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COMPARATIVE FATIGUE LOAD ANALYSIS OF DIFFERENT MATERIAL GEARS BY ANSYS

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Abstract

This paper presents the comparative analysis and optimization of Spur gears by considering the design parameters for different materials. Recently the trend of composite materials is found out in automobile sectors and in marine applications. So, Aluminium silicon carbide is one of the composite selected for manufacturing of spur gear for comparative analysis with general used Steel Alloy and Nylon gears in the application food processing sugarcane crushing machines. Tooth load and fatigue stress is calculated with the help of AGMA design procedure by Lewis equation and Hertz contact stress equation. Fatigue load and stress analysis of the gear is done by analytically and with help of Ansys software package for finding out the Von-mises stress on the tooth of the gears while in meshing.

Index Terms- AGMA procedure, Aluminium silicon carbide (AlSiC), Hertz contact stress equation, Lewis equation.

I. INTRODUCTION

The gear is one of the complex components in mechanical power transmission systems. Gears have large variety of applications in different sectors. It is the main component in a power transmission system. Advances technology comes in engineering have been brought demands for gear teeth, which can operate at ever increasing load capacities and speeds. The gears generally fail when tooth bending stress exceeds the safe endurance limit. Therefore, it is essential to need alternative Gear material. The important considerations while selecting Gears materials have the ability of the Gear material to capable high frictional temperature and less abrasive wear. Cost, manufacturability and weight are also important factors

those are need to be considered during the design parameters.

Calculate the fatigue load and stress conditions for those different materials with help of two theoretical formulas, which deal with these two fatigue failure mechanisms. One is the Hertz equation, which can be used to calculate the contact stresses and other is the Lewis formula, which can be used to calculate the bending stresses. The theoretically obtained results of fatigue load analysis can be verified by testing of gears on Software packages like ANSYS, CATIA V5, and PTC Cero under different load conditions

II. LITERATURE REVIEW

It has proposed to substitute the metallic gear of sugarcane juice machine with plastic gears to reduce the weight and noise. For this purpose they have considered two different types of plastic materials namely Nylon and Polycarbonate and their viability have been checked with their counterpart metallic gear (Cast iron). Based on the static analysis, the best plastic material has been recommended for the purpose. Gears made of Nylon 6 are especially susceptible to failure due to extreme heat accumulation in the single tooth mesh area, which results in damage that consequently shortens gear teeth life and causes transmission error by V. Siva Prasad *et al.* [1].

It has carried out experimental study and suggests modifying gear width to enhance heat transfer rate by Huseyin Imrek *et al.* [2]

Contact stresses are generally considered as deciding factor in finding out dimensions of gear and contact stress analysis of spur gear have been carried out by Shinde *et al.*[3] and they compared theoretically obtained stress values with FEA. Various national and international standards for design of gears have been given in handbook of gear design.

The gear in a gear box fitted an armoured tracked vehicle

for the purpose of power transmission and positioned of heavy mass to the desired angle with high accuracy are subjected to fluctuating loads that are random in nature. Therefore, it is analyzed for random loading and also under constant amplitude loading conditions for fatigue analysis and life of gear has been obtained by finite element package ANSYS by P.Rambabu, B.Jithendra *et al.* [4]

It has carried out analysis of spur gear made from Alloy steel and Grey Cast Iron. They have carried out theoretical analysis of gears by using Hertz equation and finite element analysis using ANSYS 14.5. They found very less error in both the results by Vivek karveet *et al.*[5]

For analysis purpose they have considered one problem and designed gears for static condition. FEM has been carried out using ANSYS. Finally results were compared for both gears. It found that for the cast iron material the stress intensity and displaced shape results are better for actual spur gear compared to optimized spur gear. Design, Modeling and optimization of spur gear using finite element methods has been carried out by Malleswara Rao and Tippa Bhimashankar Rao *et al.* [6]

An approach for predicting elastic properties of composite materials has been presented by Davis *et al.* [7]

They have stated best method for prediction of elastic modulli for composite. He has also used simulation approach for predicting properties. Also it has been designed Spur gear calculated stresses developed at root of teeth for different modules. They have carried out FEM in ANSYS and found good agreement between theoretically calculated values of Bending stress and contact stress with those obtained by using ANSYS. Stress analysis of Spur gear for various modules has been carried out by N.Lenin Rakesh *et al.* [8]

A gear has been tested in the test rig under random loading conditions and also under constant amplitude conditions for fatigue bending, and the life of a gear has been obtained experimentally by D.Hanumanna, S.Narayanan and S.Krishna Murthy *et al* [9] designed and fabricated an electro hydraulic test rig that is capable of generating different types of load pattern by adopting a suitable electronic circuit in a test rig.

