

RESEARCH ARTICLE



ISSN: 2321-7758

FINITE ELEMENT ANALYSIS AND OPTIMIZATION OF THE DESIGN OF STEERING KNUCKLE

VIVEKANANDA REDDI, MYTHRA VARUN A.V

Department of Mechanical Engineering, MVGR College of Engineering, Jawaharlal Nehru Technological University Kakinada, Andhra Pradesh, India.



ABSTRACT

Steering Knuckle is considered as a critical component in automobile suspension system. It is subjected to a repeated multi axial load during its lifecycle and subjected to fatigue failure. Hence, its design is given prime most importance in automotive industry. Presently, SG Iron is widely used. A replacement which results in weight reduction and advanced materials is highly preferable. One such alternative is Metal Matrix Composite which has higher strength to weight ratio and better performance. Al-TiC is such a material with better properties than Aluminium and SG Iron is suitable for this application. Although they have higher strength to weight ratio, they have lesser yield strength than SG Iron. Hence, MMC results in failure due to a lower factor of safety than SG Iron. This project aims to identify the critical region by performing structural and fatigue analysis and to optimize the design for the MMC. Further, the performance before and after the optimization is compared through fatigue, impact, vibration analysis and experimental vibration analysis

Key words: Steering Knuckle, Macpherson Strut System, ANSYS, Finite Element Analysis, Fatigue Analysis, Impact Analysis, Vibration Analysis

©KY PUBLICATIONS

I. INTRODUCTION

In this era of continuous development in the manufacturing sector, design and weight optimization is the prime objective. This objective leads to the research of various metal alloys and composite materials. In automotive industry, the significant increase of the demand for lighter, more fuel efficient vehicles, reduced design-testing iterations, and satisfactory reliability level has promoted this research. The suspension and steering systems are responsible for providing comfort and stability. The performance of both the systems depends on the performance of steering

knuckle which is subjected to millions of varying stresses cycles leading to fatigue failure. Due to its critical functionality, design and weight optimization are required. Therefore, the static and fatigue analysis of the steering knuckle will serve as the foundation for the usage of alternative materials such as Aluminium alloys and composite materials in manufacturing it.

A. Suspension system

Suspension system separates the vehicle mass into sprung mass and unsprung mass. It isolates the body from road shocks and vibrations and keeps the tires in contact with the road.

Therefore, a suspension system must be strong to endure loads during cornering, accelerating, braking, and uneven road surfaces.

B. Macpherson Suspension System

Macpherson suspension system is a most common suspension used in all commercial cars.

C. Components of Macpherson suspension system

1) **Steering Knuckle:** The knuckle is mounted between the upper and lower ball joints on a Short-Long Arm (SLA) suspension, and between the strut and lower ball joint



Fig.1 Macpherson Suspension System

2) **Macpherson Strut:** A strut containing shock absorber and the spring carries also the stub axle on which the wheel is mounted. The wishbone is hinged to the cross member and positions the wheel as well as resists accelerating, braking and side forces.

3) **Tie Rod or Track Rod:** A tie rod is a slender structural unit used as a tie and is capable of carrying tensile & compression loads. It is also referred to as track rod when concerned with automobiles.

4) **Wheel Hub Assembly:** Wheel Hub Assembly is located between the brake rotors and the axle. It is mounted to the holding bracket from the chassis; on the rotor side.

5) **Lower Ball Joint:** The lower control arm and ball joint are retained on Macpherson suspensions. The lower ball joint stabilizes the steering and prevents shimmy.

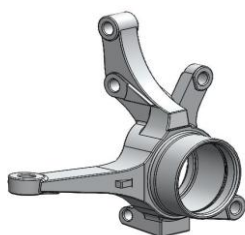


Fig. 2 Steering Knuckle

6) **Brake Calliper:** The brake calliper is the assembly which houses the brake pads and pistons. There are two types of callipers: floating or fixed.

II. Structural Analysis of Knuckle for Multi-Axial Forces

A. Multi-axial force acting on suspension system

Steering Knuckle is subjected to dynamic loads during dynamic conditions like cornering, acceleration and braking. The force acting on the knuckle is as shown:

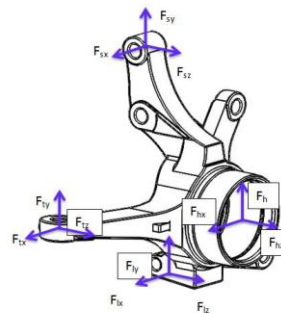


Fig.3 Forces Acting on a Steering Knuckle

B. Cornering

While cornering, a centrifugal force ($C.F=mv^2/R$) acts on the centre of gravity outwards from the centre of turn. This force is directly proportional to the velocity and mass of the vehicle and inversely proportional to the radius of curvature. Lateral load transfer takes place from the inner to outer tires of the vehicle. The weight W is given in eq.1

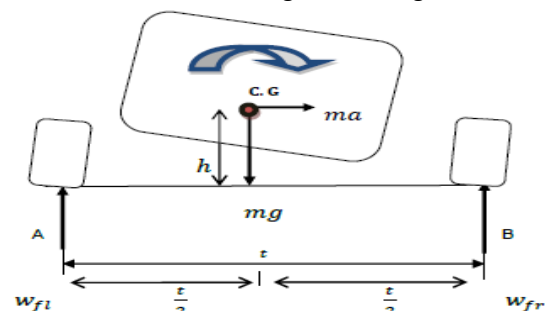


Fig.4 Load Transfer during Cornering

$$mg \times \frac{t}{2} + (CG \times m \times a) = w_{fr} \times t \quad \text{---- (1)}$$

$$w_{fr} = \frac{(1165 \times 9.81 \times \frac{1295}{2}) + (553.9 \times 1165 \times 0.8 \times 9.81)}{1295}$$

$$w_{fr} = 9624.9N$$

The wheel is subjected to vertical, longitudinal and lateral forces which are transmitted to the hub and the wheels. Theoretical calculation of these

forces is cumbersome but found easily using sensors. Hence, for this project, the industrial data is used which are shown in Table I.

TABLE I: FORCES ACTING ON STEERING KNUCKLE DURING 0.8G CORNERING

S.No	Location	FX FY FZ (N)	MX MY MZ (Nm)
1	Strut	-340 -270 -6750	-301920 78070 13430
2	Tie Rod	-120 -830 140	
3	Hub	-40 5260 6340	13764440 -17720 24110
4	LBJ	510 -4160 280	

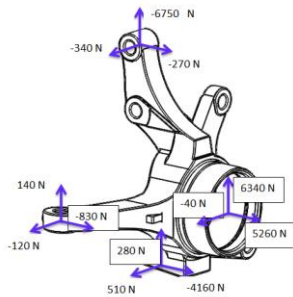


Fig. 5 Cornering Force Directions

C. Acceleration

While accelerating, an inertia force acts on the centre of gravity in a direction opposite to acceleration. This force is directly proportional to the acceleration and mass of the vehicle. A weight transfer takes place from front of the vehicle to the rear of the vehicle.

