Spur Gear Contact Stress Analysis and Stress Reduction by Experiment Method

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Abstract—The gears are used for a wide range of industrial applications. They have varied application starting from textile looms to aviation industries, automobile gear box and machine tool application to transmitting the power. Their function is to convert input provided by prime mover into an output with lower speed and corresponding higher torque. Spur gears are used to transmit the power up to velocity ratio is ten. This phase they induce high stress at the point of contact. A pair of teeth in action is generally subjected to contact stresses causing fatigue failure of gear tooth.

The main purpose of this study is to reduce the contact stress of gear by increasing the module of gear. One Spur gear train is selected for analysis. The Contact stress of existing gear train is calculated and compared with fatigue strengths of gear material. If this stress on gears are higher than fatigue strengths means gears are failed due to fatigue. To reduce the contact stress by increasing the module of gear. The contact stress are calculated by Hertz's Equation and Strain gauge is used for the experimental investigation of the stress field.

Keywords— Spur gear train, Pitting, Hertz' contact stress, Module, Molding and casting, Strain gauge, Gear ratio.

INTRODUCTION

The gears are widely used for transmitting the power from one shaft to another shaft in automobile transmission system & machine tools application. The power is transmitting from prime mover to machine with increasing or decreasing the speed. The gears are mostly using for transmitting the power because gears are positive drive, compact, reliable & transmit high power with higher efficiency. Gear is most critical components in a mechanical power transmission system. The gears are mostly classified into four types spur, helical, bevel & worm gear. Spur gears are simple types of gear. The designing of spur gear is simple and manufacturing cost is low. The spur gears are used for transmitting the power between two parallel shafts because teeth of gears are parallel to axis of rotation of gear. The power is transmitting by successfully meshing of teeth of pinion with teeth of gear.

The tooth of driving pinion exerted a force on the tooth of driven gear and power is to be tracemitted between driving & driven shaft. This force is always acts along the pressure line at pitch point called as a normal force or resultant force. This normal force is resolve in tangential & radial component of gear in horizontal & vertical plane respectively. The torque & power of spur gear train is calculating by using tangential component of force. The contact stress and bending stress are inducing on the gear due to the tangential load acts on the gear. If contact stress on the gear is higher than the wear strengths of the gear material gear failure is take place called as wear or pitting failure of the gear.

Wear is progressive removal of metal from the surface. The tooth is thins down and gets weakened. The main causes of wear are misalignment in the shaft, wrong viscosity oil selection & contact stress exceeding the surface fatigue strength of the material. Pitting is a surface fatigue failure of the gear tooth. Material in the fatigue region gets removed in the form of pit. The stress concentration is increase & crack is developed over the tooth surface. The size of crack is increase due to cyclic load acts on the gear. The size of crack is very high up to gear tooth is insufficient to absorb the load acts on it & finally it get beak. That type of failure is called as wear failure. The life of the gear drive is reducing due to the wear failure. To increase life of the gear analysis is very important against the wear failure. The wear failure in the gear is take place due to the contact stress. The contact stress of the gear is reducing up to the limiting value by increasing the module of the gear. The contact stress of the gear is calculated analytically by using hertz's contact stress theory& Experimental method by using Strain gauge. The results obtain by all these methods are comparing and find the deviation in between them.

The one gear pair is selected for the analysis that can be frequently failed at the time of working. The contact stress of gear pair are calculating by analytical method for finding the causes of failure. This stress is compared with wear strength of the gear material. If contact stresses are higher than the wear strength means gear is failed due to the wear failure. To reduce the wear failure of the gear contact stress of gear are reducing by increase the module of gear. The gears are redesign for new value of module selected form the slandered series of module and again calculate the contact stress of gear and compared it with wear strengths of gear material. The same procedure is repeated up to the value of contact stress is less than the limiting value.The contact stress of the gear is also calculating experimental method by using strain gauge. The gears are manufacture by Molding & Casting. The contact stress of the gear is calculated by strain gauge. The contact stresses are higher than the wear strength of the gear material means pitting failure take place in the gear. The contact stresses of the gear are reducing up to the limiting value by increasing the module of gear.

