An Investigative Study of Spur Gear Failure by FEA and Photoelastic Method

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Abstract— Gear is most effective part of transmission system due to efficiency and reliability. Gear is used for high load in machine tools. These gears are continuously operated under specified conditions. If gear failure occurs, it is due to pitting failure and scoring failure. In this paper, pitting failure has been studied for a gear. Modeling of gears in CATIA V5 is done by parametric formulation. The gear is analyzed in ANSYS for deformation and max contact stress which causes pitting. Experimental Analysis is done using photoelastic method on photoelastic apparatus.

Keywords – ANSYS, FEA, parametric, pitting, Photoelastic Method**.**

1. INTRODUCTION

Gear is used in gear train for power transmission of high loads. Gear failure occurs due to bending and pitting failures. Pitting is the surface fatigue failure which occurs due to many repetitions of Hertz contact stresses. The failure occurs when the surface contact stresses are higher than the endurance limit of the material, wrong or insufficient lubrication and misalignment of gears. Severe case is acting of high loads and stresses been developed continuously on pitch point area. The failure starts with the formation of pits which continue to grow resulting in the rupture of the tooth surface. In order to avoid the pitting, the dynamic load between the gear tooth should be less than the wear strength of the gear tooth.

2. LITERATURE REVIEW

Bharat Gupta [1] in his work the gear tooth failure take place if contact stresses in the gear are higher than the wear strength of the gear. He concludes this maximum contact stress decreases with increasing module of gear. The Contact stresses are higher at the pitch point of the gear.

M. Raja Roy, 2014 [2] in this paper the analysis of contact stresses induce on the spur gear train for different value of module. In this research paper developed one VISUAL BASIC program for calculate the contact stresses for different parameter like module, power & speed etc.

Ali Raad Hassan [3], In this paper, the contact stresses are calculated each 3° rotation of pinion from first location of contact at 0° to last location 30° .The observation of result gives

the 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, then it increased to the maximum value of the contact at pitch point, after that stresses start to reduce the contact ratio reduces.

Yadav S.H.[4] in this paper work select one planetary gear train used in the transmission gear box for analysis that gear train can be failed due to pitting failure. The contact stresses are reducing up to the lower than surface endurance limit of the gear material.

Konstandinos G. Raptis, Theodore N. Costopoulos, Georgios A. Papadopoulos and Andonios D. Tsolakis [5] in this paper, work done for calculating contact stresses of gear by experimental method using photo elasticity. For this research work four specimens of gear were manufactured by ISO standard having different no of teeth with same module and width. The contact stresses of these specimens are calculated by photo elasticity experiment.

Ali Kamil Jebur [6] experimental result was express by plotting the graph between maximum contact stresses Vs contact position. The experimental analysis is done by using the D.C. servomotor and planting the strain gauges in the tooth of the gear made form polyimide materials.

Abhijit Mahadev Sankpa and M. M. Mirza [7] has given contact stresses of gear are calculated for three different loads by using FEA and Experimental method This model is tested by polaroscope and contact stresses are calculated.

C.M. Meenakshi, S. Balaguru and N. Senthil Kumar,[8] in this paper the contact stresses are also calculated by Hertz equation and bending stresses by Lewis formula and AGMA standard.

Vivek KaraveerÅ, Ashish MogrekarÅ and T. Preman Reynold Joseph,[9] result that have modeling and analysis is done in Ansys. The stresses are also calculated by Hertz's Equation are compared with FEA result.

3. SELECTION OF MATERIAL

The lathe machine gearbox has been selected for analysis due to failure reasons frequently. The material of gears selected was cast iron used for durability. But its strength was low so EN24 material is selected.

4. OBJECTIVES

1. Study the causes of gear failure due to pitting.
2. Prepare the parametric model using CAD software.
3. Study the alternative material for reducing the failure.
4. 3D parametric model preparation using CAD interface.
5. Analyze the models in ANSYS Workbench interface for find out the maximum stress and deformation.
6. Experimentation for existing material using photoelastic method.
7. Validation through analytical and experimental method.