Aluminium alloy materials found to the best alternative with its unique capacity of designing the materials to give required properties. In this paper tensile strength experiments have been conducted by varying mass fraction of SiC (5%, 10%, 15%, and 20%) with Aluminium. The maximum tensile strength has been obtained at 15% SiC ratio. Mechanical and Corrosion behaviour of Aluminium Silicon Carbide alloys are also studied by Neelima Devi. C, Mahesh.V, Selvaraj. N *et al* [10]

III. OBJECTIVE

The main aim of the research work is to design the spur gear with using alternating composite material which is better than the other common materials used for gears. And compare analytical and simulation results of Ansys software package in terms of fatigue bending stress values. And show that the composite material gear having lighter in weight, corrosion resistant and nearly same fatigue strength as of the common material.

IV. COMPARATIVE GEAR MATERIALS

Mainly used gear materials are compared with new composite material,

- Steel Alloy (A216-WCB)
- Aluminium Silicon Carbide (AlSiC)
- Nylon

Gears suffer from four common failure mechanisms; breakage, wear, pitting, and scoring. Each mode of failure has its own specific characteristics but the most common failure mode, and the focus of this study, is breakage through fatigue. Tooth breakage is defined as the fracture of substantial part of a tooth. The most common causes for such a failure include overload and cyclic stressing beyond the endurance limit. The bending fatigue breakage occurs in several steps. First, there is crack initiation at a specific high-stress location on the tooth. Following initiation, the crack propagates, sub-critically, through the tooth.

The crack initiation phase is normally much longer than the crack propagation phase. Eventually, the crack reaches a critical length and catastrophic failure occurs. The most common site for fatigue crack growth is at the tooth root fillet, due to the high stress concentration. High cycle fatigue is typically associated with cycles greater than 1000 and at stress amplitudes less than the gross yield stress.

V. FATIGUE BENDING STRESS DESIGN OF WCB A216

As per application of gears we have taken Input Data: $W = 5\text{HP} = 5 \times 0.746 = 3.73 \text{ kW}$; $n_1 = n_2 = 1440 \text{ rpm}$; 20° full depth involutes spur gear.

Solution: $i = n_1 / n_2 = 1$

In order to keep the size of gears small and avoid interference also,

$Z_1 = 18$ is chosen. $Z_2 = i Z_1 = 1 \times 18 = 18$

Now, angular velocity of spur gear pair,

$$\omega = \frac{2\pi n}{60} = \frac{2\pi \times 1440}{60 \times 1000} = 150.79 \text{ rad/sec} \dots\dots(1)$$

Torque transmitted by pair,

$$T = \frac{1000 \times w}{\omega} \dots\dots(2)$$

$$= \frac{1000 \times 3.73}{150.79} = 24.7353 \text{ Nm} = 24735.3 \text{ Nmm}$$

AGMA equation for tooth bending stress,

$$\sigma = \frac{F_t}{bmJ} \times K_v \times K_m \times K_o \dots\dots(3)$$

Where $b = mZ$

$$\sigma = \frac{2T}{bzm^2 J} \times K_v \times K_m \times K_o \dots\dots(4)$$

Face width $b = 10$ to $13 m$.

$b = 10 m$ is assumed for the first trial.

$J = 0.32404$ for pinion $Z_1 = 18$ mating with gear $Z_2 = 18$

These values are obtained from the table.

$$K_v = \left(\frac{78 + (200v)^2}{78} \right)^{0.5} = 1.15 \dots\dots(5)$$

$K_o = 1.25$ Is taken assuming uniform power source and moderate shock load.

$K_m = 1.3$ Is taken assuming accurate mounting and precision cut gears for face width of about 50mm.

Substituting the value in the equation

$$\sigma_1 = \frac{2T}{bz m^2 J} \times K_v \times K_m \times K_o \dots\dots(6)$$

$$= \frac{(2 \times 24735.3 \times 1.15 \times 1.25 \times 1.3)}{(10m \times 18 \times m^2 \times 0.32404)} = \frac{1584.992}{m^3}$$

$$\sigma_1 = \frac{1584.992}{m^3} \dots\dots(7)$$

Now endurance stress of pair,

$$\sigma_e = \sigma_{e'} \times k_L \times k_v \times k_s \times k_R \times k_T \times k_f \times k_m \dots\dots(8)$$

For pinion $\sigma_{e'} = 0.5\sigma_{ut} = 0.5 \times 543.5 = 271.75 Mpa$

$k_L = 1$ for bending, $k_v = 1$ assumed expecting m to be <5mm; $k_s = 0.73$

$\sigma_{ut} = 543.5 Mpa$, $k_r = 0.897$ for 90% reliability.