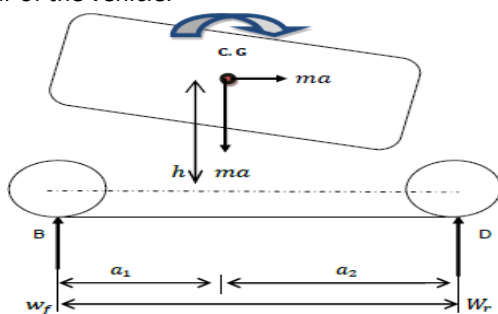


Fig.6 Load Transfer during Acceleration

$$w_f = \frac{1}{2} mg \frac{a_2}{l} - \frac{1}{2} mg \frac{h}{l} \frac{a}{g} \quad \text{----- (2)}$$

$$w_f = \left(\frac{1}{2} \times 1165 \times 9.81 \times \frac{1416}{2360} \right) - \left(\frac{1}{2} \times 1165 \times 9.81 \times \frac{553.9}{2360} \times 0.5 \right)$$

$$w_f = 670.585N$$

This load is transmitted to the body through the hub, knuckle and suspension. As a result, the knuckle experiences a multi-axial load on the attachment points. The forces acting on the right

front knuckle is obtained from the industry and tabulated as Table II.

TABLE II: FORCES ACTING ON THE KNUCKLE DURING 0.5G ACCELERATION

S.No	Location	FX FY FZ	MX MY MZ
1	Strut	70 -1080 3140	287110 135700 -40560
2	Tie Rod	130 1030 90	
3	Hub	-1690 20 2960	3580 0 10
4	LBJ	1490 30 80	

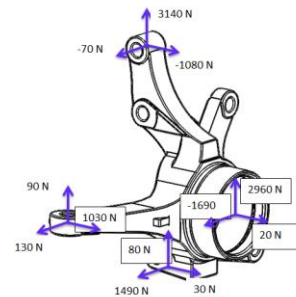


Fig. 7 Acceleration Force Directions

D. Braking

While decelerating, an inertia force acts on the centre of the gravity in a direction opposite to deceleration which resists the deceleration of the vehicle. This force is directly proportional to the mass of the vehicle and deceleration. A weight transfer takes place from rear end of the vehicle to the front end of the vehicle. The force in the front wheels will be greater than rear wheels.

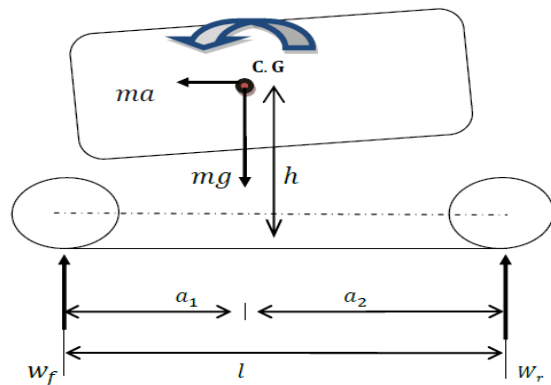


Fig.8 Load Transfer during Braking

$$w_f = \frac{1}{2} mg \frac{a_2}{l} - \frac{1}{2} mg \frac{h}{l} \frac{a}{g}$$

$$w_f = \left(\frac{1}{2} \times 1165 \times 9.81 \times \frac{1416}{2360} \right) - \left(\frac{1}{2} \times 1165 \times 9.81 \times \frac{553.9}{2360} \times 1.1 \right)$$

$$w_f = 4903.885 N$$

As a result of this load transfer from rear to front axle, the wheels are subjected to multi-axial load. This load from wheels is transmitted to the body through the hub, knuckle and suspension. As a result, the knuckle experiences a multi-axial load on the attachment points. The forces acting on the right front knuckle is obtained from the industry and tabulated as Table III.

TABLE III: FORCES ACTING ON STEERING KNUCKLE DURING 0.8G CORNERING

S.No	Location	FX (N)	FY (N)	FZ (N)	MX (Nm)	MY (Nm)	MZ (Nm)
1	Strut	-340	-270	-6750	-301920	78070	13430
2	Tie Rod	-120	-830	140			
3	Hub	-40	5260	6340	13764440	-17720	24110
4	LBJ	510	-4160	280			

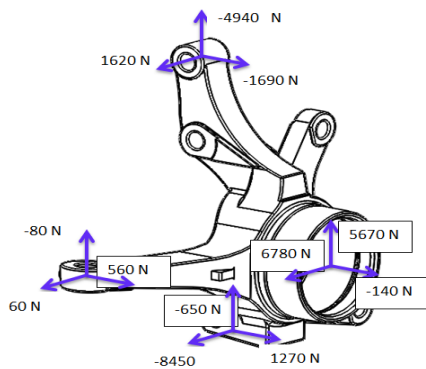


Fig.9 Braking Force Direction

E. Pre-processing for Finite Element Analysis

TABLE IV: MATERIAL PROPERTIES

Materials/ Properties	Density (kgm ⁻³)	Young's Modulus (GPa)	Poisson's Ratio	Yield Strength (MPa)	Ultimate Strength (MPa)
Unreinforced Al Alloy	2700	73	0.33	187	254
Al-10% TiC	2770	79	0.33	201	281
Al-12% TiC	2800	84	0.33	213	289
Al-15% TiC	2850	87	0.33	265	323
SG Iron	7850	170	0.275	360	540



Fig. 10 Part Modelling of Steering Knuckle

Meshing

Element type: Solid187

Nodes: 1,15,042.

Elements: 70,526

Solid 187:

It is a higher order 3-D, 10-node element. It has quadratic displacement behaviour used for modelling irregular meshes. The element is defined by 10 nodes having three degrees of freedom at each node: translations in the nodal x, y, and z directions.

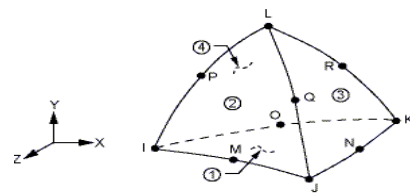


Fig. 11 Boundary Condition

F. Boundary Condition

In actual condition the hub mounting point is attached to the wheels. The forces act on the body through the wheels, hub and upright. But, for our analysis, the hub is assumed to be fixed, the forces act on the other points.

G. Result of Structural Analysis

Through static structural analysis, deformation, stress and factor of safety for the different conditions and different materials are obtained. The region of failure is predicted.

H. Deformation

From the static analysis, the deformations are obtained for various conditions and materials. Among these values, the load due to braking with acceleration has more effect on the knuckle. This is due to the greater load on the tie rod. Among the different materials, the unreinforced alloy experiences maximum deformation and SG Iron experiences minimum deformation. The MMC's have moderate deformation, but comparatively higher than SG Iron.