5. DESIGN BASED ON HERTZ CONTACT STRESS

One of the main gear tooth failure is pitting which is a surface fatigue failure due to many repetition of high contact stresses occurring in the gear tooth surface while a pair of teeth is transmitting power. Contact failure in gears is currently predicted by comparing the calculated Hertz contact stress to experimentally determined allowable values for the given material. The method of calculating gear contact stress by Hertz's equation originally derived for contact between two cylinders.

In machine design, problems frequently occurs when two members with curved surfaces are deformed when pressed against one another giving rise to an area of contact under compressive stresses. Of particular interest to the gear designer is the case where the curved surfaces are of cylindrical shape because they closely resemble gear tooth surfaces.

The value of stress acting at pitch point is given by,

$$Pcmax = \frac{4x\ F}{BxLx\pi}$$

Where,

$$B = \sqrt{\frac{8*F}{\pi*L} * \frac{(1-\vartheta 1^2/E1)+(1-\vartheta 2^2/E2)}{\frac{1}{D1}+\frac{1}{D2}}}$$

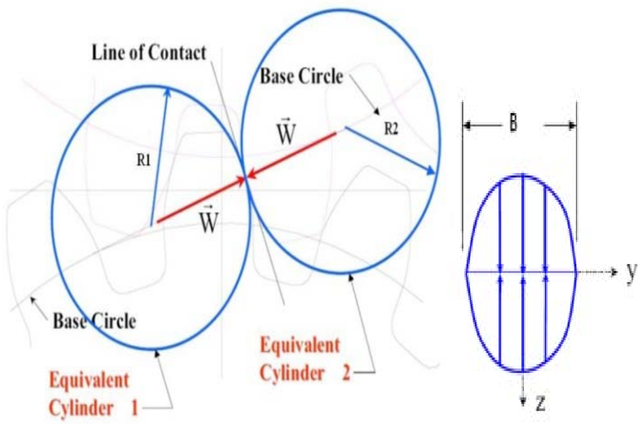

Fig 1: Ellipsoidal–prism pressure distribution

Here, F is the applied force, υ_1, υ_2 are Poison's ratio of the two materials of the cylinders with diameters D_1 and D_2, and E_1 and E_2 are the respective modules of elasticity.

Putting the values of B and assuming a value of 0.3 to poison's ratio in above equation, and by replacing diameters by respective radii,

$$Pcmax = \sqrt{0.35 * \frac{F}{L} * \frac{\left(\frac{1}{R1}+\frac{1}{R2}\right)}{\left(\frac{1}{E1}+\frac{1}{E2}\right)}}$$

6. ANALYTICAL CONTACT STRESS ANALYSIS OF SPUR GEAR

SR NO	INPUT PARAMETERS	SYMBOL	VALUE
1	MODULE	M	2.2MM
2	INPUT SPEED	P	2KW
3	TRANSMISSION RATIO	I	2
4	PINION SPEED	N_1	200
5	NO OF TEETH ON PINION	Z_1	20
6	PRESSURE ANGLE	A	20
7	MATERIAL OF GEAR	EN 24	
8	MATERIAL OF PINION	EN24	

Table 1: Input parameters for the sample calculation

Sample Calculation for Module Size of 2

Nominal torque on the pinion shaft = 95.49296 N m

Torque (T) = Force (F) * Radius (Ra)

95492 N mm = Force (F)*40 mm
{40mm is the radius of the Lever}

Force (F) = 2387.3241 N

Tangential Force: Ft = 2000 * T /d
= 2000*95.492 / 40
Ft = 4775 N

The Hertzian contact stress is given by,

$$Pp = Ym * Yp \sqrt{\frac{Ft}{b * d1} * \frac{u+1}{u}}$$

Where, *Ym* is the material Co-efficient,

$$Ym = \sqrt{0.35 * \frac{2 * (E1 * E2)}{(E1 + E2)}}$$

$$Ym = \sqrt{0.35 * \frac{2*(210*10^2*210*10^2)}{(210*10^2+210*10^2)}}$$

Ym = 85.73

Where, *Yp* is the Pitch point Co-efficient.