LITERATURE SURVEY

Bharat Gupta [1] say's the gear tooth failure take place if contact stresses in the gear are higher than the wear strength of the gear. For research purpose selecting one spur gear train for contact stress analysis. The contact stress can calculate by analytical method using hertz's contact stress theory for different value of module. The contact stresses can also calculated by FEA method. The model of gear train is formed in the Pro-E software and imported in the Anasys for calculates the contact stresses. The result found by two methods are compared and concluded that difference is within reasonable limit. He is observing the result and 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 [2] in this project work done the analysis of contact stresses induce on the spur gear train for different value of module. For research purpose one spur gear train is selected from lathe gear box to calculating the contact stresses. The contact stresses are calculating by analytical method using hertz's contact stress, FEA method by using Solidwork&Ansys FEA software. In this research paper developed one VISUAL BASIC program for calculate the contact stresses for different parameter like module, power & speed etc. This is simple method to calculating contact stresses for different iteration. The model of mating spur gears is formed in Solidwork and this model is imported Ansys Workbench for calculating contact pressure for different module of spur gear. The result obtained by all this methods are compared and concluded that difference is within permissible limit. The Last conclusion of this paper is if module of the gear is increasing the contact pressure is decreasing.

Ali Raad Hassan [3] has been selected one spur gear train for analysis. The contact stresses induced on the gear are calculating for different contact position when gear is in rotating position. The contact stresses are calculated each 3° rotation of pinion from first location of contact at 0° to last location 30° total 10 such cases are produce. Each case was represented a sequence position of contact between these two teeth. The contact stresses for all this cases are calculated by developing one computer base program in QBASIC language based on analytical method using hertz's contact stress theory. The result can express by plotting the graph of selected cases Vs max. contact stresses. The graph gives results for the profiles of these teeth in each position and location of contact during rotation. Finite element models were made for these cases and stress analysis was done in Ansys -workbench. The finite element analysis results were compared with theoretical calculations and concluded the difference is within reasonable limit. 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] say's gear is important parameter of the power transmission system. If the contact stress in the gear is higher than the surface endurance limit of the gear pitting failure is take place. To reduce this failure contact stress should be less than limiting value. To reduce the contact stress of the gear module of the gear is increasing. 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 model of the gear train is formed in the CAD software & import in the ansys for calculates the contact stress. That stress is compare with surface endurance limit of the gear. He can found that contact stress is higher than surface endurance limit of the gear. To reduce the contact stress of the gear module of the gear is increase and redesign the gear. The contact stresses are reducing up to the lower than surface endurance limit of the gear material. He was increase the life of planetary gear train by reducing the contact stress up to the limiting value of the stress.

Konstandinos G. Raptis [5] was calculating contact stresses of gear by experimental method using photoelasticty. 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. The same calculation is done by FEA method. The modeling of specimen is done in CAD software and imported in Ansys for calculating the contact stresses. The result of both methods are compared and found satisfactorily within permissible limit.

Ali KamilJebur[6] in this paper the maximum contact stresses of spur gear are calculated for different position. For research purpose selecting three spur gear trains having different number of teeth for analysis. The model of spur gear is formed in CAD software & imported in the Anasys for calculate the contact stresses for different position. The 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 gages in the tooth of the gear made form polyimide materials. The result of both methods are compared and concluded that difference is within reasonable limit.

HERTZ CONTACT STRESS (INVOLUTE GEAR TOOTH CONTACT STRESS ANALYSIS)

The pitting is the main cause of the failure of gear tooth. This is also called as wear failure. This is the type of surface fatigue failure due to many repetitive contact stresses occurring in the gear tooth surface at the time of power trancemission.If pair of teeth of gears is in contact subjected to cyclic type of loading the contact stresses are induce on the gear tooth surface are higher than fatigue strength of the gear the tooth get brake.

The method of calculating gear contact stress by Hertz's equation originally derived for contact between two cylinders by using Hertz's contact stress theory. Contact stresses between cylinders are shown in figure 1 and figure 2.

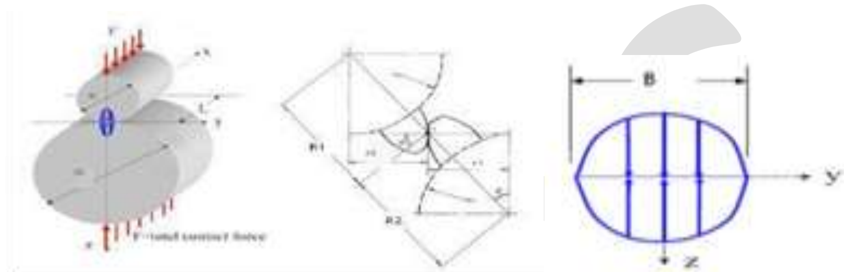


Fig. 1: Cylinders in contact under compression [4]

Fig .2: Elliptical stress distribution across the width [4].

