INDIRA GANDHI ENGINEERING COLLEGE SAGAR (M.P.)

Session 2017-2018

A MAJOR PROJECT REPORT
ON

"DESIGN AND ANALYSIS OF BICYCLE CRANK ARM"

Submitted to the Rajiv Gandhi Proudyogiki Vishwavidyalaya Bhopal (M.P.) In Partial Fulfillment Of The Degree Of Bachelor Of Mechanical Engineering

Under the supervision of

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8TH SEMESTER
MECHANICAL ENGINEERING
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INDIRA GANDHI ENGINEERING COLLEGE
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CERTIFICATE

This is to Certify that "**Vaibhav Soni** " Of B.E. 8th Sem , Mechanical has completed Major project report on "**DESIGN & ANALYSIS OF BICYCLE CRANK ARM**" towards the partial fulfillment of the requirement for the award of Degree in Mechanical Engineering of the **Rajiv Gandhi Proudhyogiki Vishwavidyalya Bhopal** for the session 2017-18.

The work presented in this report has been carried out by his under my guidance and supervisions.

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ACKNOWLEDGEMENT

It is with great reverence that I express my gratitude to our guide **" Prof. VINAYAK BAREWAR "** Department of Mechanical Engineering, Indira Gandhi Engineering college, Sagar (M.P.) for his precious guidance and help in this Major Project report . The credit for the successful completion of this report goes to his keen interest, timing, guidance and valuable suggestion otherwise our endeavor would have been futile.

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I owe sincere thanks to Principal "**Dr. N.L.Prajapati"** for His kind support which he rendered us in the envisagement for the great success of my Major Project report.

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INDIRA GANDHI ENGINEERING COLLEGE SAGAR (M.P.)

DECLARATION

I hereby declare that the following Major Project Report on "**DESIGN & ANALYSIS OF BICYCLE CRANK ARM"** is an authentic work done by me. I undertake the project as a part of the course curriculum of Bachelor of Engineering from MECHANICAL ENGINEERING of Indira Gandhi Engineering Collage, SAGAR affiliated by Rajiv Gandhi Proudyogiki Vishwavidyalaya, Bhopal (M.P.).

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CONTENTS

Abstract

The objective of this project is to analyze a bike crank arm which transfers reciprocating motion of the rider’s legs into rotational motion. In this work attempt is to be made for design and analyze in bike crank arm with composite materials.

Presently bike crank arms are manufactured using aluminum alloy, titanium, carbon fiber, chromoly steel, or some less expensive steel are used in bike crank arm. Composite bike crank arm is possessed with higher potential life. The design parameters are analyzed with the objective of minimizing the weight of composite bike crank arm.

The design is showed that significant potential improvement in the performance of bike crank arm. the modelling in the bike crank arm assembly is to be done by using CATIA V5 & ANSYS DM. CATIA V5 is used to design the bike crank arm and FEA analysis is obtained successfully by using ANSYS Workbench, The FEA results are used to estimate the deflection and stresses under subjected loads. Further comparison is to be carried out for both steel and composite materials to analyze the bending, deformation and stress induced of the bike crank arm.

LIST OF ABBREVIATION

FEA	Finite Element Analysis
FEM	Finite Element Method
ANSYS	Analysis System
CATIA	Computer-aided three-Dim. interactive application
MATLAB	Matrix Laboratory
DM	Design modeler
WB	Workbench
CAD	Computer-aided design
CAM	Computer-aided manufacturing
CAE	Computer-aided engineering
UD	Uni-directional
U.S.	United States
PDE	Partial Differential Equations
BC	Boundary condition
GFRP	Glass Fiber Reinforced Plastic
CF	Carbon Fiber
CFRP	Carbon fiber reinforced polymer