$$Yp = \sqrt{\frac{1}{\cos\alpha * \tan\alpha}}$$

$$Yp = \sqrt{\frac{1}{\cos 20° * \tan 20°}}$$

= 1.76

D1 = Module (m) * Number of the teeth (Z1)
= 2 * 20
= 40

D2 = Module (m) * Number of the teeth
= 2 * 40
= 80

u = D2/D1
= 2

Hertzian contact stress (Pp) = 451.4740 MPa

6. MODELLING OF SPUR GEAR:

Module	2mm
No of teeth on pinion	20
No of teeth on Gear	41
Pressure angle (Deg)	20

Table 2. Input parameter for program

The input parameter like module and number of teeth are input to the formulation. The module is 2.1 mm and number of teeth for pinion are 20. The parameter like module and number of teeth are input to the formulation. The module is 2 mm and number of teeth for gear are 41. Both the gear and pinion are called in assembly module.

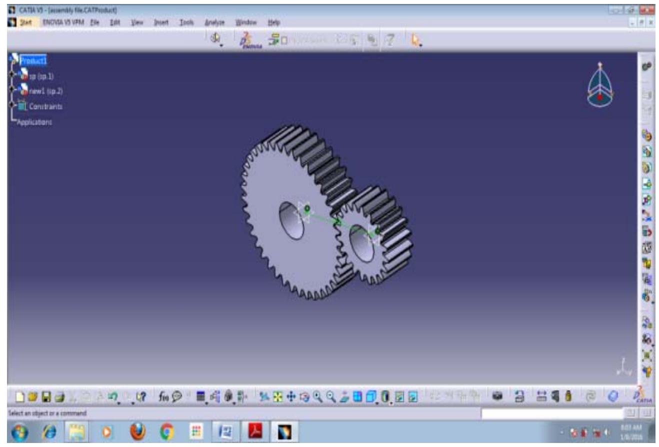

Fig 2: Gears in assembly module of CATIA

Above figure shows the gear assembly model in CATIA V5 R21 interface. For FE analysis, IGES file format exported to ANSYS 14.5.

7. FEA IN ANSYS

The assembly file is imported in .iges format in ANSYS workbench 14.5. All the static structural preference for analysis are chosen. Material of steel gear is entered with value of poison's ratio of 0.3 and modulus of elasticity as 207E5.

The gears are made in proper alignment and frictionless support is applied as boundary condition. The gears are meshed by quadrilateral 4 node material with control size and fine meshing is done.

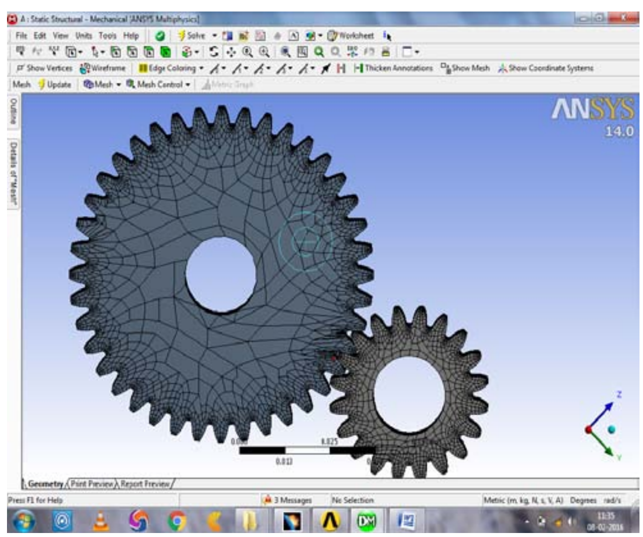

Fig 3: Meshing of gears

Figure shows the meshed model of gear assembly. Tetrahedral element used for meshing the gears. Fine mesh used at the tooth profile and coarse mesh used towards center. Mesh size considered is 5 mm.