In the machine design problem if two curved surface are subjected to cyclic type of loading during first half cycle they are subjected to compressive stresses and area of contact is increases remaining half cycle they are subjected to tensile stresses and area of contact is decreases. The gear has more interest in curved surfaces of cylindrical in shape because they are similar to the gear surfaces are in contact.

In Fig.4.1 two gear teeth are shown in mating condition at the pitch point subjected to contact stresses. The area of contact under compressive load is a rectangular in shape having width B and length L. The stress distribution pattern is elliptical in shape across the face width of the gear tooth is shown in figure 3.

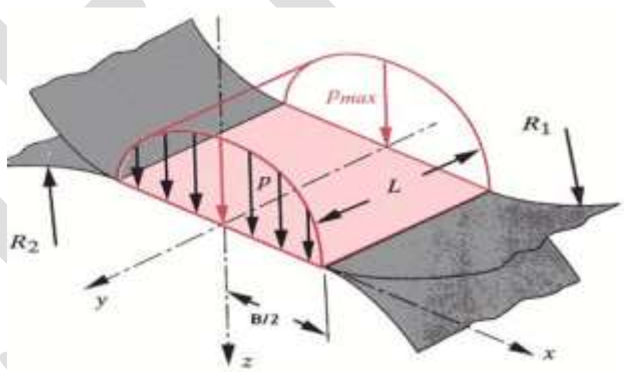


Figure.3: Ellipsoidal–prism pressure distribution [4].

Ellipsoidal–prism pressure distribution Value is given by:

$$P_{cmax} = \frac{4*F}{\pi*B*L} \quad (1) \qquad B = \sqrt{\frac{8*F}{\pi*L} * \frac{\frac{1-v1*v1}{E1}+\frac{1-v1*v1}{E2}}{\frac{1}{D1}+\frac{1}{D2}}} \quad (2)$$

Where,

F= Applied Force.V1&V2 = Poisons ratio of cylinder material.

D1&D2= Diameter of cylinders.E1&E2= Modules of elasticity of cylinder material.

By putting the values of B from Eq.1 and assuming a value of poison's ratio is 0.3 in Eq. 2, and by replacing diameters by respective radii,

$$p_{cmax.} = \sqrt{0.35 * \frac{F}{L} * \frac{\frac{1}{R1}+\frac{1}{R2}}{\frac{1}{E1}+\frac{1}{E2}}} \quad (3)$$

The Hertz equations discussed can be used to calculate the contact stresses induced in tooth surfaces of two mating spur gears. The contact stresses of such gears approximately can be taken to be equivalent to the contact stresses of cylinders having the same radii of curvature at the contact point as the load transmitting gears. Radius of curvature changes continuously in case of an involutes curve, and it changes sharply in the vicinity of the base circle.

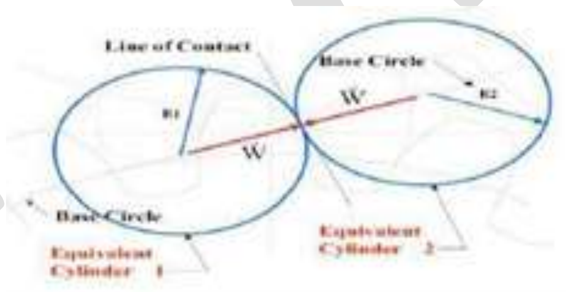


Fig. 4: Equivalent contacting cylinder [4].

Now by putting following Eq. 2

$F = \frac{Ft}{COS\alpha}$, $R_1 = \frac{d1*sin\alpha}{2}$, L=b, $R_2 = \frac{d2*sin\alpha}{2}$

Where,

F_t =tangential force or transmitted load.b = tooth width.

R_1 and R_2=The radii of curvature at pitch point. d_1 and d_2 =The pitch circle diameters of the gears.