TABLE V: COMPARISON OF DEFORMATIONS VALUES

Materials/ Properties	Cornering (mm)	Acceleration (mm)	Braking
Unreinforced Al Alloy	2700	73	0.33
Al-10% TiC	2770	79	0.33
Al-12% TiC	2800	84	0.33
Al-15% TiC	2850	87	0.33
SG Iron	7850	170	0.275

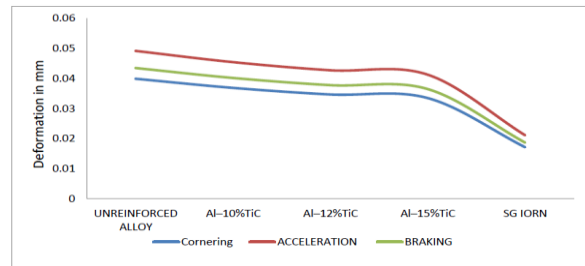


Fig. 15 Deformation in Static Structural Analysis

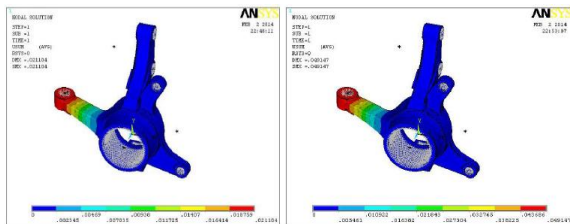


Fig.12 Deformation of SG Iron for static analysis

Fig. 13 Deformation Unreinforced Al alloy for static analysis

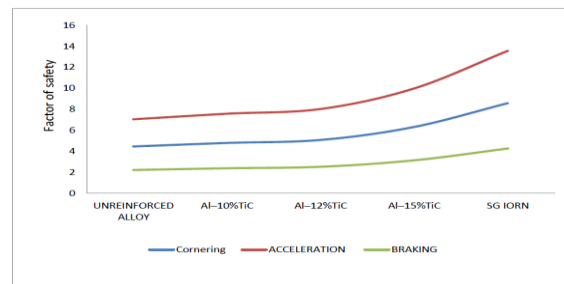


Fig. 16 Factor of Safety

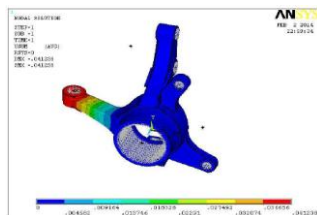


Fig. 14 Deformation of Al-15% TiC for static analysis

K. Prediction of failure region

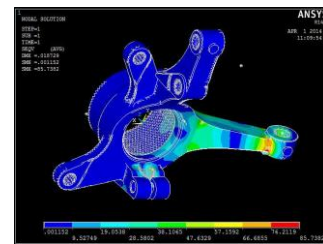


Fig.17 Max Stress produced in Cornering Condition

I. Factor of Safety

Factor of safety compares the working stress with the allowable stress and establishes a ratio.

$$F.o.S = \text{Yield Strength} / \text{Max. Stress developed}$$

From the results, the unreinforced is more liable to failure compared to SG Iron. Hence the factor of safety should be increased by modifying the design.

TABLE VI: COMPARISON OF SAFETY FACTOR

Materials/ Properties	Cornering (mm)	Acceleration (mm)	Braking
Unreinforced Al Alloy	4.458314757	7.047720986	2.21378013
Al-10% TiC	4.792092332	7.575357851	2.37951768
Al-12% TiC	5.078187397	8.027618021	2.521578437
Al-15% TiC	6.317932677	9.987412092	3.137175051
SG Iron	8.582851939	13.56780511	4.261822711

J. Results of static analysis

From the static structural analysis, the region of maximum stress is located as shown in the Fig.15.

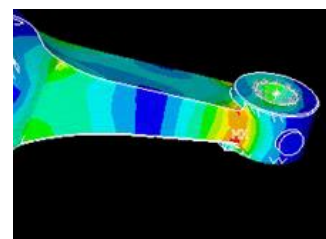


Fig. 18 Region of failure

Static structural analysis predicted the region of failure if SG Iron is replaced with MMC/Metal Alloy. SG Iron has lesser deformation and higher factor of safety compared to the other materials. Al-TiC can be a perfect alternate for SG Iron but due to low yield strength, the factor of safety in steering arm is much lower than the SG Iron. Hence, the safety at this region should analysed by more analysis for incorporating MMC as an alternate material.

III. FATIGUE ANALYSIS

A. Geometry Parameters

Fatigue provides life, damage, and factor of safety information and uses a stress life or strain life approach, with options for handling mean stress and specifying load conditions. Common use for the strain life approach is in notched areas where, although the nominal response is elastic, the local response may become plastic. The three components to a fatigue analysis are:

1. Fatigue material properties,
2. Fatigue loading options,
3. Reviewing Fatigue results.

B. Input

The steering knuckle part modal of the SolidWorks file is imported into the ANSYS Workbench through static structural module which is utilized for the fatigue analysis.

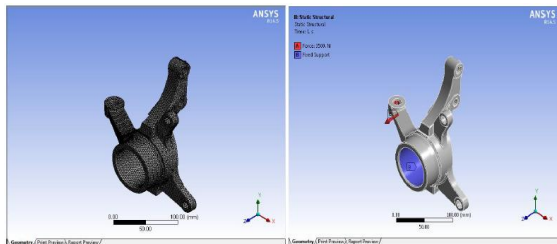


Fig. 19 FEA meshed model of a steering knuckle

Fig. 20 Boundary of Fatigue analysis Conditions.

Mesh Statistics

No of nodes: 115042
 No of elements: 70526
 Element type: Solid 187

C. Boundary Conditions

Fixed support at the hub region where the hub point is used to transmit the steering load from steering arm to ground wheel.

D. S-N Approach

The nominal stress method is widely used in applications where the applied stress is nominally within the elastic range of the material and the number of cycles to failure is large. The nominal stress approach is best suited for processes having high cycle fatigue (life cycle > 105). A strain based methodology must be used for regions where the applied strain has a significant plastic component.

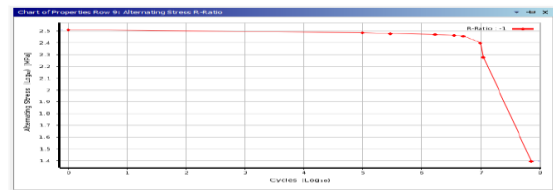


Fig. 21 S-N Curve.

E. Fatigue analysis-Methodologies

Fatigue analysis is performed for extreme steering load case to predict the region of failure in the steering arm. The results such as Life, Damage, Safety Factor, Deformation, and Stress will be plotted using ANSYS Workbench static structural. Input Parameters for Fatigue analysis are:

- Analysis type : Stress life
- Design cycle: 106 cycles
- Mean stress theory: Soderberg theory
- Loading type : Fully Reversed

F. Result of Fatigue Analysis:

Based on fatigue analysis in the Ansys software, the maximum and minimum values of fatigue life, damage, equivalent stress and deformations are:

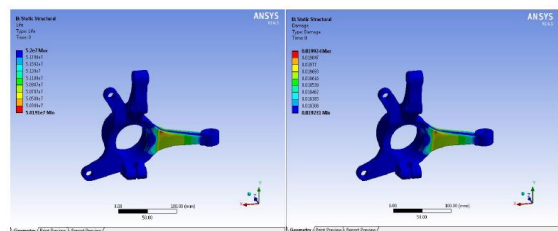


Fig. 22 Fatigue Life Of SG Iron

Fig. 23 Damage of SG Iron

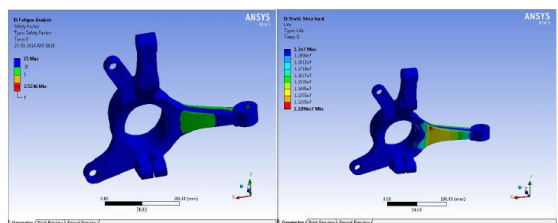


Fig. 24 Safety Factor of SG Iron

Fig. 25 Fatigue life of MMC (Al-15%TiC)