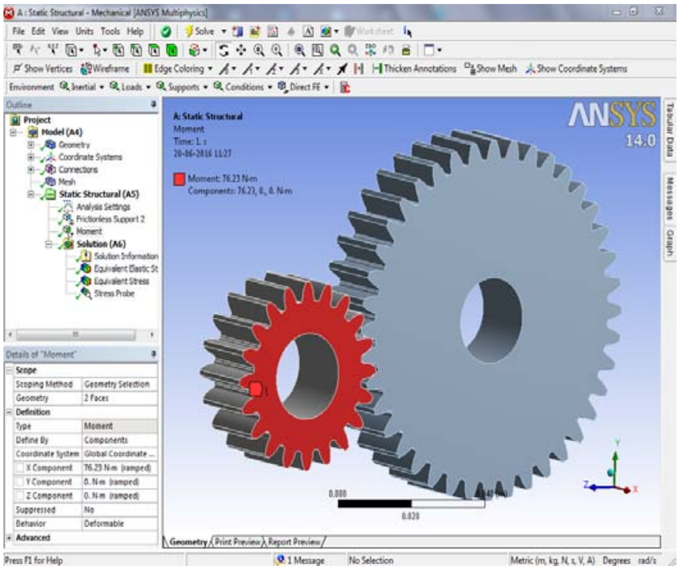

Fig 4: Torque applied on pinion

As shown in figure, torque applied on the pinion. Torque is applied as derived from calculation analysis. A load of about 451.4740 N mm is applied on both faces of driver gear.

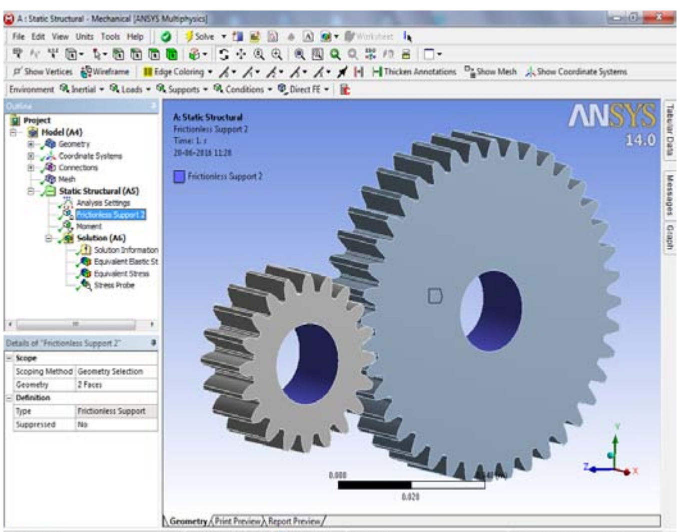

Fig 5: Frictionless support given at the center of the gear

AS shown in above figure, Frictionless support is defined at the center of the gear. Frictionless support shown in blue color.

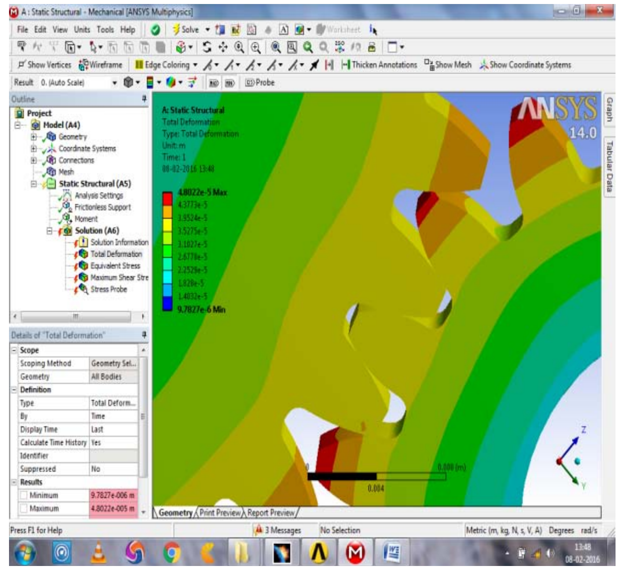

Fig 6: Total deformation and its distribution on gears

Figure shows the total deformation of the gear assembly. Maximum deformation is 4.8 e^{-5} mm. Maximum deformation is shown in red color and minimum in blue color.

