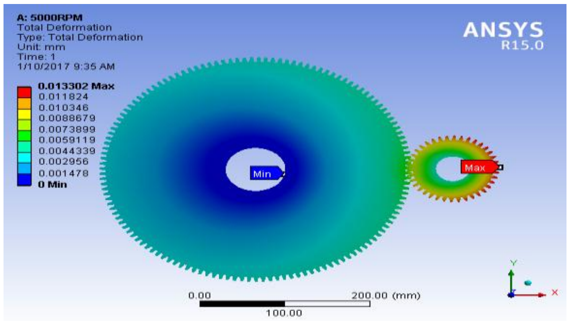
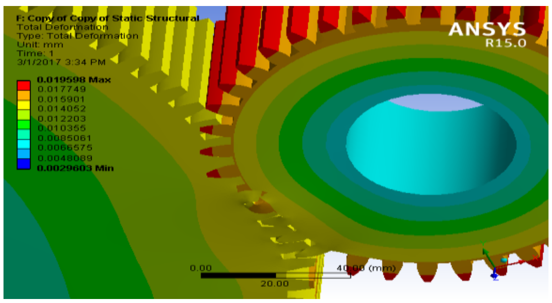
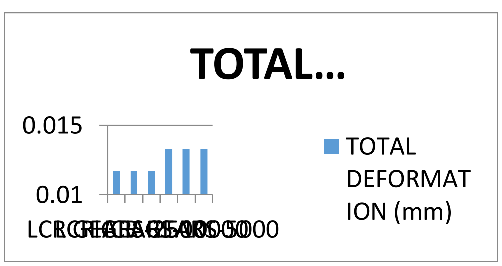


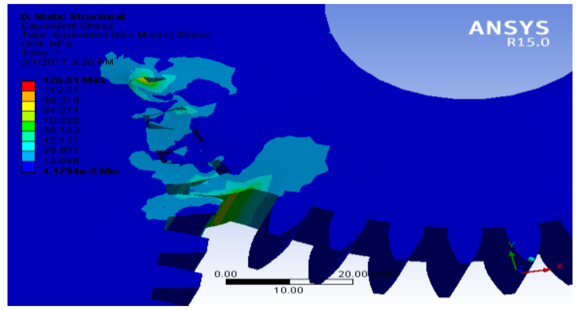
Total Deformation-HCR at 2500 rpm



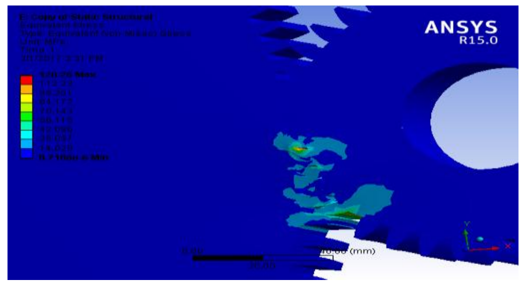
From the below graph we can observe that deformation is less in low contact ratio gears when compared to high contact ratio gears. It is because of the variation in the thickness of gear. in HCR and LCR we have taken the same pitch circle diameter but we have varied the number of teeth. so thickness 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.



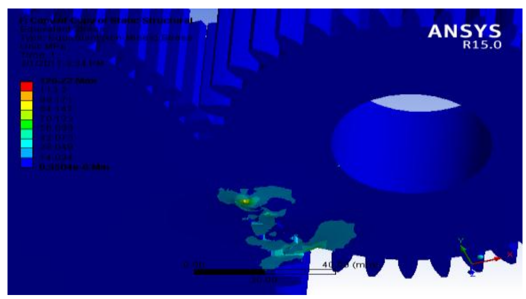
Equivalent Stress (VON-MISES) -HCR at 10000rpm



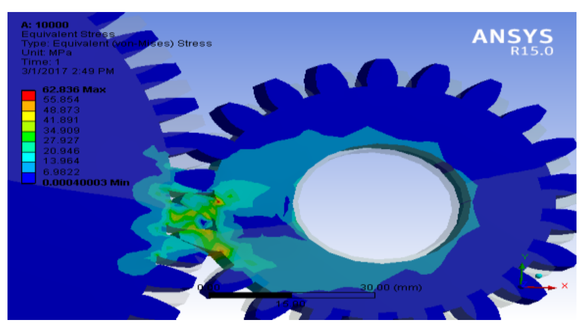
Equivalent Stress (VON-MISES) -HCR at 5000rpm



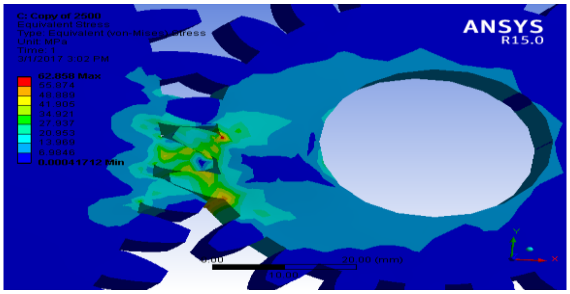
Equivalent Stress (VON-MISES) -HCR at 2500rpm