LIST OF SYMBOLS

A	*Area*
E	*Modulus of elasticity*
F	*Force*
g	*Acceleration of gravity*
M	*Bending moment*
I	*Moment of inertia*
δ	*Deflection*
σ_Y	*Yield stress*
V	*Shearing force*
ν	*Poisson's ratio*
u, v, w	*Components of displacement*
x, y, z	*Rectangular coordinates*
γ	*Shear strain*
δ	*Deflection*
ε	*Normal strain*
θ	*Angle*
σ	*Normal stress*
τ	*Shear stress*
Φ	*Angle of inclination*
ρ	*Density*

CHAPTER 1

INTRODUCTION

CHAPTER – 1

Introduction

1.1 GENERAL

Cycling is very common and famous sport and way of travelling at the same time. Human powered vehicle can be ideal means of transport and sport. Among the components of bicycle, the crank arm is the most important component for a bicycle to move forward.

Bike crank is the component of a bicycle that converts the reciprocating motion of the rider's legs into rotational motion. The motion is use to drive the chain and turn the wheel of the bicycle when moving. This crank is the most important part of a bicycle since it is where the motion is produced to move the bicycle. The importance of this crank arm is so great, that the design and the analysis of the crank arm need to be done thoroughly in order for the bicycle to be safe for riding. For the purpose of this project, a simple design of the crank arm is used to be analysed.

The finite element analysis will be done solely by using the ANSYS software, where using this software, the steady state analysis will be done upon the design and the largest displacement as well as the weakest point on the design will be determined.

When a rider is pedalling the bicycle, she or he is applying a variable force to this bike crank. To simplify the analysis, we'll neglect the variation of the force with time and focus on the response of the crank to a static force.

The aim of this project is to analyse the structural parameter of the bike crank arm with respect to time. We did analysis on ANSYS using above assumptions and we found Deformed shape and displacement field, Stress distribution in the crank. The units we used are in U.S. Customary units (in inches).

1.2. OUTLINE THE PROJECT

Chapter 1	Contains the Introduction of the project
Chapter 2	Contains the Literature survey
Chapter 3	Contains the detail of Numerical modeling and formulation
Chapter 4	Contains the detail of Designing in CATIA and ANSYS Design Modeler
Chapter 5	Contains the Static Structural Analysis in ANSYS WORKBENCH
Chapter 6	Contains the result and analysis
Chapter 7	Contains the conclusion and future scope

CHAPTER 2

LITERATURE SURVEY

CHAPTER 2

LITERATURE REVIEW

Bicycle plays an inherent role in our life. Bicycle riding is a globally popular sport and an economic transportation. The performance of crank depends on the weight of the bicycle. Optimization of weight and structure of the bicycle crank is the best scope of optimizing the overall performance of the bicycle. When the rider ridding bicycle on rough surface, the induced vibrations will cause the fatigue of its rider and the fracture of its crank. This project deals with the study of the structural design, modal analysis and optimization of bicycle crank by using composite material with help of FEA.

Firstly, structural analysis, numerical results obtained by applying dynamic loading condition. Secondly, the Modal analysis is used to identify modes of bicycle crank to calculate natural frequencies and mode shapes by using Finite Element Analysis. Finally, the analyzed crank is then optimized to reduce weight without affecting their capacity to be resistant to mechanical stresses.

In our project Although more expensive bicycles have titanium or carbon fiber crank sets, most use Al 6061-T6 because it is cheap, light, and durable. So, we in our project have used Al 6061-T6.

In this study, various literatures were reviewed about the crank material and its design.