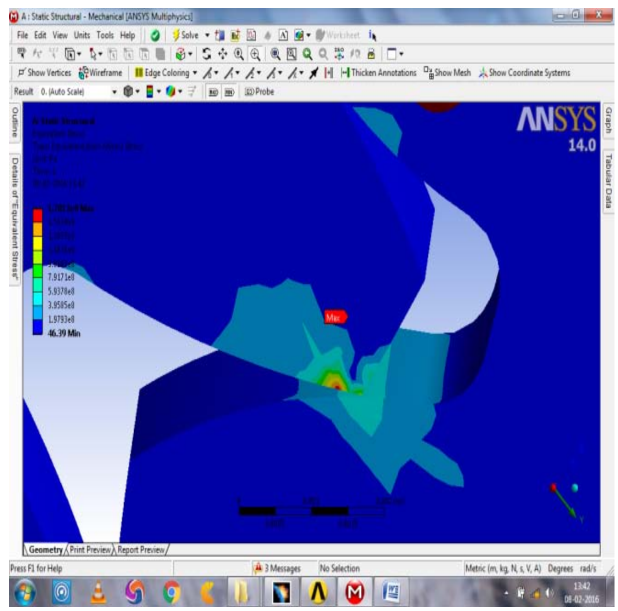

Fig 7: Maximum Contact Stress on a gear pair

The maximum contact stress induced are about 1781.1 N/mm^{2}. These stresses are causing pitting failure. As impact loads are added in dynamic condition and forces increases. Red spot shows the maximum stress point on pitch circle.

8. Experimental Set up:

Photoelasticity is an experimental method to determine the stress distribution in a material. Analytical methods of stress determination, photoelasticity gives a fairly accurate picture of stress distribution even around abrupt discontinuities in a material. The method serves as an important tool for determining the critical stress points in a material and is often used for determining the stress concentration factors in irregular geometries.

Gear Casting:-

[1] Firstly prepared the mould by using the acrylic sheet. The dimension of the mould is 16*16.

[2] The sheet dimensions are 12*12. And thickness is 8mm

[3] The volume of sheet is 150c.c. For every 100 c.c. of araldite 10 c.c. of hardener is to be mixed.

[4] I was taken 150ml araldite and 15 ml of hardener and mixed with each other.

[5] The mixture should be stirred in one direction continuously for 15 minutes till it is transparent.

[6] The mixture is ready to pouring in the mould for preparation of the sheet.

[7] This is the gear model after the machining. This is use for the experimentation.

The life and performance of gear teeth are directly related to the ability of the teeth to withstand contact stresses. Contact stresses may produce pitting within the contact area and eventually lead to tooth failure.