Putting,

$E = \frac{2*E1*E2}{E1+E2}$ and $u = \frac{d2}{d1}$

We get,

$$\frac{1}{R1}+\frac{1}{R2}=\frac{2}{sin\alpha}*(\frac{1}{d1}+\frac{1}{d2})=\frac{2}{sin\alpha}*(\frac{1}{d1}+\frac{1}{u*d2})=\frac{2}{d1*sin\alpha}*(\frac{u+1}{u})$$

Inserting these values in Eq. 6.3we get the expression for the maximum contact pressure at the pitch point is:

$$p_{p.}=\sqrt{0.35*\frac{Ft}{COS\alpha}*\frac{1}{b}*\frac{2}{sin\alpha}*\frac{1}{d1}*\frac{u+1}{u}*\frac{E}{2}} \quad (5)$$

$$p_p=\sqrt{0.35*\frac{Ft}{b}*\frac{E}{d1}*\frac{u+1}{u}*\frac{1}{sin\alpha*cos\alpha}}$$

Now by considering service pressure angle, αw

$$p_p=\sqrt{0.35*\frac{Ft}{b}*\frac{E}{d1}*\frac{u+1}{u}*\frac{1}{cos\alpha*cos\alpha*tan\alpha w}}$$

To simplify calculations, is written in the form

$$p_p=\sqrt{\frac{Ft}{b*d1}*\frac{u+1}{u}}*y_m*y_p \quad (6)$$

Where,

Y_m= The material coefficient.$=\sqrt{0.35*\frac{2E1E2}{E1+E2}}$ Y_p = The pitch point coefficient.$=\sqrt{\frac{1}{cos\alpha*cos\alpha*tan\alpha w}}$

ANALYTICAL CONTACT STRESS ANALYSIS OF SPUR GEAR (Hertz's contact stresses):

Sample Calculation for Module - 02

1. Nominal torque on pinion shaft (T) = 9550* P/n_1 = 9550*(2000/250) = 76.39 Nm

2. Tangential Force (Ft) = 2000 * T /d= 2000*76.39 / 44 = 3472.27 N

Input Parameters:

Table 1: Input parameters for Contact stress calculation.

Sr. No.	Input Parameter	Symbol	Value
1	Module	m	02
2	Nominal input power (Wt.)	P	2000
3	Gear Ratio	u	2.54
4	Pinion speed (R.P.M.)	n	250
5	No. of teeth on pinion	Z	22
6	Pressure angle	α	20^0
7	Material for pinion	Grey Cast Iron (C.I.)	
8	Material for gear	Grey Cast Iron (C.I.)	

3. Hertzian contact stress (p_p) = $\sqrt{\frac{Ft}{b*d1} * \frac{u+1}{u}} * y_m * y_p$

Where,

Y_m = The material Co-efficient: $= \sqrt{0.35 * \frac{2E1E2}{E1+E2}}$ E_1 = 110000 E_2 = 110000 Y_m = 196.21

Y_p = The Pitch point Co-efficien= $\sqrt{\frac{1}{cos\alpha * cos\alpha * tan\alpha w}}$ $\alpha = 20^0$ Yp = 1.76

D1 = Module (m) * Number of the teeth (Z1) = 2 * 22 = 44 mm

D2 = Module (m) * Number of the teeth (Z2) = 2 * 56 = 112 mm

$u = D_2/D_1$ = 44 / 112= 2.54

Putting all values in above eq^n,

Hertzian contact stress (Pp) = 809.81 MPa

These contact stresses are compared with fatigue strength of the gear material. The maximum contact stress of spur gear train is 809.81 MPa is higher than Fatigue Strength of Grey C.I.630 MPa. The gears are failed due to the wear or pitting failure. To reduce the contact stress of gear up to the limiting value by increases the module of the gear and redesign.

Table 2: Recommended Series of Module (mm).

Preferred (1)	Choice 2 (2)	Choice 3 (3)	Preferred (1)	Choice 2 (2)	Choice 3 (3)
1			8	7	(6.5)
1.25	1.125		10	9	
1.5	1.375		12	11	
2	1.75		16	14	
2.5	2.25	(3.25)	20	18	
3	2.75		25	22	
4	3.5	(3.75)	32	28	
5	4.5		40	36	
6	5.5		50	45	

Take value of module is 3 from above table choice-1.

