CONTACT STRESS ANALYSIS OF SPUR GEAR BY ANALYTICAL AND FINITE ELEMENT METHOD

[1]Mr.Suhas D.Borate, [2]Prof.Sameer D.Katekar, [3]Prof.Prasad P.Kulkarni

[1]Student, [2]Assistant Professor, [3] Assistant Professor

[1]Department of Mechanical Engineering,

SKN Sinhgad College of Engineering,Pandharpur,India

Abstract: The gears while transmitting the power generates high stresses at the mating positions over the teeth as they amend the rate of rotation of machine shaft. The axis of rotation for High speed machinery is the optimal medium for low energy loss and high accuracy. Toothed Spur gears are used to transmit the power with high velocity ratio. Various methods are used to find out contact stresses such as Hertz contact stresses, Lewis bending Equation, AGMA Equation. Most of the research work attempted on mathematical contact stress analysis and Compared with finite element analysis. The purpose of this paper is to find out the lite contact stress analysis of spur gear by analytical and Finite Element Method. Hertzian equation values are compared with the Finite Element Analysis (FEA) results. For FEA majorly Hypermesh, Abaqus and Ansys tool used. Authors have used various approaches and means to conclude their objective of finding out the contact stress and optimization using FEA. This review paper explores theoretical, numerical and analytical method for the spur gear analysis and compare the Hertz equation value with Finite Element Method value for validation purpose.

***Keywords*- Spur Gear, Contact Stress, Hertz Equation, Finite Element Analysis.**

I. INTRODUCTION

Gears are the most common means of transmitting power in the modern mechanical engineering world. They vary from a tiny size used in watches to the large gears used in lifting mechanisms and speed reducers. They form vital elements of main and ancillary mechanisms in many machines such as automobiles, tractors, metal cutting machine tools etc. Toothed gears are used to change the speed and power ratio as well as direction between input and output. The objective of the work is to begin the process of evaluating the potential benefits of asymmetric involute gear teeth and optimized root-- fillet geometry for industrial and automobile applications. This involves not only quantifying performance improvements achieved by these concepts, but evaluating the practicality of manufacturing gears with asymmetric teeth and optimized root-- fillet geometry for industrial, automobile and aerospace applications. There has been a lot of research activity on spur gears with asymmetric teeth. New gear designs are needed because of the increasing performance requirements, such as high load capacity, high endurance, low cost, long life and high speed. In some applications, gears experience only unidirectional loading. In this instance, the geometry of the drive side does not have to be symmetric to the coast side. This allows for the design of gears with asymmetric teeth. These gears provide flexibility to designers due to their non-Standard design. If they are correctly designed, they can make important contributions to the improvement of designs of gears. An asymmetric spur gear drive means that larger and smaller pressure angles are applied for the driving and coast sides. The two profiles of a gear tooth are functionally different for most gear drives. The workload on one side of profile is significantly higher than the other gears. The design intent of asymmetric gear teeth is to improve performance of the primary drive profiles at the expense of the performance for the opposite coast profiles. In many cases the coast profiles are more lightly loaded and are loaded only for a relatively short duration. Asymmetric tooth profiles make it possible to simultaneously increase the contact ratio and operating pressure angle in the primary drive direction beyond the conventional gears' limits. The main advantage of asymmetric gears is contact stress reduction on the drive flanks, resulting in reduced gear weight and higher torque density. The items addressed in this article include analysis, design, manufacture and test of gear test specimens with asymmetric teeth.

A primary requirement of gears is the constant angular velocities or proportionality of position transmission. Precision instruments require positioning fidelity. High-speed and/or high-power gear trains also require transmission at constant angular velocities in order to avoid severe dynamic problems. Constant velocity (i.e. constant ratio) motion transmission is defined as conjugate action of the gear tooth profiles. A common normal to the tooth profiles at their point of contact must, in all positions of the contacting teeth, pass through a fixed point on the line-of- centers called the pitch point any two curves or profiles engaging each other and satisfying the law of gearing are conjugate curves.