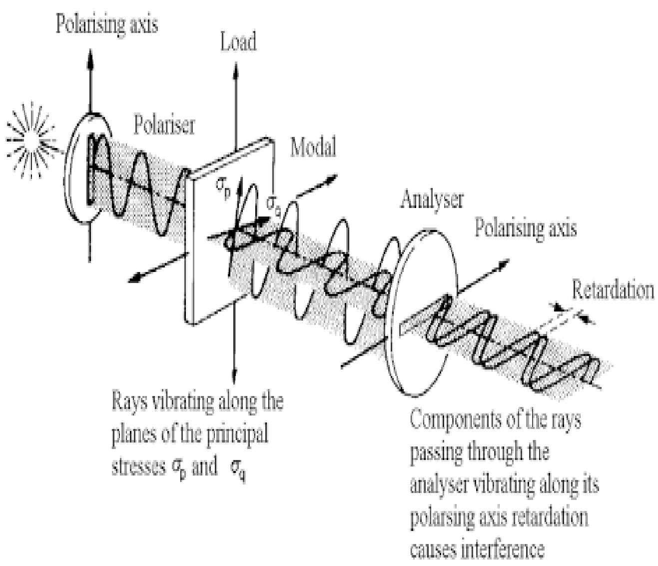

Fig 8: Showing different elements of polaroscope

In photo elastic method, a connection between optical and mechanical characteristics is given via the stress optic low [1]:

$$\sigma 1 - \sigma 2 = (N*f*\sigma /h)$$

Where; σ1 and σ2 are the principal stresses at the point, N is a fringe number, h is a thickness of the photo elastic model (mm), and fσ is the material fringe value.

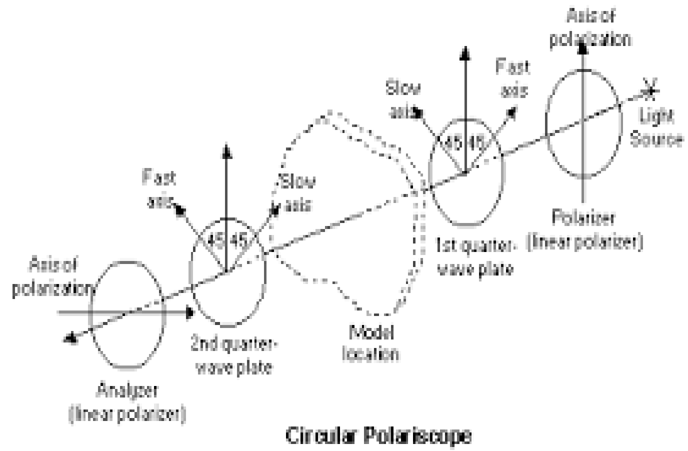

Fig 9: Internal arrangement of photo elastic set up

Specifications of polariscope:-

1. Representation of mechanical distribution of stress in photoelastic experiments.
2. 2 plane polarization filters as polarizer and analyzer.
3. 2 quarter wave filters to generate circular polarized light
4. All filters with 360° angle scale and marking of the main optical axis
5. White light generated using a fluorescent tube and two incandescent lamps
6. Monochromatic light (colour yellow) generated using a sodium vapour lamp
7. Filters roller bearing mounted and rotating
8. Frame cross-arms height-adjustable
9. Generation of compression or tension forces by means of a threaded spindle

Actual set up:

The two gears are arranged on a frame with two central strips with holes holding them. These gears are fitted on shafts. The pinion is fixed by grab screw and the gear is mounted on bearing for free moment of gear. These shafts are welded to the plate. plate is attached to the central strips with holes. Then the weight is arranged on gear by help of a tapping bolts arrangement.

Fig 10: Actual photo elastic set up

$$f\sigma = 8 \times p\pi \times D \times N = 8 \times 19.62\pi \times 30 \times 0.355 = 4.69$$

$$\sigma 1 - \sigma 2 = (N^* f\sigma / h)$$

$$\sigma 1 - \sigma 2 = (1.29^* 4.69 / 8)$$

$$\sigma 1 - \sigma 2 = 0.76 \text{ N/mm2}$$

$$\sigma 2 = 0 \quad \sigma 1 = 0.76 \text{ N/mm2}$$

p = prototype and m = model

σm = 9.26 MPa

Conversion factor for without considering any load between the two gears

$$\sigma p = 13.26x3.68x9.26$$

$$\sigma p = 451.6\ N/mm2$$

Conversion factor for considering load of 300 N between the two gears

$$\sigma p = 13.26x9.263x14.5$$

$$\sigma p = 1775.6\ N/mm2$$

8. RESULT & CONCLUSION

Sr. No.	Von-misses Stresses recorded by FEA	Stresses recorded by Experimentation
1	1781.1 N/mm^2	1775.1 N/mm^2

Table 3. Comparison table between FEA & Experimental Results

The contacts stresses of the spur gear train are calculated by analytical method for module 2mm are 451.6MPa. The gear material is EN24 having Fatigue strengths is 460 MPa. If the contact stresses are higher than the fatigue strengths of the gear material wear is take place at the time of transmitting the power between the gears. The maximum contact stress in maximum load of 300 N stress acting are 1775.1N/mm^2.

