DESIGN OPTIMIZATION OF AN AUTOMOBILE TORQUE ARM USING SIMULATION, TOPOLOGY OPTIMIZATION AND TESTING

Abhisarika Khare, Prof. Amol Patil

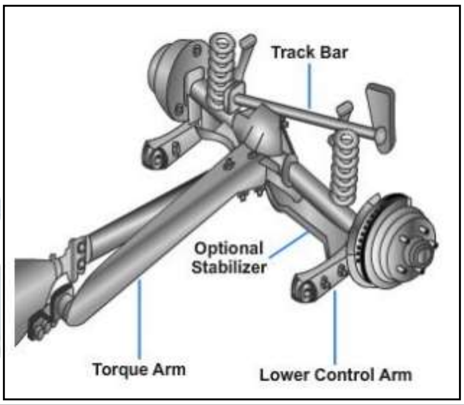
Dr D Y Patil School of Engineering, Ajeenkya D Y Patil University, Pune

Abstract- The design of lightweight automotive structures has become a prevalent practice in the automotive industry. This study focuses on design optimization of an automobile torque arm subjected to cyclic loading. Starting from an available initial design, the shape of the torque arm is optimized for minimum weight such that the fatigue life of the torque arm does not fall below that of the initial design and the maximum von Misses stress developed in the torque arm does not exceed that of the initial design. The stresses are computed using ANSYS 19 finite element software and 3D model using CATIA software. Weight optimization of torque arm is performed using topology optimization module from ANSYS workbench. Experimental validation of optimize torque arm model is performed using stain gauge and UTM.

Keyword: Design optimization, Torque arm, UTM.

I. INTRODUCTION

The torque arm has been used to reduce power-hop for solid axle suspension vehicles. However, the fundamental mechanism of torque arm action has not yet been fully explained. Donald Margolis investigated the stability of a truck which has a trailing arm extended rearwards. He used conventional eigenvalue analysis of the linearized equations of motion of the truck [2]. We will analyze the suspension equipped with a torque arm by the use of coupling forces. The approach enables a simple and effective understanding of this suspension. The concept of the coupling forces, furthermore, can also be extended to the analyses of other suspensions.



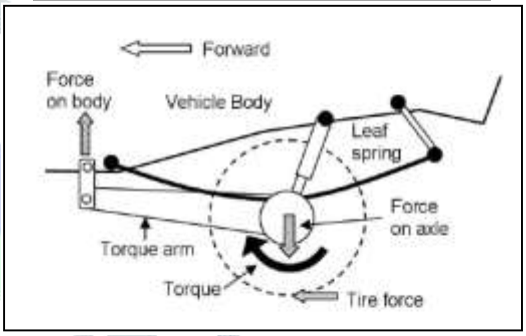


Figure 1 shows an example of a solid axle suspension with a torque arm. The torque arm is fixed to an axle. A vertical link connects the torque arm and body through two joints. These joints can be designed so that the torque arm transfers only torque and not axial load. The torque arm resists wind up of the axle and increases the torsional stiffness of the suspension. It also generates the force pushing down on the axle with accelerating torque. Without a torque arm, two spring elements of the example suspension determine the torsional stiffness of the suspension. One is the torsional stiffness of half shafts inside the axle connecting the wheel and differential gear. The other is the torsional stiffness of a leaf spring. These two are serial springs. With a torque arm, most of torque is supported by the very stiff torque arm not by leaf springs. Thus, overall torsional stiffness of system is mainly determined by the half-shaft. In the case of a solid axle suspension with a coil spring, the torsional stiffness of the suspension is mainly determined by the stiffness of the half-shafts.

A torque arm provides additional resistance to help prevent axle wind up. It is attached to the axle housing and runs forward under the vehicle parallel to the drive shaft between the rear axle and transmission, cushioned through a bracket to allow some flex.

HOW DOES THE TORQUE-ARM IMPROVE TRACTION AND HANDLING?

The complete Torque-arm Suspension System replaces the troublesome upper control arms with the Torque-arm and Panhard Bar. This separates the two functions once performed by the upper arms. The Torque Arm controls axle housing rotation during acceleration and braking, and the Panhard Bar controls the side-to-side position of the axle housing. With each part dedicated to a specific job, each can perform its function without compromise, greatly improving the performance of your Mustang.

The Torque-arm plants the rear tires much better during acceleration than the stock four-link design, and the Panhard bar locates the axle much more precisely. Your cars overall performance will be greatly improved, with much better acceleration out of corners, along with improved handling, stability, better straight-line launching ability, and increased predictability..