Dhavale A.S *et.al.*study deals with minimizing the stress in root fillet region of the gear tooth. Inserting stress relief features (holes) of different diameter at different locations (Beside root fillet) & varying number of holes. A maximum of 15% reduction in maximum principal stress is obtained. As the Number of holes & hole diameter increases the maximum principal stress decreases but after a certain extent the strength of the gear reduces. Rubén d. Chacón *et.al.* study deals with the stresses in the contact zone among a couple of spur gears using the FEM & results are evaluated by comparing the Hertz contact theory for curved elements in two dimensions with the theory provided by the AGMA for SG. The contact pressure and stress state are highest for higher points on the involute and lower were a single pair of teeth assumes the full load transmitted, and minimal for the contact at the pitch point. The contact stresses obtained by FEM and the theory of Ali Raad Hassan *et.al.* Study deals with Contact stress analysis between two spur gear teeth was considered in different contact positions, representing a pair of mating gears during rotation. He considered steel (C45) material The maximum stress result obtained from AGMA stress calculation method was 587MPa while the maximum contact stress obtained from the finite element contact analysis was 595MPa (case five) under the same conditions, it is clear that the agreement is good. The results shows a high value of contact stress in the beginning of the contact, and then it starts to reduce until it reaches the location of single tooth contact, here it increased to the maximum value of the contact, but exactly after exceeding this single contact region it was reduced. At the end of the contact, the stress increased suddenly to a high value almost close to the maximum value, at this stage a sliding was occurred in the contact region at the maximum stress points.Vivek karaveer *et.al.*study deals with stress analysis of mating teeth of spur gear to find maximum contact stress in the gear teeth. The results obtained from Finite Element Analysis (FEA) are compared with theoretical Hertzian equation values. For the analysis, steel and grey cast iron are used as the materials of spur gear Also deformation for steel and grey cast iron is obtained as efficiency of the gear depends on its deformation. The results show that the difference between maximum contact stresses obtained from Hertz equation and Finite Element Analysis is very less and it is acceptable. The deformation patterns of steel and grey cast iron gears depict that the difference in their deformation is negligible.

II MATERIAL PROPERTY

In this, grey cast iron is used as the spur gear materials. The material properties of grey cast iron are given in Table1.

Table 1 Material Property of Grey Cast Iron

Material Property	Symbol	Value	Unit
Density	ρ	7100	Kg/m^3
Poisson Ratio	ϑ	0.26	-
Young's Modulus	E	165e3	MPa
Tensile Yield	S_{yt}	250	MPa
Tensile Ultimate	S_{ut}	350	MPa

III.MODELING OF SPUR GEAR

In this study, maximum contact stress is determined, during the transmission of torque of 5936.5 N-mm by grey cast iron spur gear, using finite element analysis. The spur gear is sketched and modeled in the ANSYS Design Modeler. The dimensions of the gears are given in Table 2.

Table 2 Dimensions of Spur Gear:

Dimension	Symbol	Value (For Spur Gear)	Value (For Pinion Gear)	Unit
Number of Teeth	Z	100	56	
Pitch Circle Diameter	D	160	87.8	mm
Pressure Angle	ϕ	14.5	14.5	degree
Addendum Radius	R_A	80.95	46.2	mm
Dedendum Radius	R_D	77.5	42.5	mm

IV. Theoretical Calculation of Contact Stresses by Analytical Method (Hertz Equation)

Hertz equation is used to determine the contact stresses in the mating teeth of gear. Hertzian equation for contact stress in the teeth of mating gears is given by,

$$\sigma_c = \frac{\sqrt{F\left(1+\frac{R1}{R2}\right)}}{\sqrt{R1 B \pi\left[\frac{1-\vartheta 1^{2}}{E1}+\frac{1-\vartheta 2^{2}}{E2}\right] \sin\phi}} \quad \text{.........(1)}$$

Where, σ_c is the contact stress in mating teeth of spur gear, F is the force, and R_1,R_2 are pitch radii of two mating gears, B is the face width of gears, ϕ is the pressure angle, ϑ_1,ϑ_2 are the Poisson ratios and E_1, E_2 are the moduli of elasticity of two gears in mesh.

Allowable maximum stress is given by,

$$\sigma_c = \frac{\sigma c}{FOS} \quad \text{..................(2)}$$

Here FOS is the factor of safety which can be taken from the ANSYS results or other tables. Equation (2) gives the allowable maximum contact stress in the mating gears. In this paper, factor of safety is taken from ANSYS results. But minimum factor of safety from the ANSYS results is preferred in order to get accurate allowable maximum contact stress at the point of contact of gear teeth.