References

From Journal

[1] Bharat Gupta, June-2012, "Contact stress analysis of spur gear", International Journal of Engineering and Research (IJER), Vol.1, Issue 4.

[2] M.Raja Roy, 2014, "Contact pressure analysis of spur gear using FEA", International Journal of Advanced Engineering Applications (IJAEA), Vol.7 Issue 3, pp.27-41.

[3] Ali Raad Hassan, 2009-2010, "Contact stress analysis of spur gear tooth pair," World Academy of Science, Engineering and Technology, Vol. 3, pp. 595-600.

[4] Yadav S.H., 2013, "Failure investigation of planetary gear train due to pitting", International Journal of Mechanical Engineering and Robotics (IJMER), Vol.1, Issue 2, pp.86-92.

[5] Konstandinos G. Raptis, Theodore N. Costopoulos, Georgios A. Papadopoulos and Andonios D. Tsolakis, 2010, "Rating of Spur Gear Strength Using Photoelasticity and The Finite Element Method", American J. of Engineering and Applied Sciences, Vol.3, Issue 1, pp.222- 231.

[6] Ali Kamil Jebur, April 2011, "Numerical and Experimental Dynamic Contact of Rotating Spur Gear", Modern Applied Science, Vol. 5, Issue 2, pp.254 -263.

[7] Abhijit Mahadev Sankpa and M. M. Mirza, 2014, "Contact Stress Analysis of Spur Gear by Photoelastic Technique and Finite Element Analysis", International Journal on Theoretical and Applied Research in Mechanical Engineering (IJTARME), Vol. 3, Issue- 2, pp. 50-54. 59

[8] C.M. Meenakshi, S. Balaguru and N. Senthil Kumar, 2013 "Spur Gear Model with CAD and Stress Analysis with FEM", Middle-East Journal of Scientific Research Vol.18 Issue12, pp.1832-1836.

[9] Vivek KaraveerȦ, Ashish MogrekarȦ and T. Preman Reynold Joseph, December 2013, "Modeling and Finite Element Analysis of Spur Gear", International Journal of Current Engineering and Technology, Vol.3, Issue 5, pp.2104-2107.

[10] Rubén D. Chacón, Luis J. Andueza,Miguel A. Díaz And José A. Alvarado "Analysis of stress due to contact between spur gears", Advances in Computational Intelligence. Vol.8, Issue 2, pp.216-220.

[11] C.M. Meenakshi, Akash Kumar, Apoorva Priyadarshi, Digant Kumar Dash and Hare Krishna,2012,"Analysis of Spur Gear Using Finite Element Analysis", Middle- East Journal of Scientific Research, Vol.12, Issue 12, pp. 1672-1674.

[12] R.S.Khurmi, J.K. Gupta, 2005, *Machine Design*, Ist Edition, Eurasia publishing house (pvt.) ltd, Ram nagar, new delhi-110 055, pp.1043-1044.

[13] Gitin M. Maitra, *Handbook of gear design*, II Edition, Tata McGraw-hill Companies, 2.105-2.107.

[14] J. Shigley, 2006, *Mechanical Engineering Design*, Eighth Edition, McGraw‒Hill, USA.

[15] Faydor L. Litvin and Alfonso Fuentes, 2014, *Gear Geometry and Applied Theory*, Second Edition, Cambridge University Press, New York, USA.

[16] V.B.Bhandari "*Design of Machine Element*", Third Edition, a McGraw Hill Education Private Limited.

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