ADVANTAGES OF TORQUE ARM:

- The Torque-arm controls rear axle housing rotation.
- This design is not over-constrained and allows the rear suspension to move smoothly into any combination of ride height and roll angle.
- Those components do all the jobs the 4-link did; there's nothing left for the upper control arms to do. They are no longer needed and are discarded.
- Each part is dedicated to a specific job, and so each can perform its function without compromise.

II. LITERATURE REVIEW

Vyankatesh Sneha A. Bramhankar, P. N. Deshmukh, et.al [1], The expectation of this paper is to examine the impact of every boundary and along these lines the blend of different boundaries on firmness and modular recurrence. This paper will give the rules to the creator for planning DE connector for various appraisals. Drive End Adaptor needed for interfacing the alternator and flywheel lodging is displayed utilizing creo. Solidness in even, vertical and pivotal bearing because of power just as second are coming more prominent than benchmark esteem. Reaction surface enhancement method is utilized to improve the drive end connector. The impact of four boundaries, for example, rib thickness, rib width, connector ID and pitch circle distance across of alternator side are noticed. Streamlining plot and relapse condition are produced. Plan of new drive end connector is 20 rate not exactly the seat mark esteem which saves the material expense. Thus the greatest weight is diminished and this works on the benefit. Bowing recurrence of the planned connector with alternator is more prominent than the benchmark esteem. Thusly drive end connector against bowing recurrence is protected. In this manner according to primary solidness perspective plan of drive end connector is protected.

Mr.Pankaj Shende, Mr. Satish Sonawane et.al [2], In this exploration paper, the disappointment (crack) of Torque arm section was chosen as examination subject. The focal point of this task is to explore how a break proliferates and fills a regular way of Torque arm section However; this investigation is proposed for presenting crack mechanics from an application perspective. It basically centers around both pressure and weariness analyses.WAG-9 sort electric trains, of Indian Railways' armada utilized for Goods train and traveler train kept up with at Ajani, loco shade of focal railroad has the historical backdrop of continuous disappointment of the force arm section .The investigation of disappointments uncovered that, the specific time and occurrence of disappointment can't be pretty much as determined as it doesn't stop the loco and it is seen just when the loco comes for support. The issue distinguished here is the break disappointment of Torque arm section that has been tackled by insightful and FE investigation. The breaks saw close to the side of section flopped because of the shifting burden which cause for the weakness disappointment. This issue has been dissected by thinking about the two cases, static burden and dynamic burden.

Youthful Jin Seo, Kenneth L. Oblizajek and Steven P. Fuja et.al [3], This investigation clarifies the dependability of strong pivot, force arm suspension vehicles under hefty speed increase and slowing down. Scientific methods use ordinary straight investigation and a non direct coupling power in a 4 level of opportunity dynamic model. The force arm is a basic however successful gadget among these instruments to create the strong on-direct coupling power. The force arm successfully changes the rubbing slip bend among tire and street by presenting a non-direct coupling power and viably adding damping. This paper completely clarified the soundness and insecurity of vehicles furnished with strong hub, force arm suspensions during weighty speed increase and slowing down occasions.

G. Chiandussi, I. Gaviglio, A. Ibba et.al[4], The paper shows the outcomes acquired by utilizing a geography enhancement code to take care of a three-dimensional issue concerning a real automotive part. The carried out advancement strategy depends on the boost of the complete expected energy with a volume limitation by optimality measures. The volume of the ideal arrangement relies upon the forced static (removal, stress, firmness) and dynamic (normal recurrence) requirements and has not to be determined deduced. The enhancement interaction joins toward a very distinct design made of the base material with an almost no level of components described by transitional material properties.

The proposed geography streamlining strategy has been used to tackle a numerous stacking condition issue concerning the McPherson back suspension sub edge of a medium size vehicle. The calculation of the ideal geography is very clear cut with under 1% of the plan space components with a transitional worth of material properties (Young modulus and material thickness). The ideal primary arrangement agrees with the necessities concerning the greatest Von Misses stresses and the base first normal recurrence of the segment. The outcomes got from the geography enhancement issue have been utilized to characterize the speculative ideal state of the segment. The mechanical qualities of the part with its last mathematical setup are marginally not quite the same as those got toward the finish of the streamlining issue because of an enormous decrease of the main normal recurrence. The mechanical property variety is stringently connected to the quantity of still mass components and their format into the plan space.

Varatharaj Neelakandan, Thulsirajan Ganesan, Praveen chakrapani rao et.al [5], This paper presents the plan improvement techniques for hearty plan to meet the item weight and weariness life cycle necessities considering the assembling practicalities of projecting divider thickness and draft recompenses with meta stream range for the starter working conditions. In the advancement of lodging section, limited component examination programming was utilized for ascertaining the pressure and weariness life and afterward the weight was enhanced. The improved new lodging section was proposed for new starter undertakings to catch new business and the plan was executed to decrease cost in productionized starter.