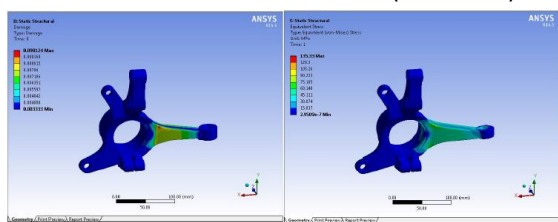


Fig.26 Damage of MMC (Al-15%TiC)

Fig. 27 Safety Factor of MMC (Al-15%TiC)

G. Results of Fatigue Analysis

TABLE VII: FATIGUE RESULTS BEFORE OPTIMIZATION

Materials/ Properties	Life Cycle	Damage	Safety Factor
Unreinforced Al Alloy	3001200	0.33319	1.0399
Al-10% TiC	11045000	0.09054	1.5661
Al-12% TiC	11059000	0.090422	1.6265
Al-15% TiC	11097000	0.090117	1.9137
SG Iron	50191000	0.019924	2.5236

H. Graph Comparison

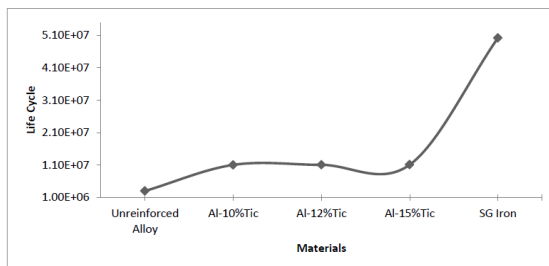


Fig.28 Fatigue Life Cycle Graph

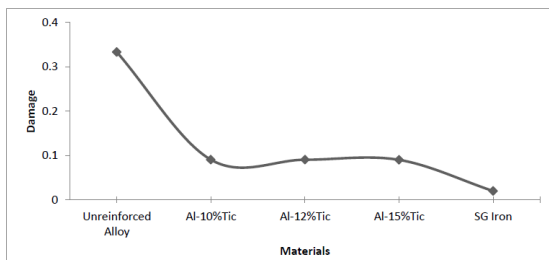


Fig.29 Fatigue Damage Graph

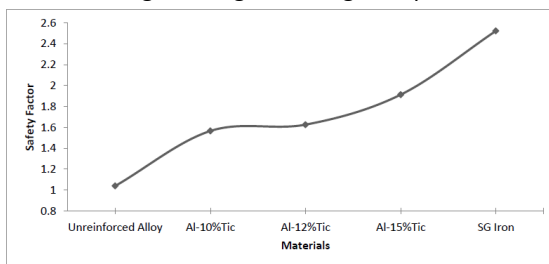


Fig.30 Fatigue Safety Factor Graph

I. Mathematical Calculation of Life

- Goodman theory,
 $(\sigma_a/\sigma_n) + (\sigma_m/\sigma_u) = 1/n$
 - Soderberg theory,
 $(\sigma_a/\sigma_n)^2 + (\sigma_m/\sigma_u)^2 = 1/n$
 - Gerber theory.
 $(\sigma_a/\sigma_n) + (\sigma_m/\sigma_y) = 1/n$
- Al - 15% TiC:
 Young's modulus $E = 133 \text{ N/mm}^2$
 Poisson ratio $\gamma = 0.3087$

Yield strength $\sigma_y = 264 \text{ N/mm}^2$

Ultimate strength $\sigma_u = 323 \text{ N/mm}^2$

Solution

Alternating Stress $(\sigma_a) = (\sigma_{\max} - \sigma_{\min}) / 2 = [200 - (-200)]/2$

$\sigma_a = 200 \text{ MPa}$.

Mean Stress $(\sigma_m) = (\sigma_{\max} + \sigma_{\min}) / 2 = [200 + (-200)]/2$

$\sigma_m = 0 \text{ MPa}$.

Referring S-N curve of Al- 15%TiC Metal Matrix Composite, we can find the total life of the steering knuckle for the alternating stress of 200 MPa.

Total Life (N) = 1.0809e+007 cycles

Number of Cycles Applied (n) = 100000 cycles.

Damage = $n/N = 1000000 / 10809000$

Damage = 0.092

Factor of safety = $\frac{\text{Endurance Strength } h}{\text{Amplitude Stress}} = \sigma_e / \sigma_a = 310/200$

Factor of safety = 1.55

Goodman Theory:

$(\sigma_a/\sigma_n) + (\sigma_m/\sigma_u) = 1/n$

$200/\sigma_n + 0/323 = 1/1.5$

$\sigma_n = 300 \text{ Mpa}$.

Life Estimation:

$N = N_o (\sigma/\sigma_o)^{-0.09}$

$= 106(300/323)^{-0.09}$

$= 1.006e+6 \text{ cycles}$.

Gerber Theory:

$(\sigma_a/\sigma_n)^2 + (\sigma_m/\sigma_u)^2 = 1/n$

$(200/\sigma_n)^2 + (0/323)^2 = 1/1.5$

$\sigma_n = 245 \text{ Mpa}$.

Life Estimation:

$N = N_o (\sigma/\sigma_o)^{-0.09}$

$= 10^6(245/323)^{-0.09}$

$= 1.2824e+6 \text{ cycles}$.

Soderberg Theory:

$(\sigma_a/\sigma_n)^2 + (\sigma_m/\sigma_y)^2 = 1/n$

$(200/\sigma_n)^2 + (0/265)^2 = 1/1.5$

$\sigma_n = 300 \text{ Mpa}$

Life Estimation:

$N = N_o (\sigma/\sigma_o)^{-0.09}$

$= 106(300/323)^{-0.09}$

$= 1.006e+6 \text{ cycles}$.

J. Inference of Fatigue Analysis

It is observed that the steering knuckle fails at the steering arm during fatigue loading. It fails at the

fixed end and needs to be optimized for higher life. It is also observed that; Al- 15%TiC have good fatigue properties. Hence, the steering arm of the steering knuckle has to optimize at the hub end.

IV. OPTIMIZATION OF STEERING ARM

A. Introduction

Optimization is the method of finding the optimized solution for the given objective function. Optimization is done to find the maximum or minimum of the given function (either Absolute or Local extreme).

Technique involved in Optimization:

1. Genetic Algorithm
2. Simulated Annealing
3. Particle swarming

B. Genetic Algorithm

The genetic algorithm is a method for solving both constrained and unconstrained optimization problems. It is based on natural selection. It repeatedly modifies a population of individual solutions. Genetic algorithm can be applied to solve a wide variety of optimization problems that are not well suited for standard optimization algorithms. This includes problems in which the objective function is discontinuous, non-differentiable, stochastic, or highly nonlinear. The genetic algorithm uses three main types of rules at each step to create the next generation from the current population:

- Selection rules select the individuals, called parents that contribute to the population at the next generation.
- Crossover rules combine two parents to form children for the next generation.
- Mutation rules apply random changes to individual parents to form children.

C. Optimization of Steering Knuckle

From the above finite element analysis has been observed that the steering arm region of the steering knuckle is the most vulnerable. Hence, it requires optimization in the steering arm region near the hub region.