The relation between Power and RPM is:

$$P = \frac{2\pi NT}{60 \times 10^{\wedge}3}$$

$$T = \frac{P \times 60 \times 10^{\wedge}3}{2\pi N} \text{Si}$$

We have,

Power, P= 1 HP= 746 Watt

N = RPM of driver (1200 RPM)

π = 3.14

Therefore,

Torque T = 5936.48 N-mm.

Now, Torque is given by,

Torque = Force (F) * Shaft Radius (Rs)

5936.48 (N-mm) = * 22.2 (mm)

Force, F=267.40 N

By using Hertz Equation,

$$\sigma_c = \frac{\sqrt{F(1+\frac{R1}{R2})}}{\sqrt{R1B\pi}\left[\frac{1-\vartheta 1^{\wedge}2}{E1}+\frac{1-\vartheta 2^{\wedge}2}{E2}\right]Sin\phi}$$

$$\sigma_c = \frac{\sqrt{267.40(1+\frac{80}{44.8})}}{\sqrt{80*20*\pi}\left[\frac{1-0.26^{\wedge}2}{165*10^{\wedge}3}+\frac{1-0.26^{\wedge}2}{165*10^{\wedge}3}\right]Sin(14.5)}$$

σ_c =238.38MPa

Allowable maximum stress is given by,

$$\sigma_a = \frac{\sigma c}{FOS}$$

Here, Factor of Safety (FOS) is selected as 1 for Grey Cast Iron (Vivek Karaveer *et al 2013*)

$$\sigma_a = \frac{238.38}{1}$$

σ_a =238.38MPa

V. FINITE ELEMENT ANALYSIS:

5.1 Geometry

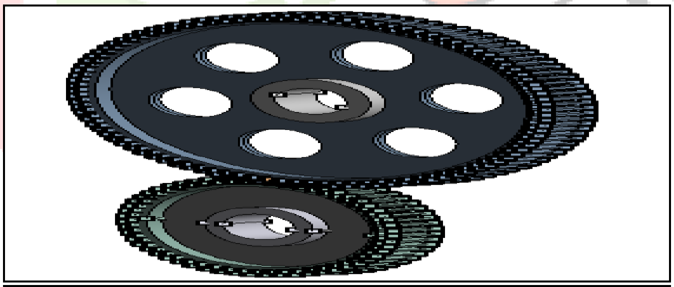
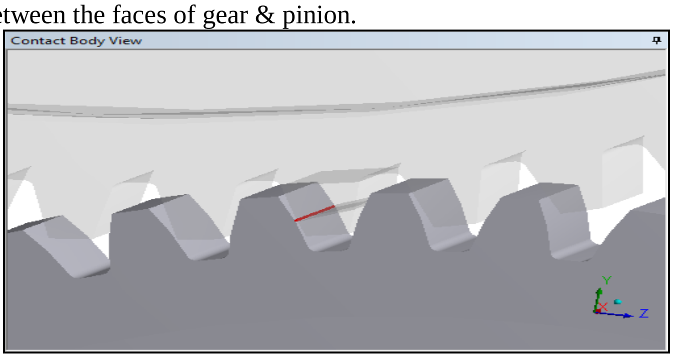


Fig.1 Assembly of Spur Gear

5.2 Contact

Bonded type contact is defined between the faces of gear & pinion.



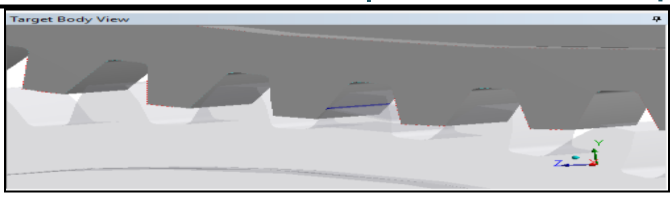


Fig.2 Surface Contact between Two Gears

5.3 Meshing

The assembly is meshed with tetrahedron higher order elements with 1207498 nodes & 492052 elements.