D1 = Module (m) * Number of the teeth (Z1) = 3 * 22 =66 mm

D2 = Module (m) * Number of the teeth (Z2) = 3 * 56 = 168 mm

$u = D_2/D_1 = 66 / 168 = 2.54$

Y_m = The material Co-efficient=196.21

Yp = The Pitch point Co-efficient=1.11

Hertzian contact stress (Pp) = 417.012 MPa.

These contact stresses are compared with fatigue strength of the gear material. The maximum contact stress of the spur gear train is 417 MPa is less than fatigue strength of Grey C.I. 630 MPa. Hence contact stresses of the gear are reducing up to the limiting value by increases the module of the gear.

MANUFACTURING OF GEAR BY MOLDING & CASTING (Sand Casting)

Casting is a process of forming metallic products by melting the metal in furnace and pouring it into a cavity known as the mold, and allowing it to solidify for some time. When casting is removed from the mold it will be of the same shape as the mold.

Steps to creating a sand casting:

1. Patternmaking
2. Core making
3. Molding
4. Melting & Pouring
5. Cleaning

Fig. 5 (a)   Fig. 5 (a)

Assembly of spur gear train Fig.5 (a) For module 2 And Fig.5 (b) For module 3

EXPERIMENTAL CONTACT STRESS ANALYSIS OF SPUR GEAR

Object:

To determine the maximum contact stress of a spur gear train by using the strain gauge.

Apparatus:

Parallel axis gear-testing machine.

Construction:

The one shafts of the gear pair is coupled to a motor of 0.5 HP Variable speed D.C. motor. The other shaft will rotate in the opposite direction as the gears are meshed together. Thus the two shafts rotate when the motor shaft rotates and the two identical spur gears mesh with each other. Load is applied to the gears through commentator. The idea is to find the strain in the gear teeth for various positions at particular load and speed. The induced strains in a gear tooth are measured with the help of a calibrated strain gauge indicator circuit.

Experimental set up:

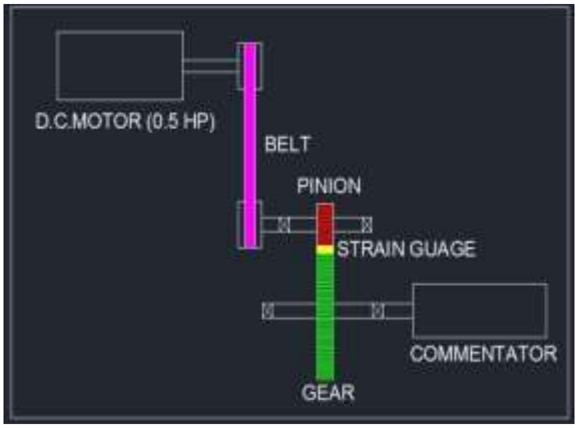


Fig.6 Assembly of experimental setup.

Procedure:

Step no.	Steps
1	Adjust the dimmer stat voltage value to zero.
2	Start the motor.
3	Initial set up will run under no load condition.
4	Bridge circuit balance at zero reading
5	Slowly increase the speed of motor.
6	Note down the no load reading on strain gauge indictor.
7	Increase the speed of motor and set at 250 R.P.M.
8	Increase the load on gear and set at 3472.27 R.P.M.
9	Note down the strain at various position.

Calculate contact stress for module – 02.

1. Observations

   1. Modulus of elasticity of Gear material (Grey Cast Iron) (E): 110000Mpa.

   2. I/P Speed: 250 R.P.M.
   3. Tangential Load :3472.27N

2. Calculation**:**

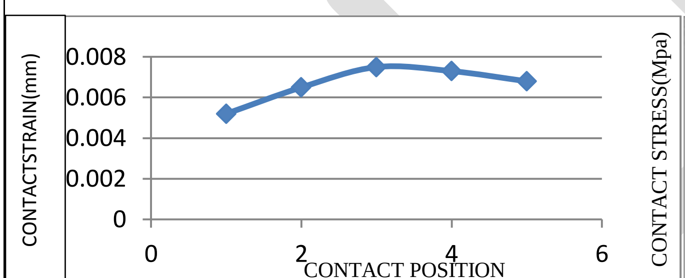
Stress = Modulus of Elasticity * Strain

e = E*€

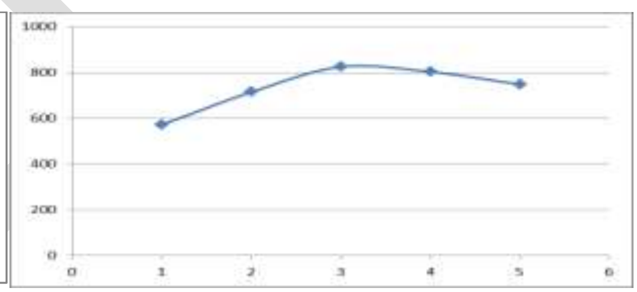
3. Results:

Table 2 Observation table.