$k_T = 1$ assumed based on operating temperature <120oC

$k_f = 1$ and $k_m = 1.33$ for $\sigma_{ut} = 543.5 Mpa$

$$\sigma_e = \sigma_{e'} \times k_L \times k_v \times k_s \times k_R \times k_T \times k_f \times k_m$$

$$\sigma_e = 271.75 \times 1 \times 1 \times 0.73 \times 0.897 \times 1 \times 1 \times 1.33 = 236.66 Mpa$$

Factor of safety on bending of 1.5 assumed,

$$[\sigma] = \frac{\sigma_e}{s} = \frac{236.66}{1.5} = 157.77 Mpa \dots\dots(9) \quad (2)$$

Now From tooth bending fatigue considerations,

$$\sigma_1 = \frac{1584.992}{m^3} \leq \sigma \dots\dots(10)$$

$m = 2.15$ = we have to take next module = 3mm. So gear specification becomes,

TABLE I
GEOMETRICAL DIMENSIONS OF SPUR GEAR FOR M=3

Gear	Z	M	B=10m	D	V=Wrv	Material
Pinion	18	3	30	54	4.07 M/S	Wcb A216

Tangential load on gear tooth becomes,

$$F_t = \frac{T}{r} = \frac{24735.3}{27} = 916.12 \dots\dots(11)$$

So tooth checked by surface durability consideration, Contact stress equation from AGMA,

$$\sigma_H = C_p \sqrt{\frac{F_t \times K_v \times K_o \times K_m}{bdI}} \dots\dots(12)$$

$C_p = 191 Mpa$ for steel Vs steel,

$$I = \frac{\sin \theta \times \cos \theta}{2} \times \frac{i}{i+1} \dots \text{where } i=1$$

$$I = \frac{\sin(20) \times \cos(20)}{2} \times \frac{1}{1+1} = 0.0803$$

$$K_v = \left(\frac{78 + (200v)^2}{78} \right)^{0.5} = 1.15$$

$$\sigma_H = 191 \sqrt{\frac{916.12 \times 1.168 \times 1.25 \times 1.33}{30 \times 54 \times 0.0803}} = 706.31 Mpa \dots\dots(13)$$

Calculating surface fatigue strength,

$$\sigma_{sf} = \sigma_{sf'} \times K_L \times K_R \times K_T \dots\dots(14)$$

$$\sigma_{sf'} = (2.8 \times Bhn) - 69 = 390.21 Mpa$$

Assuming a factor of safety $s = 1.1$

$$[\sigma_H] = \frac{\sigma_{sf}}{s} = \frac{351.18}{1.1} = 354.73 Mpa$$

$$\sigma_H = 706.31 Mpa$$

so

$$\sigma_H = 706.31 Mpa > [\sigma_H] = 354.73 Mpa$$

The design is not safe and surface fatigue failure will occur. Increase the surface hardness of the material to 210 Bhn and also increase the module a next $m=4$, $b=10 \times 4=40$,

So again check for new terminology of spur gear pair,

TABLE II
GEOMETRICAL DIMENSIONS OF SPUR GEAR FOR M=4

Gear	Z	m	b=10m	D	V=wrv	Material
Pinion	18	4	40	72	5.42 m/s	WCB A216

Tangential load on gear tooth becomes,

$$F_t = \frac{T}{r} = \frac{24735.3}{36} = 687.09 N \dots\dots(15)$$

So tooth checked by surface durability consideration, Contact stress equation from AGMA,

$$\sigma_H = C_p \times \sqrt{\frac{F_t \times K_v \times K_o \times K_m}{bdI}} \dots\dots(16)$$

$C_p = 191 Mpa$ for steel Vs steel,

$$I = \frac{\sin \theta \times \cos \theta}{2} \times \frac{i}{i+1} \dots \text{where } i=1$$

$$I = \frac{\sin(20) \times \cos(20)}{2} \times \frac{1}{1+1} = 0.0803 \dots (17)$$

$$K_v = \left(\frac{78 + (200v)^2}{78} \right)^{0.5} = 1.15$$

$$K_m = 1.168 \quad K_v = 1.25 \quad K_o = 1.33$$

$$\sigma_H = 191 \times \sqrt{\frac{916.12 \times 1.168 \times 1.25 \times 1.33}{40 \times 72 \times 0.0803}} = 458.76 \text{ Mpa}$$