D. Optimization for Steering Arm Region

The steering arm can be assumed as an equivalent cantilever beam. Based on the Flexure Equation for beams, the bending stress equation is formulated as the objective function.

$$\frac{M}{I} = \frac{\sigma}{y} = \frac{E}{R} \quad \text{----- 6}$$

The cantilever beam's formula for deflection is taken as the second objective function.

$$\delta = \frac{Wl^3}{3EI} \quad \text{----- 7}$$

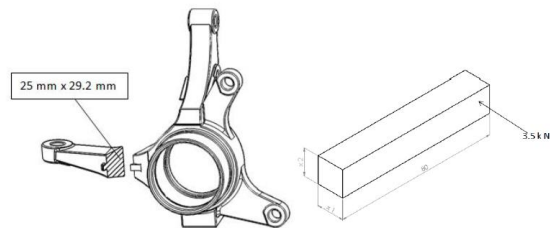


Fig.31 Available Cross section of Steering Arm

E. GA Problem Formulation

A mathematical model for the steering knuckle optimization is done in problem formulation. In MATLAB, the code for optimization is written and optimized using genetic algorithm.

F. Objective Function

To obtain an optimized cross section with minimum bending stress and deflection:

$$f(x_1, x_2) = \frac{Wl^3}{3EI} \quad \text{----- 8}$$

$$g(x_1, x_2) = \frac{My}{I} \quad \text{----- 9}$$

f(x) - Deflection Objective function

g(x) - Stress objective function

M - Bending Moment, y - Neutral axis distance, W - load

L - Length of Cantilever Beam, I - Area moment of inertia

1) *Design Variables:* In the present work the dimension of the cross section is considered as the design variables.

Bounds

$$x_{1max} = 30\text{mm}, x_{1min} = 25 \text{ mm}$$

$$x_{2max} = 35\text{mm}, x_{2min} = 29.2 \text{ mm}$$

2) *Design Parameters:* The parameter which affect the stress and deflection of the beam are considered as

1. M - Bending moment
2. y - Distance of neutral axis from the axis considered,
3. I - Area moment of Inertia,

- 4. W - Load
- 5. L - Equivalent length

3) *Design Constraints*: The constraints considered are of non-linear constraints as involves two independent variables in a single function.

$$c_1 = x_1 \times x_2 \times \rho \leq m \quad \text{-----10}$$

m – 120% of Mass of volume considered before optimization,

$$c_2 = \frac{x_1 \times x_2^3}{12} \geq I \quad \text{-----11}$$

I – Maximum Area moment of Inertia considered before optimization.

4) *Fitness Function*: The fitness Function has non-linear constrained function on undergoing Generations. The best Fitness value gives the optimized dimensions. The code is done using Mat lab.

$$H(x_1, x_2) = \{f(x_1, x_2), g(x_1, x_2)\} \quad \text{-----12}$$

G. Results and Discussions

1) *Deflection*: The election function is taken as rank method which directly depends on the rank of the score. For the reproduction, the elite count is taken about 20% of the population, 80% of the remaining individual undergoes cross over. Only about 16% of the individual is mutated. The algorithm followed, commands the process to stop at the end of 100th generation. The optimized individual corresponds to the dimensions (31.036mmX30mm) with minimum deflection.

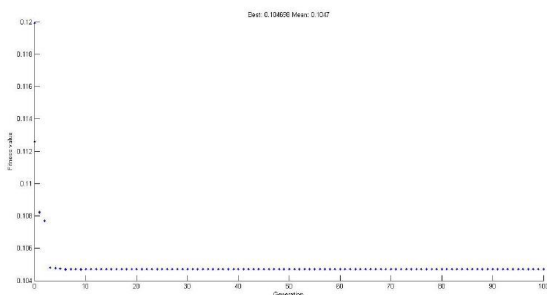


Fig. 32 Best fitness for deflection

2) *Bending Stress*: It is observed that the value of the fitness function decreases from the high score to the low score. In the 100th generation the optimal solution is found. The best fitness value (score) is the one with lowest score, which shows the bending stress is minimum. The optimized individual will have the best score, with minimum deflection for Al-15%TiC. Since, optimisation for deflection and

bending stress yields same cross section, the dimension of the cross section of the steering arm at the hub point is optimised as:

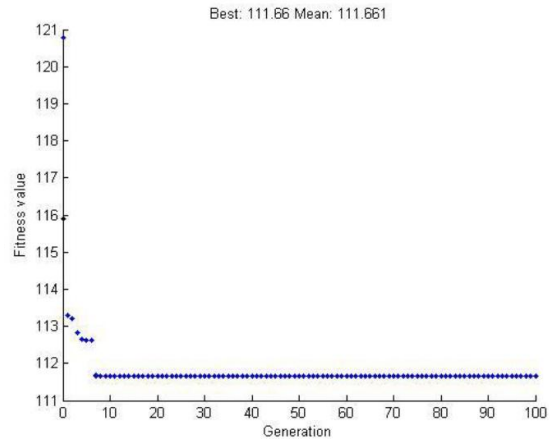


Fig.33 Best fitness for bending stress

TABLE VIII: OPTIMIZED DIMENSIONS FOR MMC

Optimised for Materials	Optimised Cross section $X_1 \times X_2$
SG Iron	25 x 29.2
MMC Al-10% TiC	31.229 x 30
MMC Al-12% TiC	31.155 x 30
MMC Al-15% TiC	31.036 x 30

H. Material Index Number

Strength Index is the ratio of yield strength of the material to the density of the material. Stiffness Index is the ratio of Young's modulus to the density of the material.

$$\text{Stiffness Index} = E/\rho^* \quad \text{Strength Index} = S_y/\rho^*$$

E= Young's Modulus, ρ = density, S_y = Yield strength
 Al- 15%TiC has both highest stiffness index and highest strength index number. Hence, this material is chosen as a best alternative.

TABLE IX: MATERIAL INDEX

Material	Stiffness Number	Strength number
	E/ρ	S_y/ρ
SG Iron	0.7056	2.41
Un reinforced Alloy	1.548	5.06
Al -10% TiC	1.549	5.118
Al -12% TiC	1.551	5.212
Al -15% TiC	1.554	5.71

I. Fatigue performance of MMC after Optimisation

After optimisation of steering knuckle, the cross section at the critical region is increased. The

maximum stress at that region is decreased, and so the fatigue life of the product is increased.

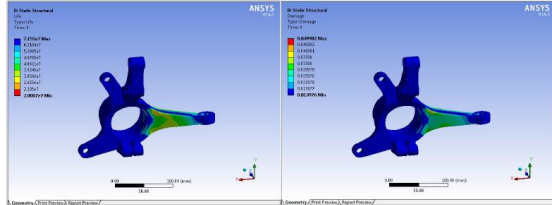


Fig. 34 Fatigue Life of optimized MMC

Fig. 35 Damage of optimized MMC

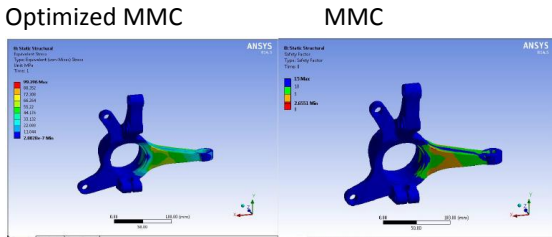


Fig. 36 Equivalent Stress of MMC

Fig. 37 Safety Factor of Optimized MMC

TABLE X: COMPARISON OF FATIGUE RESULTS OF MMC

Parameters	Before Optimization	After Optimization
Stress (MPa)	135.33	99.396
Deformation (mm)	0.33407	0.2458
Life (cycles)	11096000	20070000
Damage	0.090124	0.049982

J. Inference of Optimization

Al-15% TiC has more strength and stiffness index than other materials. Hence, it is best suited for the steering knuckle. After optimisation, same material can withstand higher cycles of fatigue load and the fatigue life of the component is increased. Also, the fatigue performance of optimised MMC knuckle is nearing the performance of SG Iron.