Dr.K.K.Dhande, Prof .N.I.Jamadar, Sandeep Ghatge. et.al [6], In this exploration paper they say that, the cutting edge car businesses are supplanting accelerator and grasp pedal by light weight materials like polymer composites, plastic, aluminum and its amalgams, and so forth The reason for substitution is decrease weight, cost, and improvement in consumption resistance. In aeronautics; the steel material is supplanted by light materials. In this examination different lightweight materials are contrasted and traditional steel for brake pedal. These materials are broke down for various segments for various stacking and limit conditions. The point of this examination is to plan and investigate the brake pedal utilizing CATIA and ANSYS programming.

Mohd Nizam Sudin, Musthafah Mohd Tahir, Faiz Redza Ramli, et.al[7], In this paper they say,automotive industry is proceeding to make progress toward light weight vehicle in further developing eco-friendliness and discharges decrease. To deliver a superior exhibition vehicle plan vehicles with ideal weight. To diminish the heaviness of vehicle without forfeiting its honesty, this venture intends to utilize geography streamlining procedure to propose an ideal plan of a car part in beginning stage of item advancement. In this undertaking the material utilized for a current brake pedal is unaltered as this examination centers around diminishing load of existing brake pedal without material replacement. The advanced model of a current brake pedal was produced utilizing CATIA V5 strong demonstrating programming. Geography improvement was performed by utilizing Altair Optistruct programming under straight static pressure investigation. At last, another light weight configuration brake pedal is proposed. The aftereffect of the examination shows that the heaviness of another planned brake pedal was 22% less when contrasted with a current brake pedal without forfeiting its exhibition necessity.

Dr Hossein Saidpour et.al[8], The selection of materials for vehicle parts is reliant upon a particular organic market measure, subject to a tough arrangement of necessities. In addition to other things, this incorporates monetary viability, wellbeing, recyclability and lightweight execution. Metals like steel, aluminum and magnesium are basically utilized for components of the body design and boards. Plastics are applied for outside connections to the body. Vehicles comprise fundamentally of steel and iron, however it is normal that the measure of steel and iron utilized is diminished because of the looming utilization of multimaterial developments. Materials, for example, steel will be subbed, specifically by aluminum, magnesium and plastics. Options in contrast to the steel uni-body are multimaterial unibodies and aluminum space outlines. Steel and magnesium space outline ideas for volume applications are as yet a work in progress. More customary materials are being supplanted by superior carbon fiber composites for their toughness and explicit strength/firmness.

III. PROBLEM STATEMENT

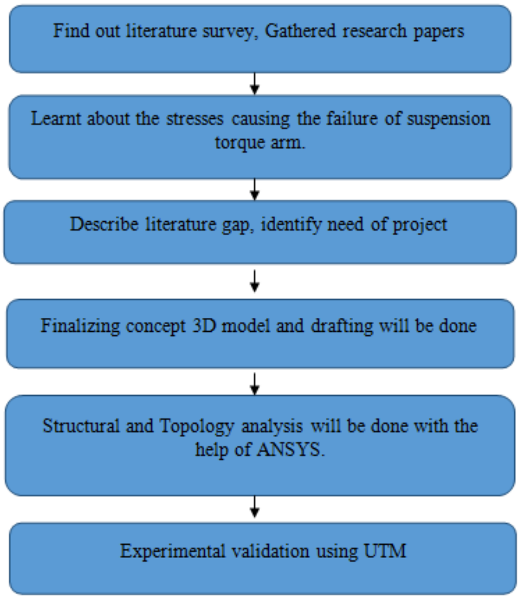
In this study, weight optimization of an automobile torque arm subjected to cyclic loading is conducted. The torque arm is a component of an automobile suspension system mounted on the rear-drive axle and allows the automobile to accelerate in a straight line without rotation of the rear axle.

.

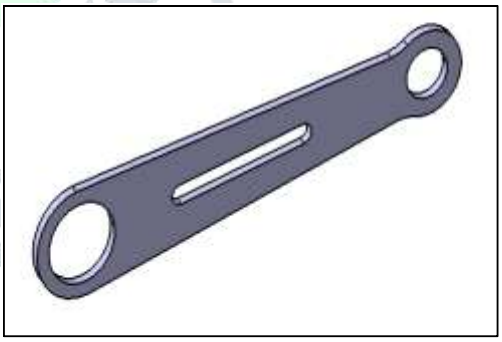
IV. OBJECTIVES

- In this study, design optimization of an automobile torque arm subjected to cyclic loading will be perform.
- Starting from an available initial design, the shape of the torque arm was optimized for minimum weight such that the fatigue life of the torque arm did not fall below that of the initial design and the maximum von Mises stress developed in the torque arm did not exceed that of the initial design.
- Weight optimization of torque arm will be perform using topology optimization module from ANSYS workbench.
- Experimental validation of optimize torque arm model perform using stain gauge and UTM.