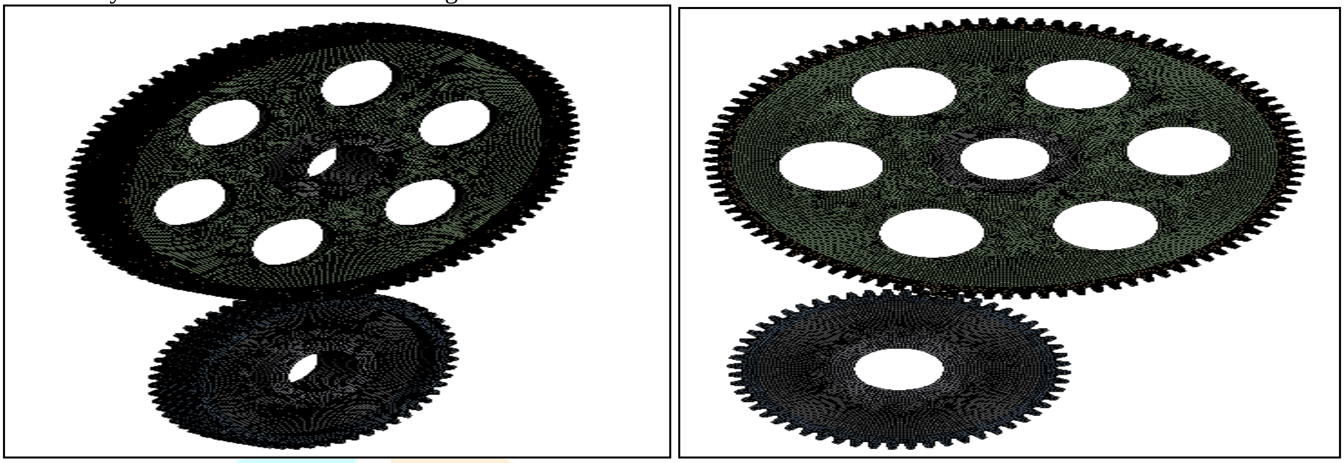


Fig.3 Meshing

5.4 Boundary Conditions

The shaft mounting region of the gear is fixed & shaft mounting region of the pinion is defined with frictionless support which is free to rotate. The moment of 5936.5 N-mm is applied at the shaft mounting region of pinion in clockwise direction w.r.t x axis as shown in the fig below.

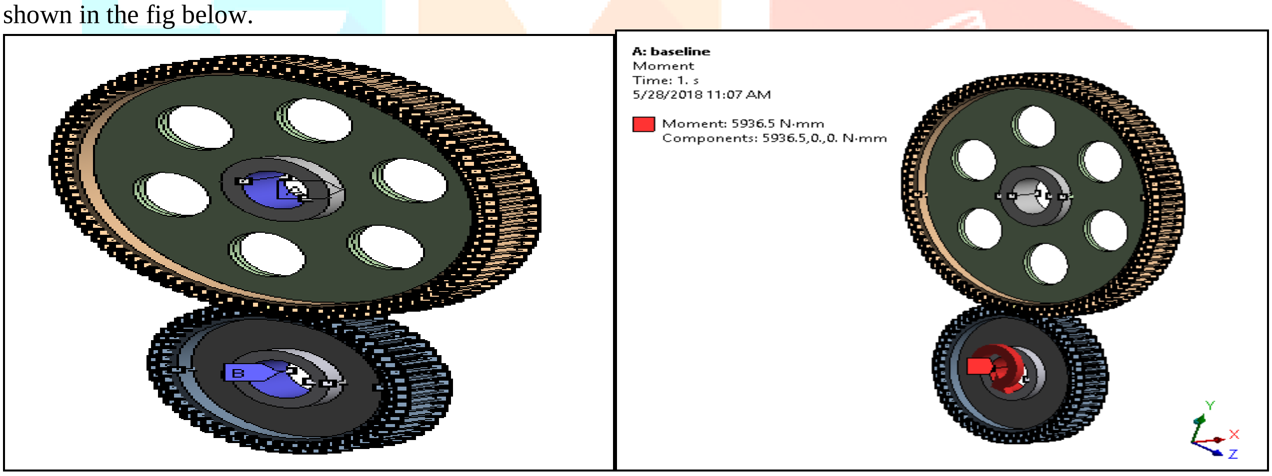
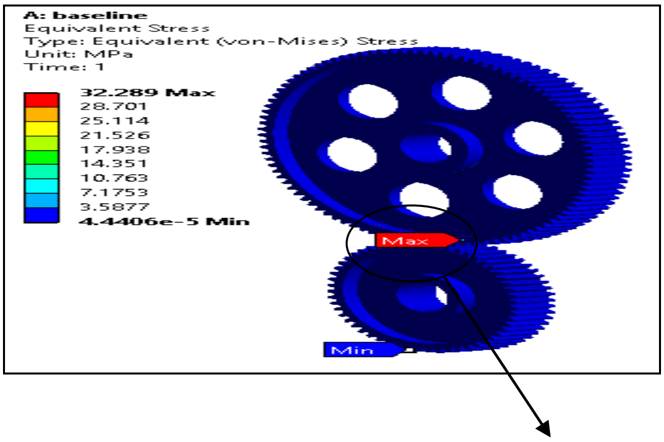