.....(18)

Calculating surface fatigue strength,

$$\sigma_{sf} = \sigma_{sf'} \times K_L \times K_R \times K_T \dots (19)$$

$$\sigma_{sf'} = (2.8 \times Bhn) - 69 = 519 \text{ Mpa}$$

$K_L=1$ for 10^7 cycle $K_R=1$ $K_T=1$
So,

$$\sigma_{sf} = \sigma_{sf'} \times K_L \times K_R \times K_T = 519 * 1 * 1 * 1 = 519 \text{ Mpa}$$

Assuming a factor of safety $s = 1.1$

$$[\sigma_H] = \frac{\sigma_{sf}}{s} = \frac{519}{1.1} = 471 \text{ Mpa}$$

$$\sigma_H = 471 \text{ Mpa}$$

so

$$\sigma_H = 458.76 \text{ Mpa} < \sigma_H = 471 \text{ Mpa}$$

The design is safe and surface fatigue failure will not occur.
So new terminology of spur gear pair.

A. Loading Details

Fatigue load is calculated by the given data and this load is acted on the contact point meshing test gear. In the paper it is given that there is only one teeth contact at a time.

Gear box input Torque (t) = 24735.3 N-mm,

Gear Ratio = 1,

r = Radius of Master gear = 36mm,

Tangential load on test gear (W) = T/R

W = 687.09 N.

B. Fatigue Stress Calculation of Details

Use The bending stress can be calculated by the Lewis formula analytically.

Surface fatigue bending stress= 471 MPa.

The maximum Surface fatigue bending stress induced in the test gear is 471 MPa.

TABLE III
ANALYTICAL RESULTS FOR DIFFERENT MATERIAL GEAR

Material	Hertz Contact Stress Mpa	Hertz stress by Fatigue consideration Mpa
WCB A216	458.62	471
AlSiC	389.53	407.27
Nylon	82.27	153.63

VI. FINITE ELEMENT ANALYSIS

Finite element modeling is described as the representation of the geometric model in terms of finite number of element and Nodes, which are the building blocks of the numerical representation of the model. It is actually a numerical method employed for the solution of structures or a complex region. Defining a continuum Solutions obtained by this method are rarely exact. In FEM, complex structure or continuum is divided into finite number of small regions called as elements. The material properties and governing equations are considered over these elements and the field quantity is expresses as unknown values at corners of these elements called as the nodes. The meshing gears and were manufactured using the same materials respectively.

TABLE IV
MATERIAL PROPERTIES OF GEARS

Material	WCB A216	Alsic composite	Nylon
Young's Modulus (Gpa)	210	188	2.25
Poisson's ratio	0.3	0.3	0.3
Ultimate Tensile strength (Mpa)	543.5	260	82

A. STRESS ANALYSIS BY USING ANSYS

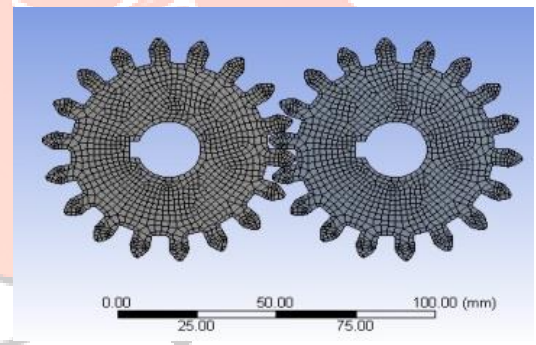


Fig1: Loads and boundary conditions of the gear

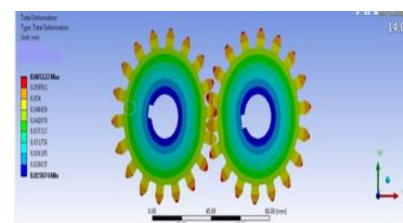


Fig 2: Displacement pattern for wcb a216 gear.