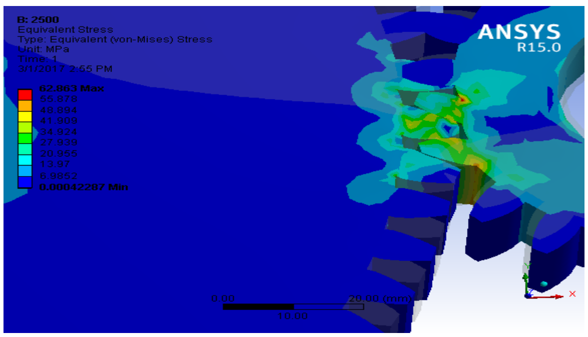
Equivalent Stress (VON-MISES) -LCR at 10000rpm

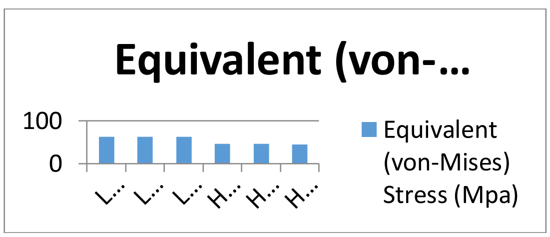


Equivalent Stress (VON-MISES) -LCR at 5000rpm



Equivalent Stress (VON-MISES) -LCR at 2500rpm

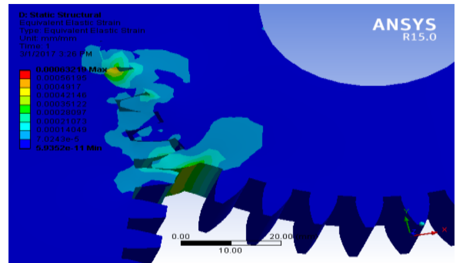


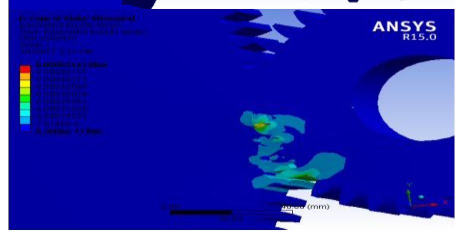


In the above graph of Equivalent Stress (VON-MISES) for HCR and LCR gears at different rotational velocities we can see the very slight variation in the stress with the rotational speed for same set of gear train. But we can see a tremendous decrement in the stress values for HCR gears when compared 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.

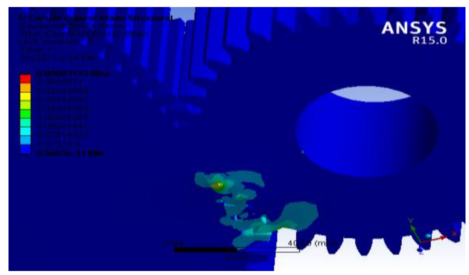
Equivalent elastic Strain-HCR at 10000rpm

Equivalent elastic Strain-HCR at 5000rpm

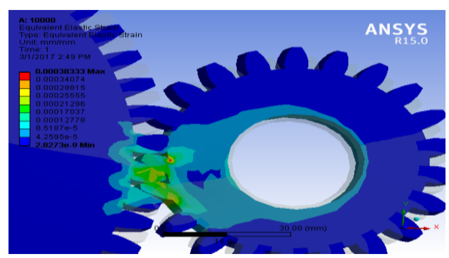




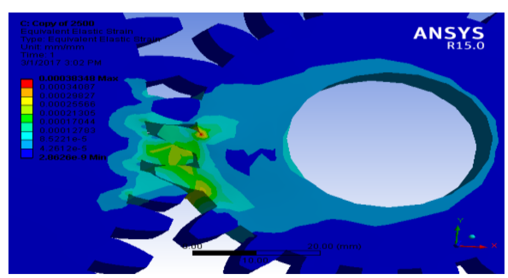
Equivalent elastic Strain-HCR at 2500rpm