V. IMPACT ANALYSIS

A. Types of Impact Test

There are basically two types of impact test

- Pendulum test
 1. Izod
 2. Charpy
 3. Tensile impact
- Drop weight test

B. Pendulum Test

A pendulum of a known weight is hoisted to a known height on the opposite side of a pivot point.

By calculating the acc. due to gravity (9.8 m/sec^2), one can know that the amount of impact energy at the bottom of the swing. By clamping or supporting a specimen on the bottom, the sample can be released to strike and break the specimen. The pendulum will continue to swing up after the break event to a height somewhat lower than that of a free swing. We can use this lower final height point to calculate the energy that was lost in breaking the specimen.

C. Drop Weight Impact Test

A second method was to drop a weight in a vertical direction, with a tube or rails to guide it during the "free fall." Once again, with the height and weight known, impact energy can be calculated. In the early days, there was no way to measure impact velocity, so engineers had to assume no friction in the guide mechanism. Since the falling weight either stopped dead on the test specimen, or destroyed it completely in passing through, the only results that could be obtained were of a pass/fail nature.

D. Impact Energy

Impact energy is a measure of the work done to fracture a test specimen. When the striker impacts the specimen, the specimen will absorb energy until it yields. At this point, the specimen will begin to undergo plastic deformation at the notch. When the specimen can absorb no more energy, fracture occurs.

E. Importance of Impact Testing

Impact resistance is one of the most important properties for to be considered as a critical measure of service life, product safety and liability.

F. Impact test using FEA

In structural analysis, forces are assumed to be applied slowly and the FEA is performed quasi statically. The resulting stresses can be much higher if a load is the result of an impact incident. Using ANSYS Mechanical APDL FEA tool analyze the steering knuckle under impact condition. Transient analysis type is used to give impact load time dependent load is given. Impact load acts on object for a few fractions of second.

G. Geometry Parameters

Modelling: 3D part using SolidWorks software
 Element type: Solid Brick node 185

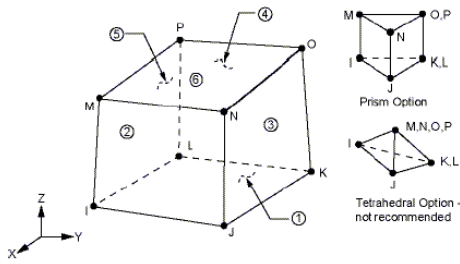


Fig.38 Solid 185 Elements

Eight-node hexahedrons provide reliable FEA solutions since the element is linear ($p = 1$), with a linear strain variation displacement mode.

Meshed Result

No of nodes: 17120

No of elements: 70526

Analysis type: Transient

Define load

Constrain: All degree of freedom arrested at hub surface.

Impact load: 1kN load is applied at steering arm end.

Impact time: 0.1s

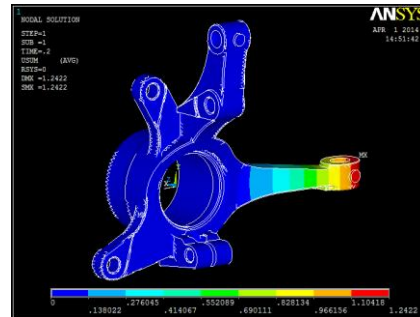


Fig. 41 Deformation of SG Iron

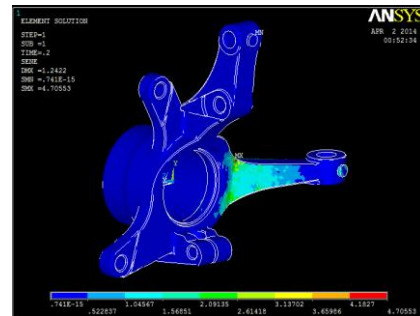


Fig.42 Strain Energy of SG Iron

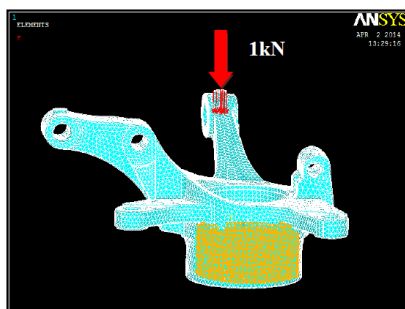


Fig. 39 Impact Loading Region
 H. Impact Analysis Result

I. Comparison of results of MMC knuckle

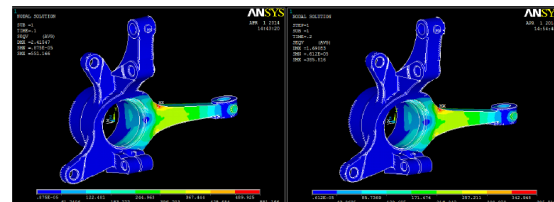


Fig. 43 Equivalent Stress of MMC before optimization
 Fig.44 Equivalent Stress of MMC after optimization

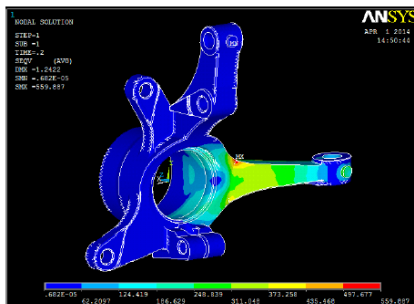


Fig. 40 Equivalent Stress of SG Iron

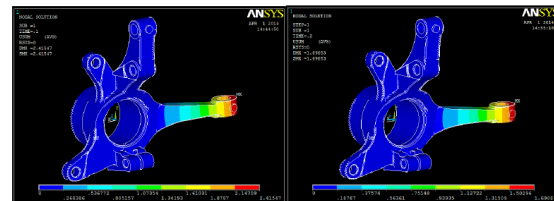


Fig. 45 Deformation of MMC before Optimization
 Fig. 46 Deformation of MMC after Optimization

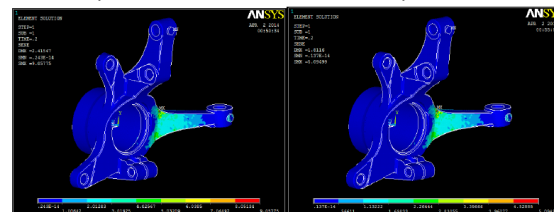


Fig.47 Strain Energy of MMC before Optimization
 Fig.48 Strain Energy of MMC after Optimization