Case study 1 : Material Optimization of Crank

Case study 2 : Stress Analysis of Bicycle Paddle

2.1 Stress Analysis of Bicycle Paddle

S. Abey Gunasekara and T.M.M. Amarasekara discussed the stress analysis of bicycle paddle and optimized by finite element method, describes and proposed improvements of designs with regard to minimize the weight, cost and optimum factor of safety. Failure of paddle crank means the progressive of sudden deterioration of their mechanical strength because of loading effect. Paddle make materials shown different properties as a result many advantages as well as disadvantages. However material strength should have ability to withstand an applied stress without failure. Generally, cranks are manufactured of an aluminum alloy, titanium, carbon fiber, chromyl steel or other less expensive steel. The pedal force is changing every second in the process of turning the pedal and magnitude and direction of pedal force is different according to different riding posture. First half of the round pressure is positive and second half pressure is negative. Maximum load is coming vertically downward and magnitude is depending on the road condition, slope of the road and as well as weight of the rider. In this literature considered 95 % man's weight of the population is about 116Kg. This is the maximum load acting on pedal as well as crank in downward. Due to this load bending stress in crank and it will create twisting of the crank. The maximum bending stress gives the load acting at the end of the pedal.

From this literature, we found that there is a maximum stress in sharp edges in the crank near to fixed hole so need to apply some fillets on sharp edges and more thickens near to fixed hole than the pedal fixing hole by adding material. Always it is needed to keep equivalent stress as much as low. It will benefit to durability of the component.

2.2 Material Optimization of Crank

Ventzi G. Karaivanov and David A. Torick discussed the title of optimization of a bicycle crank and spider using finite element software, describes the evolution of a bicycle crank and spider design. The pedal is attached to the bicycle crank and the spider is the component that transfers the torque produced from the pedal and crank arm length to the chain sprocket. They show the deflection and stress analysis of an entry level design.

Design change was to select another material to decrease the weight of the crank and spider assembly. They selected a Glass Fiber Reinforced Plastic (GFRP). This material has a Young's Modulus of 26 GPa and a Poisson's Ratio of 0.28. This material change caused a substantial decrease in total mass of the model. The estimated weight for the crank/spider assembly decreased from 265grams (for Aluminum alloy) to 176.6 grams. However, our maximum stress of 75 MPa is much closer to the yield stress of 125 MPa for GFRP. The deflection of the end of the crank arm has also increased by a factor of three from 0.66 mm to 2.1mm. Filets throughout the model are necessary to avoid stress concentrations and injuries from sharp edges.

CHAPTER 3

NUMERICAL MODELING AND FORMULATION

CHAPTER – 3

Numerical Modeling and Formulation

1.1 Introduction

The mathematical model is composed of governing equations, domain and boundary conditions.3D elasticity and Euler-Bernoulli beam theory are two different mathematical models of the same problem.

The classical theory of Euler-Bernoulli's beam assumes that :

1. Cross-sectional plane perpendicular to the axis of the beam remain plane after deformation.
2. The deformed cross-sectional plane is still perpendicular to the axis after deformation.
3. The classical theory of beam neglect the transverse shearing deformation, where the transverse shear is determined by the equation of equilibrium.

In Euler – Bernoulli beam theory, shear deformations and rotation effects are neglected, and plane sections remain plane and normal to the longitudinal axis. Euler-Bernoulli beam theory is used for the hand calculations in the bike crank.

Moment: a tendency to produce motion, especially about an axis.

Area Moment of Inertia : It is a geometrical property of an area which reflects how its points are distributed with regard to an arbitrary axis.

Normal Stress : It is a stress that occurs when a member is loaded by an axial force.

Shear Stress : It is the component of stress coplanar with a material cross section.

3.2. Finite Element Analysis

The Finite Element Analysis (FEA) is the simulation of any given physical phenomenon using the numerical technique called Finite Element Method (FEM). Engineers use it to reduce the number of physical prototypes and experiments and optimize components in their design phase to develop better products, faster.

It is necessary to use mathematics to comprehensively understand and quantify any physical phenomena such as structural or fluid behaviour, thermal transport, wave propagation, the growth of biological cells, etc. Most of these processes are described using Partial Differential Equations (PDEs). However, for a computer to solve these PDEs, numerical techniques have been developed over the last few decades and one of the prominent ones, today, is the Finite Element Analysis.