Equivalent elastic Strain-LCR at 10000rpm



Equivalent elastic Strain-LCR at 5000rpm



Equivalent elastic Strain-LCR at 2500rpm

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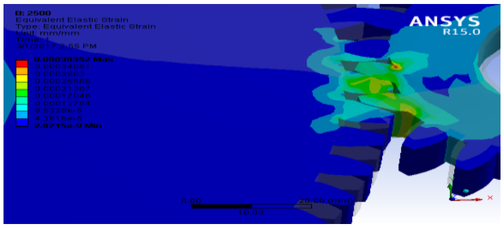
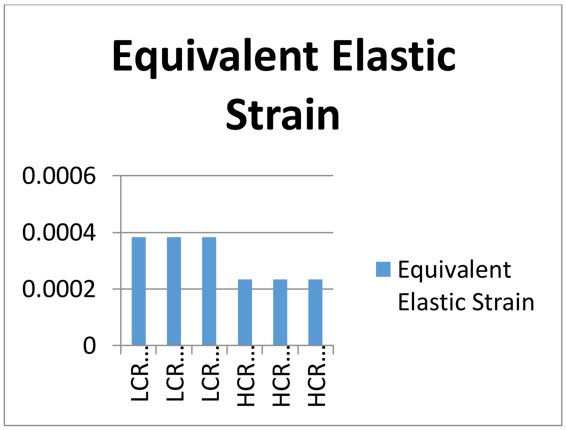
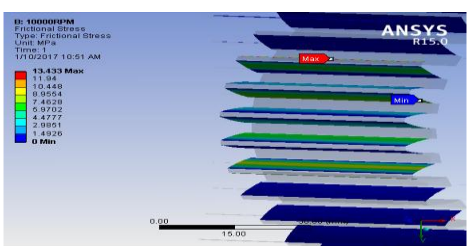




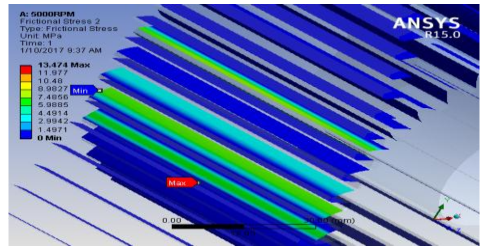
In the above graph we can see the variation of strain between HCR- gears and LCR-gears at different rotational velocities. we can see more than 40% difference in strain in HCR-gear set when compared to LCR-gear set.

Frictional Stress

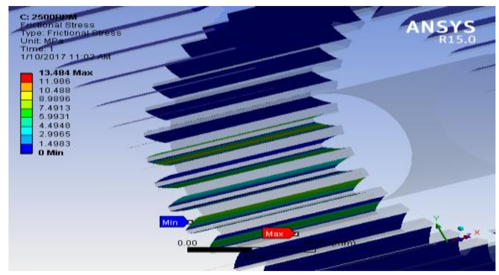
Frictional Stress of HCR-10000rpm



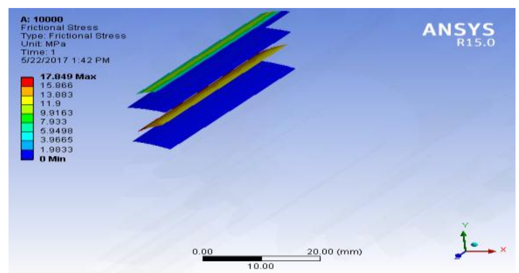
Frictional Stress of HCR-5000rpm



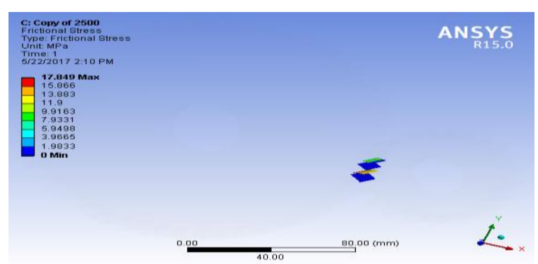
Frictional Stress of HCR-2500rpm



Frictional Stress of LCR-10000rpm



Frictional Stress of LCR-5000rpm



Frictional Stress of LCR-2500rpm

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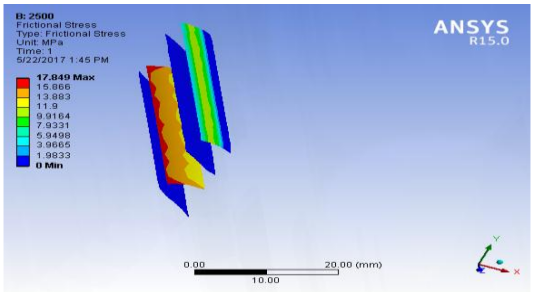

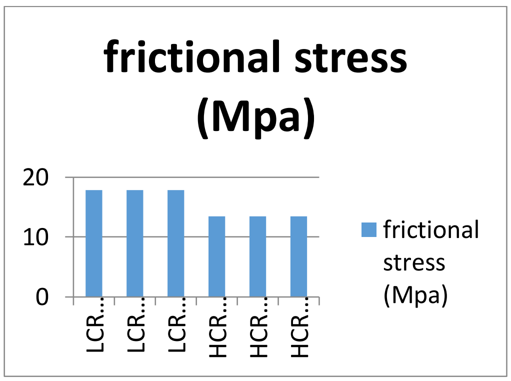

we can see more than 20% decrement in the frictional stress values between HCR and LCR- gear sets this is due the load shared by the frictional load among teeth. in case of lcr gear set the hole frictional load is shared by two pairs of teeth whereas the same has been shared by more than four pairs of 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.