TABLE XI: COMPARISON OF RESULTS

Material	Induced Stress (MPa)	Deflection (mm)	Energy Observed (J)
SG Iron	559.8	1.2	4.7
Al Tic 15%	551.1	2.4	9.0
Optimized Al Tic 15%	385.8	1.6	5.1

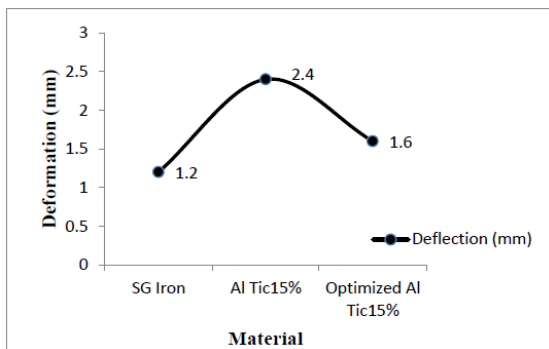


Fig. 49 Comparison of Deflection

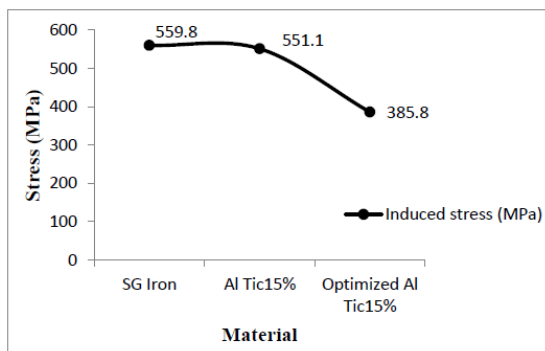


Fig. 50 Comparison of Stresses

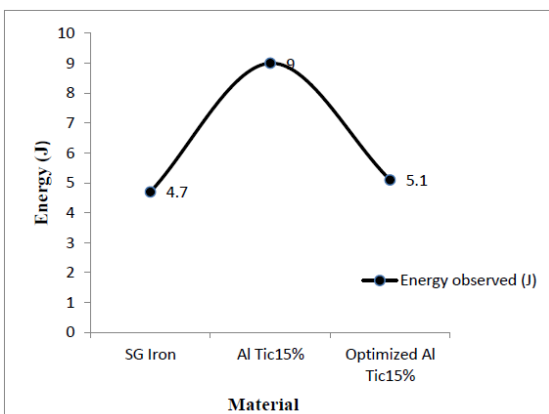


Fig. 51 Comparison of Energy

J. Inference of Impact Analysis

Impact load of 1kN with impact time 0.1s is applied. 550MPa stress is induced. Due to lower value of elastic modulus and ultimate strength, MMC

experience more deflection and is less safe than SG iron. But after optimization of MMC knuckle, the same impact load is applied and a reduced stress of 385MPa is induced. Deflection is also reduced after optimization. Optimized Al TIC15% has greater energy absorption than the SG Iron.

VI. VIBRATION ANALYSIS

A. Software Analysis of Vibration

In finite element method, frequency of free is obtained by using Modal analysis in ANSYS software.

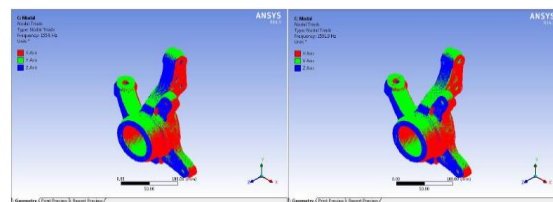


Fig. 52 First set frequency of free vibration (1554 Hz) Fig.53 Second set frequency of free vibration is (1591.9 Hz)

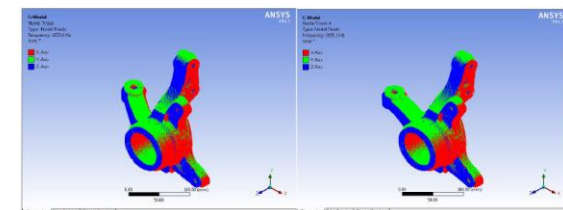


Fig. 54 Third set frequency of free vibration (1622.6 Hz) Fig. 55 Fourth set frequency of free vibration (1695.1Hz)

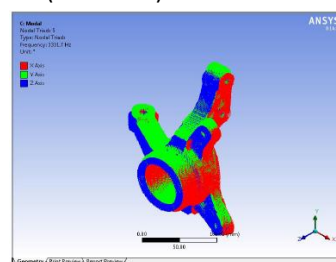


Fig. 56 Fifth set frequency of free vibration (3331.7 Hz)

B. Experimental Analysis of Vibration

FFT analyzer test is used to verify the obtained values of free vibration of steering knuckle.

C. Experimental setup

The PULSE software analysis was used to measure the frequency ranges to which the foundation of various machines are subjected to under full and no load conditions. The equipment

used is - Modal Hammer, Accelerometer, Portable pulse, Connection- Model no: AO 0087D, Specimen, Display Unit.

D. Result of FFT Analyzer:

TABLE XII: RESULT OF FFT ANALYZER

Frequency Set	Frequency of FFT Analyzer (Hz)
1	1554.0
2	1591.1
3	1622.6
4	1695.1
5	3331.7

E. Comparison of Frequency of Software and FFT Analyzer

TABLE XIII: COMPARISON OF FREQUENCY OF SOFTWARE AND FFT ANALYZER

Frequency Set	Software	FFT Analyzer	% of Error
1	1554.0	1521	2.123
2	1591.1	1583	0.509
3	1622.6	1592	1.885
4	1695.1	1670	1.480
5	3331.7	3318	0.411

From the above tabled data the error between the software frequency and the experimental frequency are very low, from this we have to conclude that the software results are more reliable for further modal analysis.

F. Software Analysis result for MMC

Before optimisation After optimisation

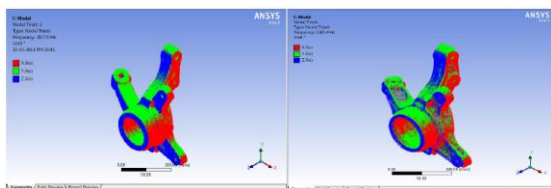


Fig.57 First set frequency of free vibration (1827.6 Hz)

Fig.58 First set frequency of free vibration (2163.4 Hz)

G. Result comparison of MMC

TABLE XIV: COMPARISON OF RESULTS OF MMC BEFORE AND AFTER OPTIMIZATION

Frequency Set	Before Optimization	After Optimization
1	1827.6	2163.4
2	1872.0	2183.7
3	1918.0	2870.2
4	1958.0	3118.7
5	3857.8	4433.7

H. Inference of Vibration Analysis

From the comparison table the natural frequencies of the MMC (Al-15% TiC) material is varies after optimization, based on this the steering knuckle will not meet any failure because the natural frequency is higher than the before optimization frequency.

VII. RESULTS

The optimized knuckle has increased strength and performance. The stress is reduced from 135.33 MPa to 99.396 MPa. Hence, the knuckle is optimized with a sacrifice of weight of 33.22 grams.