Differential equations can not only describe processes of nature but also physical phenomena encountered in engineering mechanics. These partial differential equations (PDEs) are complicated equations that need to be solved in order to compute relevant quantities of a structure (like stresses ($\epsilon\epsilon$), strains ($\epsilon\epsilon$), etc.) in order to estimate a certain behaviour of the investigated component under a given load. It is important to know that FEA only gives an approximate solution of the problem and is a numerical approach to get the real result of these partial differential equations. Simplified, FEA is a numerical method used for the prediction of how a part or assembly behaves under given conditions. It is used as the basis for modern simulation software and helps engineers to find weak spots, areas of tension, etc. in their designs. The results of a simulation based on the FEA method are usually depicted via a colour scale that shows for example the pressure distribution over the object.

3.3. Mathematical Model

Governing Equations (3D Equilibrium Equations):

$$\frac{\partial \sigma_x}{\partial x} + \frac{\partial \tau_{xy}}{\partial y} + \frac{\partial \tau_{xz}}{\partial z} = 0$$

$$\frac{\partial \tau_{xy}}{\partial x} + \frac{\partial \sigma_y}{\partial y} + \frac{\partial \tau_{yz}}{\partial z} = 0$$

$$\frac{\partial \tau_{xz}}{\partial x} + \frac{\partial \tau_{yz}}{\partial y} + \frac{\partial \sigma_z}{\partial z} = 0$$

Boundary Conditions :

Traction or displacement specified at every point at every point on boundary in normal and tangential direction.

Additional Equations:

Hooke's Law:

$$\begin{bmatrix} \sigma_x \\ \sigma_y \\ \sigma_z \\ \tau_{yz} \\ \tau_{xz} \\ \tau_{yx} \end{bmatrix} = \frac{E}{(1+\nu)(1-2\nu)} \begin{bmatrix} 1-\nu & \nu & \nu & 0 & 0 & 0 \\ \nu & 1-\nu & \nu & 0 & 0 & 0 \\ \nu & \nu & 1-\nu & 0 & 0 & 0 \\ 0 & 0 & 0 & 1-2\nu & 0 & 0 \\ 0 & 0 & 0 & 0 & 1-2\nu & 0 \\ 0 & 0 & 0 & 0 & 0 & 1-2\nu \end{bmatrix} \begin{bmatrix} \epsilon_x \\ \epsilon_y \\ \epsilon_z \\ \gamma_{yz} \\ \gamma_{xz} \\ \gamma_{yx} \end{bmatrix}$$

Strain-Displacement Relations:

$\epsilon x=(\partial u/\partial x)$ $\epsilon y=(\partial v/\partial y)$ $\epsilon z=(\partial w/\partial z)$

$\gamma_{xy}=\partial u/\partial y+\partial v/\partial x$ $\gamma_{yz}=\partial v/\partial z+\partial w/\partial y$ $\gamma_{zx}=\partial w/\partial x+\partial u/\partial z$

We need to determine 15 unknown functions:

6 stress components: σ_x, σ_y,σ_z,τ_{xy},τ_{xz},τ_{yz}

6 strain components: ϵ_x, ϵ_y, ϵ_z, γ_{xy} ,γ_{xz} ,γ_{yz}

3 displacement components: u,v,w

Now no. of equations is equal to no. of unknown functions, so the problem is fully determined.

3.4. Force Calculation

Force Analysis :

The Figure below shows a bicycle going uphill at an angle of inclination Φ, and with a velocity V.

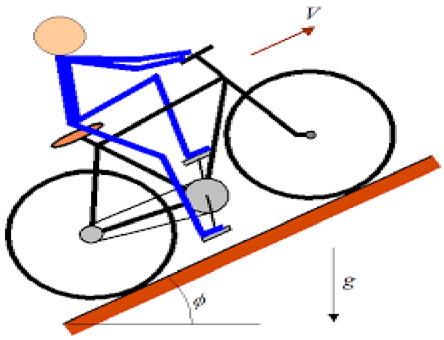


Fig. - Bicycle rider at uphill position

To propel the bicycle uphill the rider must push down on the pedals. The pedals are offset 180° which means that only one pedal can be pushed at a time, from the top position to the bottom position, and then switching to the other pedal.