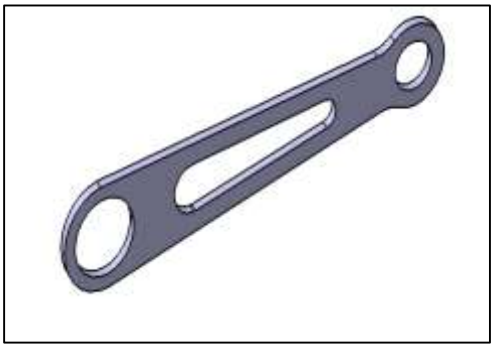
V. METHODOLOGY



3D Cad model prepared in Catia R20 Torque arm model . Modeling of Torque arm model done using surfacing tool.



Torque arm model



Optimized Torque arm model

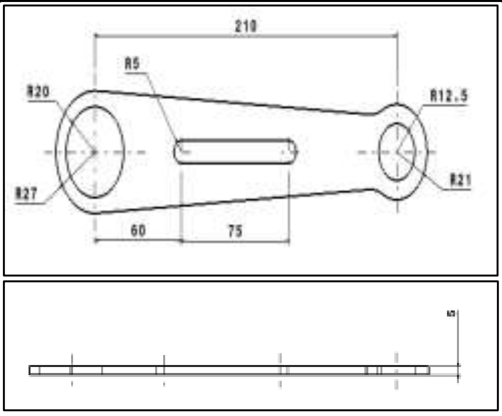
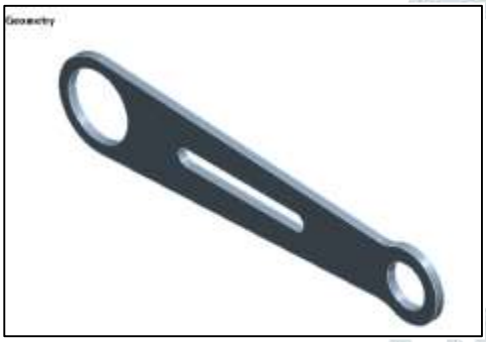


Fig. Drafting of Torque arm

FEA (FINITE ELEMENT ANALYSIS).
STRUCTURAL ANALYSIS:

STATIC STRUCTURAL ANALYSIS OF TORQUE ARM MODEL:
GEOMETRY:



MATERIAL USED:

Properties of Outline Row 3: Carbon steel

	A	B	C
1	Property	Value	Unit
2	Material Field Variables	Table	
3	Density	7.845	g cm^-3
4	Isotropic Elasticity		
5	Derive from	Young's Modulu...	
6	Young's Modulus	2.1E+05	MPa
7	Poisson's Ratio	0.3	
8	Bulk Modulus	1.75E+11	Pa
9	Shear Modulus	8.0769E+10	Pa
10	Tensile Yield Strength	415	MPa
11	Tensile Ultimate Strength	620	MPa

MESHING:

NODES AND ELEMENTS:

Statistics	
Nodes	1821
Elements	206

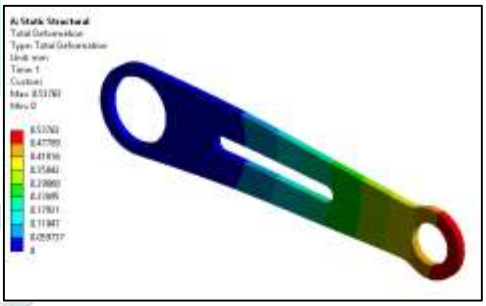
WEIGHT OF EXISTING CASING:

Properties	
Volume	41445 mm^3
Mass	0.32514 kg

BOUNDARY CONDITION:
A boundary condition for the model is the setting of a known value for a displacement or an associated load. For a particular node you can set either the load or the displacement but not both.

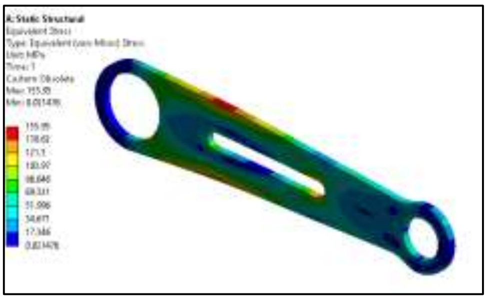
As the torque arm is connected to the rear dead axle to avoid the twisting of the rear dead axle. These torque arms are mainly used in vintage muscle cars or sports car and which are very difficult to find on Indian roads. So we have modelled a scale down model of a Experimental Torque arm to carry out the testing the force applied on the Torque arm model is taken from the research. This force is mainly applied due to the self weight of the car. So we are taking the value of a force as 1266.50N.