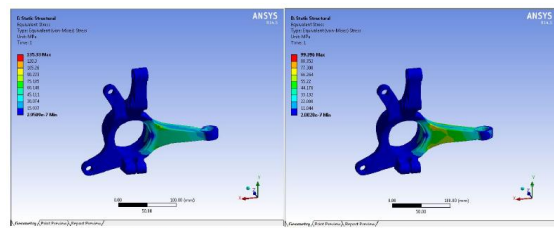


Fig 59 Stress before Optimisation

Fig 60 Stress after Optimisation

The deformation is reduced by nearly 0.1 mm, and is reduced below the deformation of knuckle made of SG Iron.

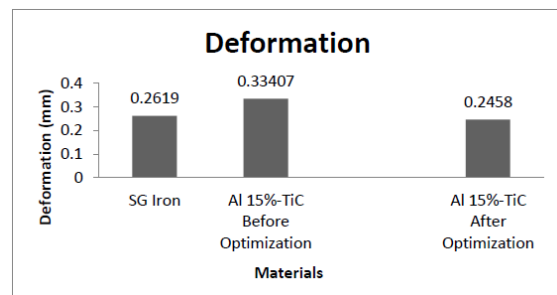


Fig. 61 Deformation Graph

It is also evident that the fatigue life of the optimized knuckle is increased to 2.01e+7 cycles from 1.1e+7 cycles. Although the fatigue life of SG Iron is 5.02e+7 cycles, optimized MMC can withstand about 40% of the life as SG Iron.

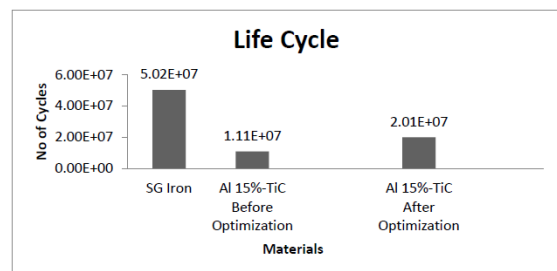


Fig.62 Fatigue Life Cycle Graph

It is proved that the damage during fatigue loading is reduced from 0.09 to 0.04 and is nearer to that of SG Iron knuckle of 0.02.

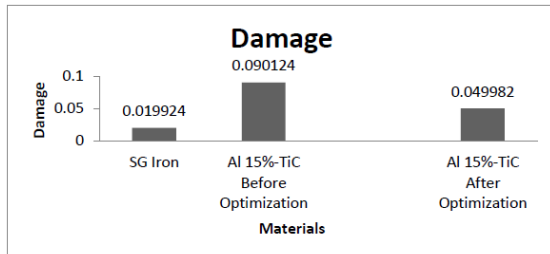


Fig. 63 Fatigue Damage Graph

It is proved that the safety factor during fatigue loading is increased from 1.9103 to 2.6551 and is greater than the safety factor of SG Iron knuckle of 2.5236.

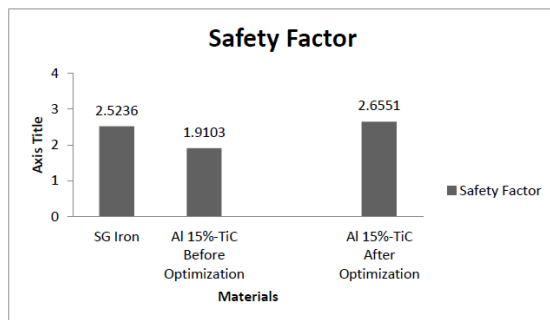


Fig. 64 Fatigue Safety Factor Graph

Impact analysis after optimization proved that the component can withstand higher load compared to the former one. The stress produced during impact load is reduced from 550MPa to 385.8 MPa by optimization.

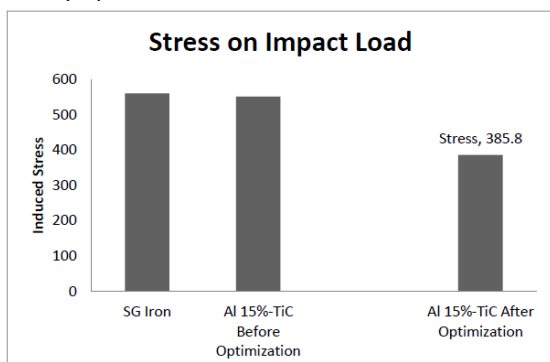


Fig. 65 Stresses on Impact Load Graph

Modal analysis proved that the variation in the frequency is smaller and optimization of MMC knuckle increased the frequency.

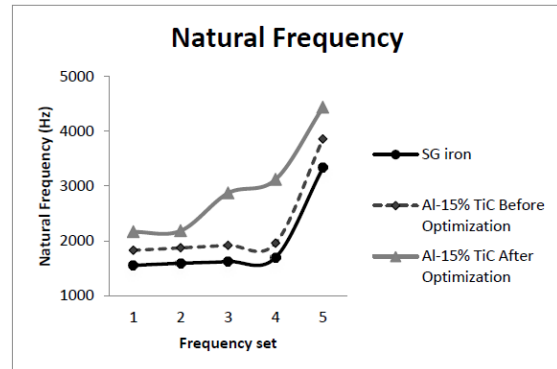


Fig. 66 Natural Frequency Graph

VII. CONCLUSION

Design of steering knuckle is developed by using design software SolidWorks in standard dimension of YE2 model knuckle. The knuckle is first analysed for the multi-axial force during cornering, acceleration and braking. As a result of static analysis the region of failure is guessed as steering arm. Then fatigue analysis is performed on the steering arm for the steering force. The point of failure is localised and the cross section is studied. Then the knuckle is optimised for different materials using Genetic Algorithm in MATLAB. The Metal Matrix Composite Al-TiC 15% is selected by calculating material index number. Then the knuckle is optimised for the optimum cross section and the increase in fatigue performance as a result of optimisation is compared. Then impact analysis and vibrational analysis are performed to compare the performance of optimised knuckle with the former non-optimised knuckle. As a result, the MMC knuckle is optimised to the performance of SG Iron knuckle (2.3463 kg) with an increase of mass from 1.267 kg to 1.300 kg. Thus the weight is reduced by 44.59 % from SG Iron.

REFERENCE

- [1]. V.Sivananth, "Evaluation of fatigue and impact behavior of titanium carbide reinforced metal matrix composites"
- [2]. S. Vijayarangan, "Evaluation of metal matrix composite to replace spheroidal graphite iron for a critical component, steering knuckle"

-
- [3]. Reddy J.N., "An Introduction to the Finite Element Method", McGraw Hill, International Edition, 1993.
- [4]. Logan D.L., "A First Course in the Finite Element Method", Third Edition, Thomson Learning, 2002.
- [5]. Cook, Robert Davis et al "Concepts and Applications of Finite Element Analysis", Wiley, John & Sons, 1999.
- [6]. Segerlind L.J., "Applied Finite Element Analysis", John Wiley, 1984.
- [7]. S.S.Rao, "Finite Element Analysis", 2002 Edition.
- [8]. Zienkiewicz, O.C. and Taylor, R.L., "The Finite Element Method", Fourth Edition, Volumes 1 & 2, McGraw Hill International Edition, Physics Services, 1991.
- [9]. Bathe K.J., "Finite Element Procedures in Engineering Analysis", Prentice Hall, 1990.
- [10]. R. d'Ippolito, , Improving the fatigue life of a vehicle knuckle with a reliability-based design optimization approach.
-