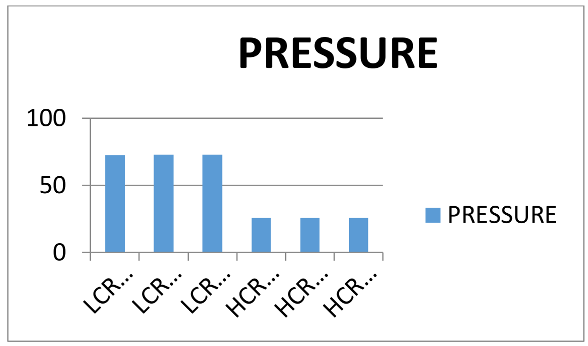

TABLE OF RESULTS:

	LCR GEARS-2500	LCR GEARS-5000	LCR GEARS-10000	HCR GEARS-2500	HCR GEARS-5000	HCR GEARS-10000
TOTAL DEFORMATIO N (mm)	0.011696	0.011696	0.011698	0.013302	0.013302	0.013303
Equivalent Elastic Strain	0.00038352	0.00038348	0.00038333	0.00023449	0.00023442	0.0002341 8

Equivalent (von-Mises) Stress (Mpa)	62.863	62.858	62.836	45.462	45.449	45.398
Shear Stress (Mpa)	11.89	11.83	11.8	16.353	16.341	16.295
penetration	3.81E-06	3.81E-06	3.81E-06	4.99E-05	4.99E-05	5.00E-05
frictional stress (Mpa)	17.884	17.856	17.849	13.484	13.474	13.433
PRESSURE (Mpa)	72.416	72.445	72.56	25.359	25.357	25.348

VI. 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.

REFERENCES

1. Elkholy,A.H., “Tooth load sharing in high contact ratio spur gears.” Trans. ASME. Journal of Mechanisms, Transmission and Automation in Des.107:11-16.

2. Rosen, M.K., and Frint, H.K., Design of high contact ratio gears, J. Am. Helicopter Soc., (1982) 65-73.

3. Leming, J.C., “High Contact Ratio (2+) Gears”, Gear design manufacturing and inspection manual, SAE, Warrendale PA, 1990 pp. 73–83.

4. Sivakumar, P., Hanumanna, D. and Gopinath, K., High contact ratio gearing concept for military tracked vehicle application, 17th national convention of production engineers on Recent trends in design and manufacture of gears and bearings, Chennai (2002) pp. 71-78.

4a. Mohanty, S.C., Tooth load sharing and contact stress analysis of high contact ratio spur gears in Mesh. IE Journal 84 (2003) 66 – 71.

5. Coy, J.J., and Chao, C.H., “ A method of selecting grid size to account for Hertz Deformation in Finite Element Analysis of Spur Gears,” 1982, ASME J. Mech Des. 103, pp. 759-459.

6. Wang, J., Howard, i., Finite Element Analysis of High Contact Ratio Spur Gears in Mesh, ASME J. of Tribology.,127, pp.469-482, 2005.

7. Buckingham, E., Analytical mechanics of gears, Dover publication, New York, 1963.

8. Cornell,R.W., Compliance and stress sensitivity of spur gear teeth, ASME Trans., J. Mech. Des., April 1981.

9. Arafa, M.H., and Megahed, M.M., “ Evaluation of spur gear mesh compliance using the finite element method,” Proc Instn Mech Engrs Vol 213 Part C, pp.569-579.

10. Walton, D., Tessema, A.A., Hooke, C.J., and Shippen, J. Load sharing in metallic and non metallic gears. Proc. Instn Mech. Engrs, Journal of

Mechanical Engineering Science, 1994, 208 (C2), 81-87.

11. ANSYS Release 11.0, Structural analysis guide, ANSYS Inc., Canonsburg, PA – 15317, USA 200

12. Li, S., Effect of addendum on contact strength, bending strength and basic performance parameters of a pair of spur gears, Mechanism and Machine Theory, (2008).

13. Dolan, T.J., and Broghamer, E.J., A photoelastic study of the stresses in gear tooth fillets, Bulletin No.355. University of Illinois, Engineering experiment station, 1942.

14. Cornell, R.W., and Westervelt, W.W., Dynamic Tooth Loads and Stressing for High – Contact Ratio Spur Gears, American Society of Mechanical Engineers, pp. 77-DET-101, 1977.