RESULTS AND PLOTS:
TOTAL DEFORMATION PLOT:



The total deformation observed in the Torque arm model of AISI 1040 carbon steel is very minimal which is 0.53763 mm.

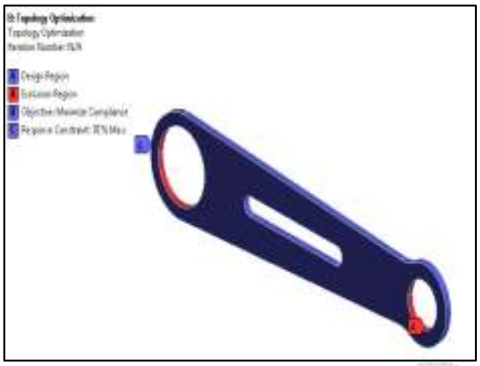
EQUIVALENT STRESS PLOT:



The equivalent stress developed inside torque arm is 155.95 MPa, which is less than Yield strength of the material i.e 415 MPa.

TOPOLOGY OPTIMIZATION

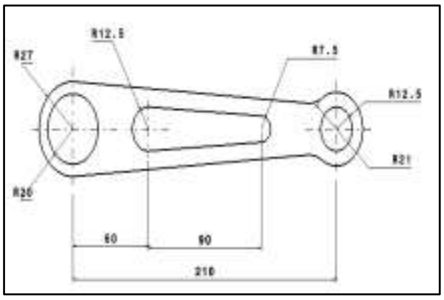
PROCESS OF TOPOLOGY OPTIMIZATION:
Based on Hyper Works platform topology optimization holder, first, according to the engine mounting position, we establish the three-dimensional geometric model of engine bracket, and then pretreated in Hyper-Mesh, define design area, objective function and constraints under the optimization panel.
TOPOLOGY OPTIMIZATION BOUNDARY CONDITION:



WE ARE GOING TO RETAIN 70%OF MASS, MEANS WE CAN REDUCE APPROXIMATLY 30% MASS

RESULTS AND PLOTS:

TOPOLOGY DENSITY TRACKER



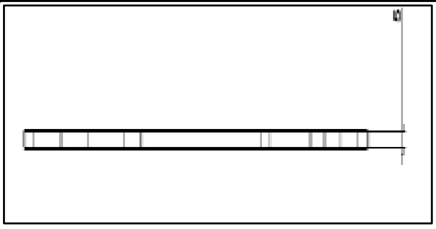
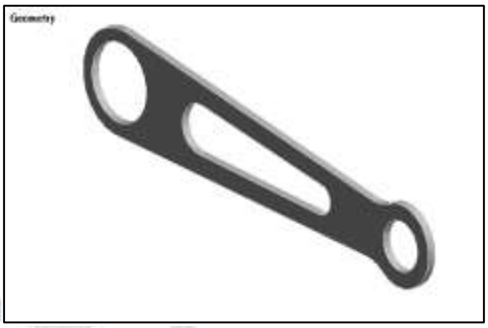


Fig. Grafting of optimized model of Torque arm.

GEOMETRY:



MESHING:

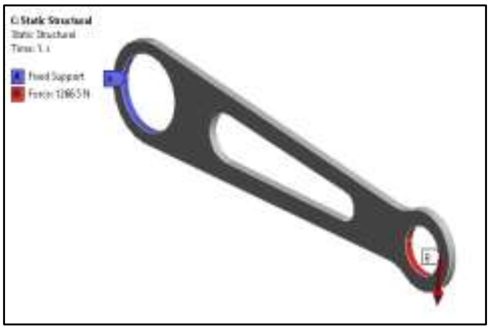
Statistics	
Nodes	1683
Elements	187

NODES AND ELEMENTS:

WEIGHT OF OPTIMIZED CASING:

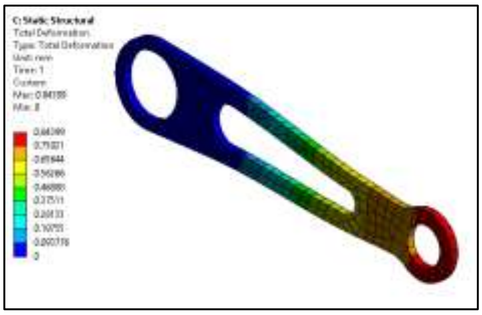
Properties	
Volume	32835 mm^3
Mass	0.25759 kg

STRUCTURAL ANALYSIS BOUNDARY CONDITION:



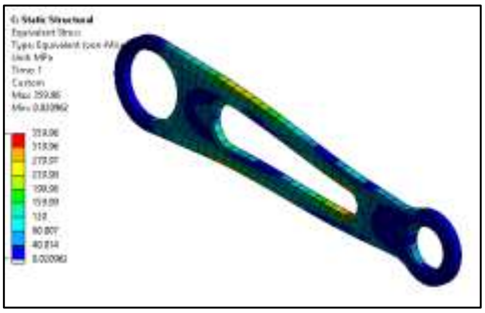
\

TOTAL DEFORMATION PLOT:



The total deformation observed in the Torque arm model of AISI 1040 carbon steel is very minimal which is 0.84399 mm.