Fig. 4 Boundary Condition

VI. RESULTS AND DISCUSSION

6.1 Equivalent Stress

The maximum equivalent stress is found to be 32.289 MPa at the contact region & minimum at other than contact region



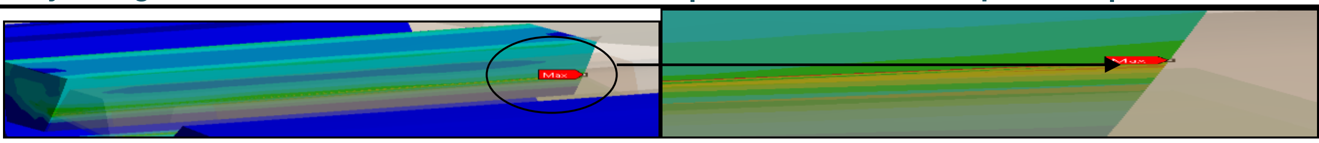


Fig.5 Equivalent Stress

6.2 Total Deformation

The Maximum total deformation is found to be 0.0028517 mm at the tooth region of the pinion & minimum at the fixed region of the gear.

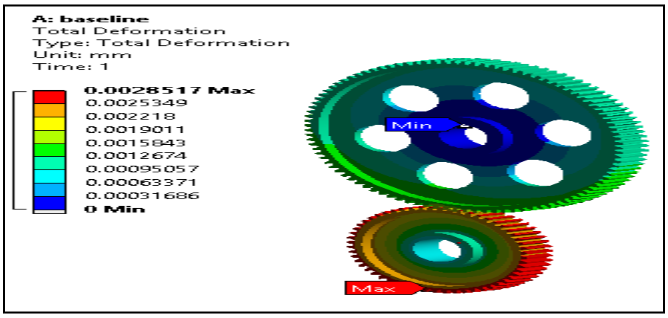


Fig.6 Total Deformation

6.3 Contact Pressure

The averaged maximum pressure at the contact region is found to be 238.38 M Pa.

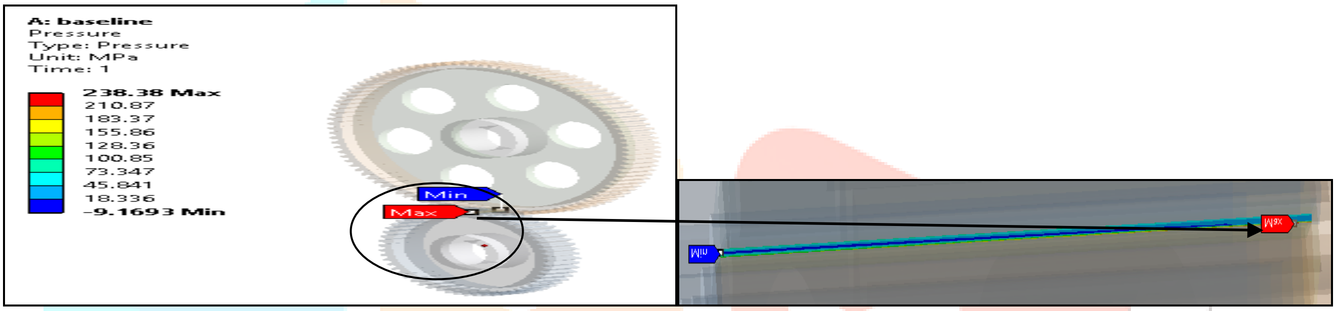


Fig.7 Contact Pressure

Comparison of maximum contact stresses, for grey cast iron, obtained from Hertz equation and ANSYS 14.5 is given in Table 3.