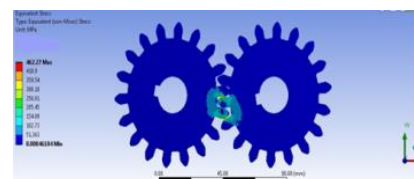


Fig 3: Stress distribution for WCB A216 gear

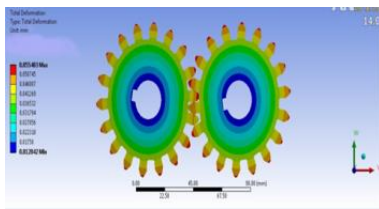


Fig 4: Displacement pattern for alsic gear.

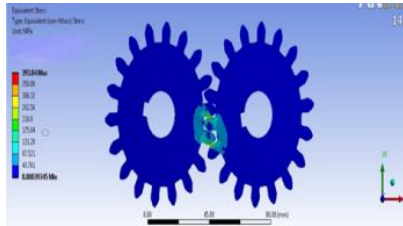


Fig 5: Stress distribution for alsic gear.

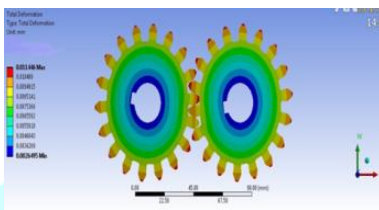


Fig 6: Displacement pattern for nylon gear.

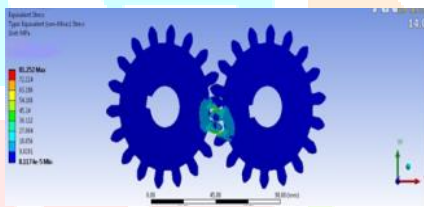


Fig 7: Stress distribution for nylon gear.

B. RESULTS OBTAINED BY ANSYS

From the transmitted load find out the deflections and vonmisse stress values for the steel alloy WCB A216, AISic and Nylon material by Ansys software obtained as following shown in below tables.

TABLE V
FOR WCB A216 MATERIAL GEAR

Tangential load N	Vonmisse stress(N/mm2)	Deflection (mm)
140	92.426	0.01302
280	184.85	0.02604
420	277.35	0.03907
560	369.77	0.05209
700	462.27	0.06512

TABLE VI
FOR ALSIC COMPOSITE MATERIAL GEAR

Tangential load N	Vonmisse stress(N/mm2)	Deflection (mm)
140	78.779	0.0111
280	157.54	0.02219
420	236.31	0.03329
560	315.01	0.04438
700	393.84	0.05548

TABLE VII
FOR NYLON MATERIAL GEAR

Tangential load N	Vonmisse stress(N/mm2)	Deflection (mm)
140	16.278	0.00229
280	32.556	0.00459
420	48.834	0.00688
560	65.112	0.00917
700	81.252	0.01145

C.GRAPHICAL REPRESENTATION OF DISPLACEMENT VS FATIGUE BENDING STRESS

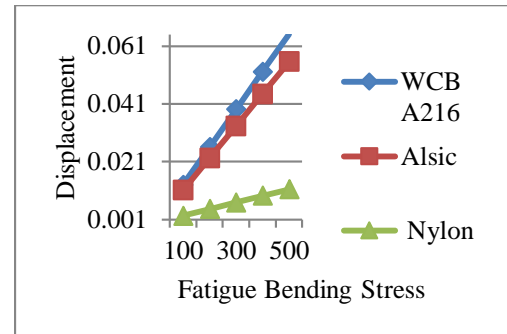


fig 8: Displacement vs Fatigue bending stress.

TABLE VIII
COMPARISON BETWEEN ANALYTICAL AND SIMULATION OF FATIGUE STRESS VALUES

Material	Vonmisse stress(N/mm2)	Analytical stress (N/mm2))
WCB A216	462.27	458.76
AISic	393.84	389.53
Nylon	81.252	82.62

VII. CONCLUSION

From this project work it study out the suitable spur gears design with less weight and less cost, corrosion resistance, frictionless also. So, with the use of alternating composite material Alsic have nearly closer to fatigue strength as compared with steel alloy and Nylon material for sugarcane juice crusher. With less cost, self-lubricates neat and clean hygienic juice.

So from this work Alsic gives maximum 393.81 Mpa and which is nearly closer to fatigue strength as of WCB steel alloy is 462.27 Mpa and more than 81.52 Mpa for Nylon material from Ansys software package and these are nearly same by theoretical method values by AGMA procedure.

Also in weight parameter the composite AISic and Nylon having lighter than steel alloy. From these three materials composite material Alsic makes comparatively same fatigue stress, lighter in weight, corrosion resistant and efficient arrangement for sugarcane cane juice crusher machine.

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