EQUIVALENT STRESS PLOT:

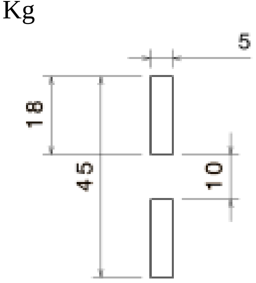


The optimized Torque arm model induces stress about 359.96 Mpa which is much lesser than the Yield strength of the material used which 415 MPa. Hence the optimized casting is safe.

We are going to find out the Maximum bending stress at Cross sections, where the slots are provided
So as we know the formula for finding Maximum bending stress

$\sigma b = \frac{My}{I}$

M= Calculated bending moment
Y= Vertical distance away from the neutral axis
I= Moment of Inertia Around the neutral axis
CROSS SECTION 01
The force applied the Arm is 1266.5 N = 1266.5/9.81 = 130 Kg



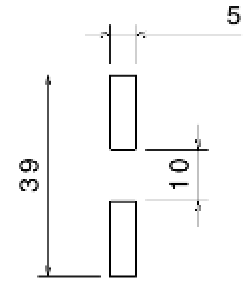
Y= 45/2 = 22.5 mm
Moment of inertia = BH3/12 – bh3/12 =(5*453/12) – (5*103 /12) =37969 – 416.67

= 37552 mm 4
Distance from the neutral axis is = 22.5 mm
Maximum bending moment of Cross section
This a cantilever beam so the formula for cantilever maximum bending moment is as follows,
As the cantilever is 80mm from fixed support.
Mb = WL = 1266.5 X 210 = 265965 N-mm
As the length of the beam is 210 mm

$\frac{Mb}{I} = \frac{\sigma b}{Y}$

265965 * 22.5 / 37552 = 159.36 N/mm2
As we know the N/sq.mm= MPa
So the theoretical bending moment is calculated as
As we are taking factor of safety as 1,
σb= Syt / F.S = 415 / 1 = 415 MPa

CROSS SECTION 02
Now if we calculate the bending stress at smaller cross section area of the Torque arm
The force applied on torque arm is 1266.5 N



Y= 39/2 = 19.5 mm

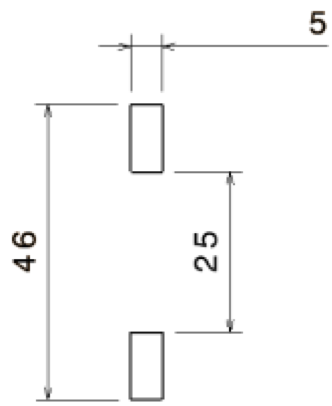
Moment of inertia = BH3/12 – bh3/12 = (5*393/12) – (5*103/12) = 24344.58 mm4
Maximum bending moment of Cross section
This a cantilever beam so the formula for cantilever maximum bending moment is as follows,
As the cantilever is 135 mm from fixed support.
Mb = WL = 1266.5 X 210 = 265965 N-mm

$\frac{Mb}{I} = \frac{\sigma b}{Y}$

265965* 19.5 / 24345 = 213.03 N/mm2
So the theoretical bending moment is calculated as
As we are taking factor of safety as 1,
σb= Syt / F.S = 415 / 1 = 415 MPa
So the Existing torque arm is safe.

Same procedure we will do for Optimized model

OPTIMIZED MODEL CROSS SECTION 01
Now if we calculate the bending stress at cross section area of the Torque arm
The force applied on torque arm is 1266.5 N
Y= 46/2 = 23 mm



Moment of inertia = BH3/12 – bh3/12 = (5*463/12) – (5*253/12) = 34047 mm4

Maximum bending moment of Cross section

This a cantilever beam so the formula for cantilever maximum bending moment is as follows,
As the cantilever is 80 mm from fixed support.
Mb = WL = 1266.5 X 210 = 265965 N-mm

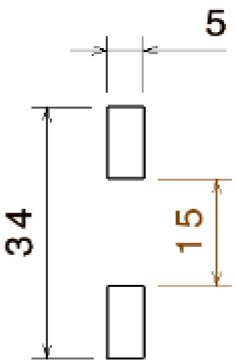
$$\frac{Mb}{I} = \frac{\sigma b}{Y}$$

265965* 23 / 34047 = 179.66 N/mm2
So the theoretical bending moment is calculated as
As we are taking factor of safety as 1,
σb= Syt / F.S = 415 / 1 = 415 MPa

CROSS SECTION 02
Now if we calculate the bending stress at cross section area of the Torque arm
The force applied on torque arm is 1266.5 N

Y= 34/2 = 17 mm
Moment of inertia = BH3/12 – bh3/12 = (5*343/12) – (5*153/12) = 14971 mm4

Maximum bending moment of Cross section
This a cantilever beam so the formula for cantilever maximum bending moment is as follows,
As the cantilever is 150mm from fixed support.