Contact position	Strain(€) (mm)	Stress(e) = €*E(MPa)
01	0.0052	572
02	0.0065	715
03	**0.0075**	**825**
04	0.0073	803
05	0.0068	748



Graph 1- StrainVs Contact position.



Graph 2- StressVs Contact position.

Calculate contact stress for module – 03:

1. Observations

   1. Modulus of elasticity of Gear material (Grey Cast Iron) (E): 110000Mpa.

   2. I/P Speed: 250 R.P.M.
   3. Tangential Load :3472.27N

2. Calculation**:**

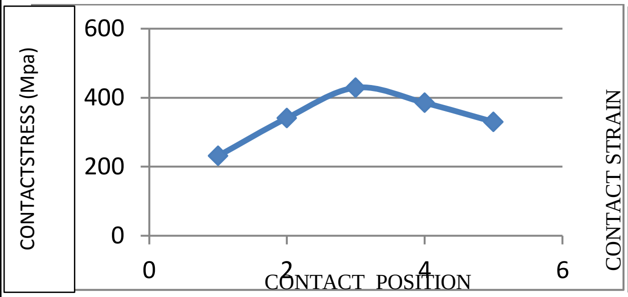
Stress = Modulus of Elasticity * Strain

e = E*€

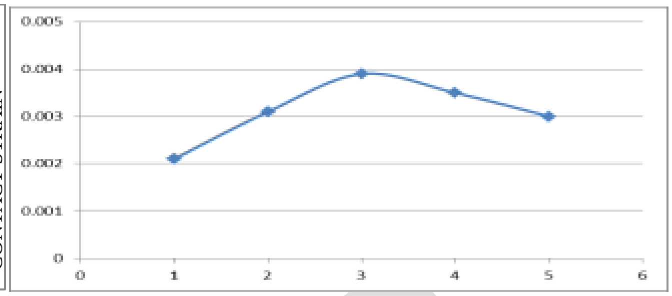
Table 3 Observation table.

Contact position	Strain(€) (mm)	Stress(e)= €*E (MPa)
01	0.0021	231
02	0.0031	341
03	**0.0039**	**429**
04	0.0035	385
05	0.0030	330

3. Results:



Graph 3- Stress Vs Contact position.



Graph 4- StrsainVs Contact position.

RESULT AND DISCUSSION

Contact Stress (N/mm^2)		Wear strength (N/mm^2)
Module - 02	Module - 03	Grey C.I.
809.81	417	630

Table 4 Result of Analytical method.

Contact Stress (N/mm^2)		Wear strength (N/mm^2)
Module - 02	Module - 03	Grey C.I.
825	429	630

Table 5 Result of Experimental method.

1. The contact stress of the spur gear train by analytical method at module - 02 is 809.81 MPa. is reduce up to the 417Mpa by taken higher value of module 03.This value is less than the wear strength of the Grey C.I. 630 Mpa of the gear material.
2. The contact stress of the spur gear train by Experimental method at module - 02 is 825 MPa. is reduce up to the 429Mpa by taken higher value of module 03.This value is less than the wear strength of the Grey C.I. 630 Mpa of the gear material.

Table 6 Comparison of Analytical & Experimental results.

Module	Contact stress (MPa)		Difference (%)
	Analytical method	Experimental Method	
02	809.81	825	1.87
03	417	429	2.87

CONCLUSION

The model of spur gear train is formed by molding and casting calculates the contact stress of gear by experimental method using strain gauge. From result it is found that the contact stresses of the gear are higher than the Fatigue strength of the gear material. From result it is also concluded that contact stress are cause of pitting failure of the gear.

Module is important parameter of gear. To reduce the contact stress of gear the module of gear is increases. If module of the gear is increasing contact stress are decreasing up to the limiting value. Same result is verified by analytical& experimental Method. It found that the deviation between the results of the applied methods in between reasonable limits.

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