Table 3 Comparison of maximum contact stress obtained from Hertz equation and ANSYS 14.5

Gear	$\sigma_{a(Hertz)}$ (MPa)	$\sigma_{a(ANSYS)}$ (MPa)	Difference (%)
Grey CI	228.4	238.38	4.01

CONCLUSION

Here the theoretical maximum contact stress is calculated by Hertz equation. Also the finite element analysis of spur gear is done to determine the maximum contact stress by ANSYS 14.5. It was found that the results from both Hertz equation and finite element analysis are comparable.

ACKNOWLEDGMENT

I take this opportunity to thank Prof.S.D.Katekar and Prof.P.P.Kulkarni for valuable guidance and for providing all the necessary facilities, which were indispensable in completion of this work. Also I sincerely thanks to all the authors who worked on chain link weight optimization and material optimization.

REFERENCES

1) "Study of Stress Relief Features at Root of Teeth of Spur Gear", Dhavale A.S., Abhay Utpat, and IJERA ISSN: 2248-9622 Vol. 3, Issue 3, May-Jun 2013, pp.895-899.

2) "Analysis of stress due to contact between spur gears", rubén d. Chacón, advances in computational intelligence, man-machine systems and cybernetics, isbn: 978-960-474-257-8.

3) "Design and finite element method analysis of gear mechanism for raw material crushing in plastic industry", prof. Ajitabh pateriya, ijpret, 2015; volume 3 (7): 48-55.

4) "Contact Stress Analysis of Spur Gear Teeth Pair", Ali Raad Hassan, World Academy of Science, Engineering and Technology, Vol: 3, No: 10, 2009.

5) "Modelling and Finite Element Analysis of Spur Gear", Vivek Karaveer Å, International Journal of Current Engineering and Technology ISSN 2277 – 4106.

6) "A Study on the Influence of using Stress Relieving Feature on Reducing the Root Fillet Stress in Spur Gear", Nidal H., Proceedings of the 2014 International Conference on Mathematical Methods, Mathematical Models and Simulation in Science and Engineering.

7) "Contact stress analysis for a pair of mating gears", Seok-Chul Hwang, Mathematical and Computer Modelling 57 (2013) 40–49.

8) "A Finite Element Approach to Bending, Contact and Fatigue Stress Distribution in Helical Gear Systems". S. Jyothirmai, Procedia Materials Science 6 (2014) 907 – 918.

9) "An investigation of contact stresses and crack initiation in spur gears based on finite element dynamics analysis", W.J. Qin, International Journal of Mechanical Sciences 83 (2014)96–103.

10) "Dynamic analysis of spur gear teeth", v. Ramamurti, compvrers & sfrucrures vol. 29, no. 5, pp. 831-843. 1988.

11) "A finite element analysis of bending stresses induced in external and internal involute spur gears",j. D. Andrews, journal of strain analysis vol 26 no 3 1991.

12) "Comparative Finite Element Analysis of Reconstructed New and Worn Tooth of Spur gear", Atul Kumar, P. K. jain and P. M. Patha.,

13) Proceedings of the 1st International and 16th National Conference on Machines and Mechanisms (iNaCoMM2013), IIT Roorkee, India, Dec 18-20 2013.

14) "Analysis of Spur Gear Using Finite Element Analysis", Mrs. C.M. Meenakshi, akash Kumar, Apoorva Priyadardsi, Digant Kumar Dash and Hare Krishna, Middle-East Journal of Scientific Research 12 (12): 1672-1674, 2012 ISSN 1990-9233.

15) " Contact stress analysis of spur gear ", Mr.Bharat Gupta, Mr.Abhishek choubey, 2012, International Journal of Engineering Research & Technology (IJERT), ISSN: 2278-0181, Vol. 1 Issue 4, June – 2012.

16) "Modal and stress analysis of gear train design in portal axle using finite element modeling and simulation", Jong Boon Ooi1,Xin Wang, Ching Seong Tan, Jee-Hou Ho, Ying Pio Lim, Journal of Mechanical Science and Technology 26 (2) (2012) 575~589.

17) "Contact Stress Analysis of Helical Gear by Using AGMA and ANSYS", S.Sai Anusha, P.Satish Reddy, P.Bhaskar, M Manoj, International Journal of Science Engineering and Advance Technology, Vol.2, Issue 12, (2014).