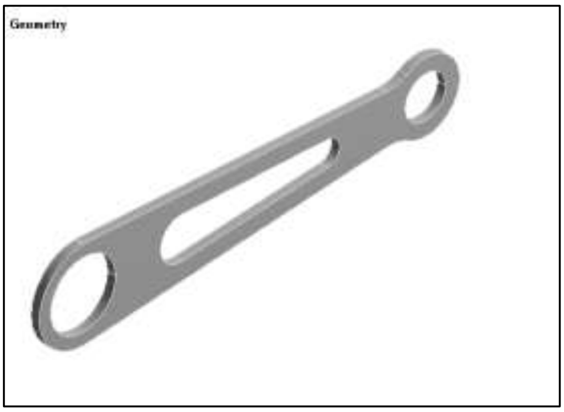


Mb = WL = 1266.5 X 210 = 265965 N-m

$$\frac{Mb}{I} = \frac{\sigma b}{Y}$$

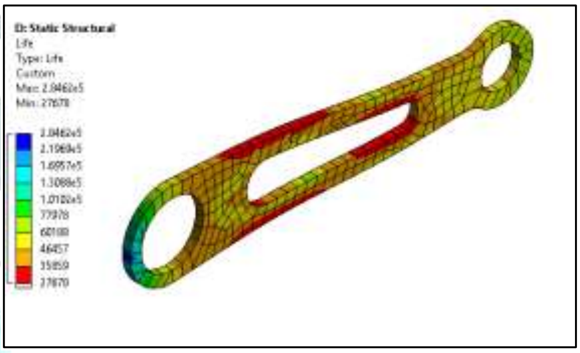
265965* 17 / 14971 = 302 N/mm2
So the theoretical bending moment is calculated as
As we are taking factor of safety as 1,
σb= Syt / F.S = 415 / 1 = 415 MPa
Hence the Optimized design is safe

FATIGUE ANALYSIS OF OPTIMIZED TORQUE BAR:
GEOMETRY:



Details of "Fatigue Tool"	
Domain	
Domain Type	Time
Materials	
Fatigue Strength Factor (Kf)	1.
Loading	
Type	Fully Reversed
Scale Factor	1.
Definition	
Display Time	End Time
Options	
Analysis Type	Stress Life
Mean Stress Theory	Goodman
Stress Component	Equivalent (von-Mises)
Life Units	
Units Name	hours
1 cycle is equal to	2.7778e+005 hours

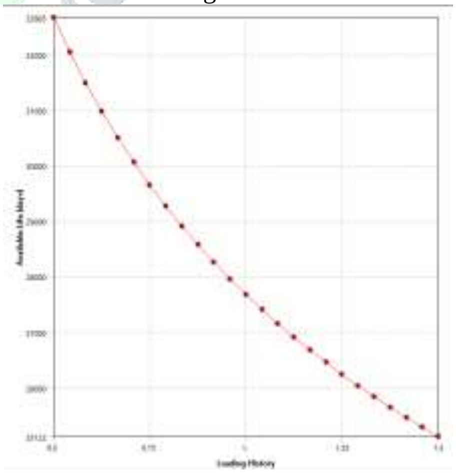
LIFE OF TORQUE ARM IN NO OF DAYS:



MINIMUM NUMBER OF DAYS:

Results	
Minimum	27678 days
Minimum Occurs On	Part1

Fatigue Life



Sr.no	Model	Total deformation (mm)	Equivalent stress (MPa)	Weight (Kg)
1.	Existing model	0.53763	155.95	0.325
2.	Optimized model	0.84399	359.96	0.257

The weight of the torque arm is reduced by 20 %.

EXPERIMENTAL VALIDATION:

A Universal Testing Machine (UTM) is used to test both the tensile and compressive strength of materials. Universal Testing Machines are named as such because they can perform many different varieties of tests on an equally diverse range of materials, components, and structures.
Universal Testing Machines can accommodate many kinds of materials, ranging from hard samples, such as metals and concrete, to flexible samples, such as rubber and textiles. This diversity makes the Universal Testing Machine equally applicable to virtually any manufacturing industry.
The UTM is a versatile and valuable piece of testing equipment that can evaluate materials properties such as tensile strength, elasticity, compression, yield strength, elastic and plastic deformation, bend compression, and strain hardening. Different models of Universal Testing Machines have different load capacities, some as low as 5kN and others as high as 2,000kN.

SPECIFICATION OF UTM

1	Max Capacity	400KN
2	Measuring range	0-400KN
3	Least Count	0.04KN
4	Clearance for Tensile Test	50-700 mm
5	Clearance for Compression Test	0- 700 mm
6	Clearance Between column	500 mm
7	Ram stroke	200 mm
8	Power supply	3 Phase , 440Volts , 50 cycle. A.C
9	Overall dimension of machine	2100*800*2060
10	Weight	2300Kg

- Fixture is manufactured according to component designed.
- Single force is applied as per FEA analysis and reanalysis is performed to determine strain by numerical and experimental testing.
- Strain gauge is applied as per FEA results to maximum strained region and during experimental testing force is applied as per numerical analysis to check the strain obtained by numerical and experimental results.
- During strain gage experiment two wires connected to strain gage is connected to micro controller through the data acquisition system and DAQ is connected to laptop. Strain gage value are displayed on laptop using DEWESOFT software.

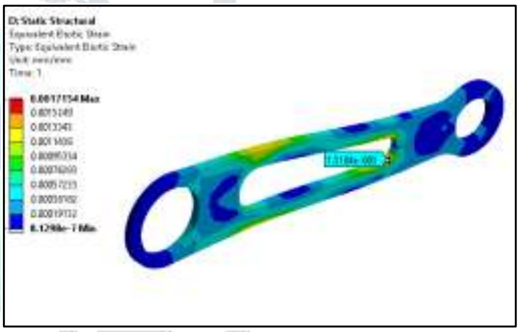
PROCEDURE:

Experimental validation of optimized Torque arm is done by strain gauge technique.

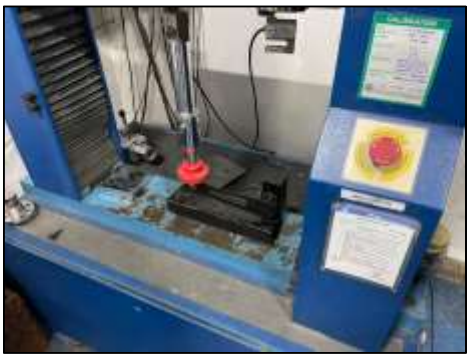
The optimized model of torque is mounted on the fixture for calculating the strain values obtained.

The torque arm fixed at one end and the force is applied on the other end of the arm. The force applied is 1266.5 N

After the application of force the maximum strain developed inside torque is observed.



The strain developed on the optimized model is 1510 Micron.



CONCLUSION

The Torque arm is optimized by doing Topology analysis on the digitalized model. After optimizing the structural analysis is carried out which gives us the result that the optimization done on the Torque arm does not have any effect on the performance of the Torque arm as the stress induced inside the casing is much lesser than the yield strength of the material used. 20% weight reduction has been achieved by optimization process.

	Baseline	Optimized
Von Misses Stress	155 MPA	360 MPA
Weight	325 gms	257 gm

Fatigue analysis of Optimized torque is also calculated by analytical method the fatigue of the torque arm is determined in terms of day's i.e 27678 days. This life can easily be considered as infinite life for the application.

The strain induced in torque arm is calculated analytically as 1510 micron and the Validation of experimental value of strain is done by strain gauge method. The experimental value of strain is observed to 1513 Micron.

REFERENCES

1. Sneha A. Bramhankar, P. N. Deshmukh, Swapnil "Design, Optimization and Stiffness Analysis of De Adaptor for Alternator" IJIRSETVol. 5, Issue 6, June 2016

2. Mr.Pankaj Shende, Mr. Satish Sonawane "Analysis and Optimization of Torque arm bracket"

3. Young-Jin Seo, Kenneth L. Oblizajek "Stability Analysis of Solid Axle, Torque Arm Suspension Vehicles under Heavy Acceleration and Braking".

4. G. Chiandussi, I. Gaviglio, A. Ibba"Topology optimization of an automotive component without final volume constraint specification."

5. Varatharaj Neelakandan, Thulsirajan Ganesan"Weight optimization of Housing motor for Electrical starter Motor Using FEA"

6. Dr.K.K.Dhande, Prof.N.I.Jamadar, Sandeep Ghatge "Conceptual Design and Analysis of Brake Pedal Profile".

7. Mohd Nizam Sudin, Musthafah Mohd Tahir"Topology Optimization in Automotive Brake Pedal Redesign"

8. Dr. Hossein Saidpour "Lightweight High Performance Materials for Car Body Structures".