Given a force F1 acting on the pedal we can calculate the resulting force F4 between the rear wheel and ground. This is the force that propels the bicycle forward. Hence, we can treat this as a static problem.

Consider the figure below , with forces and radial dimensions shown.

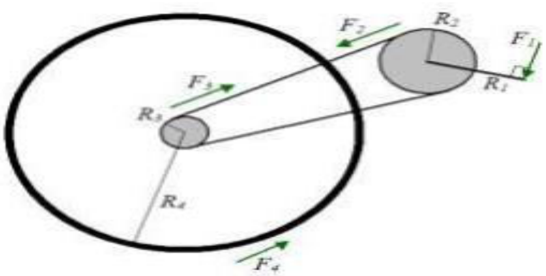


Fig. - Force Diagram

[For this project we take parameters from Hercules Atlas Roadster bicycle]

Where:

F1 is the force applied to the pedal

R1 is the pedal radius=180 mm

F2 is the force acting on the main crank, due to chain contact

R2 is the main crank radius=100 mm

F3 is the force acting on the rear gear, due to chain contact

R3 is the rear gear radius=50 mm

F4 is the force acting on the rear wheel, due to contact with the ground.

Note that the coefficient of static friction between wheel and ground must

be large enough to support this force, otherwise slipping occurs

R4 is the rear wheel radius=350 mm

Using the static equilibrium assumption, we can write the following torque equations:

F1*R1=F2*R2 (1)

F3*R3=F4*R4 (2)

Since F2 = F3,

we can combine the above two equations to give an expression for F4:

F4= F1*R1*R3 / R2*R4 (3)

The force F4 is what propels the bicycle forward.

If we assume that the bicycle is moving at constant velocity (no acceleration) then the force F4 must equal the resisting forces opposing the bicycle's motion. These resisting forces are gravity, rolling resistance, air drag, and internal bicycle friction. If we neglect the latter we can then write the following mathematical expression [A]:

Where:

$F = mg \sin \Phi + C_r\, mg(\cos \Phi) + (1/2)C_d\, \rho A v^2$ (4)

Where:

m= weight of rider=100kg

g=acceleration of gravity = 9.8 m/s2

F is the force propelling the bicycle forward. Note that F ≡ F4

Cr is the coefficient of rolling resistance, which can be 0.0022 to 0.005 for bicycle tires [B]

Cd is the drag coefficient=1.2 [C]

ρ is the density of the air through which the bicycle is moving=1.2 Kg/m3

A is the projected cross-sectional area of the bicycle + rider perpendicular to the flow direction (that is, perpendicular to v) =0.5 m^2 [D] and v is the speed of the bicycle relative to the air=18.5Km/h=5.1 m/s

The first term on the right side of the above equation is the gravity contribution. The second term is the rolling resistance contribution. The third term is the air drag contribution.

If we consider Φ= 40[maxi. condition]

By putting all these values in formula (4), we get

F4= (100*9.8* sin 40)+ (0.005*100*9.8*cos40)+ (1/2*1.2*1.2*0.5*5.1*5.1)

F4=82.686 N (5)

We know that,

F4= F1*R1*R3 / R2*R4

We know values of F4, R1, R2, R3, and R4

From this we calculate F1 which is pedal force.

82.686=F1*180*50 / 100* 350

F1= 82.686*100*350 / 180*50

F1=321.6 N (6)

But for our calculation purpose we choose US customary unit system

Loading Force = 100 lbf (444.82N)

Direction of This force is upward as Opposite Reaction to pedal force.

This is an approximation of the actual loads and constraints on the bike crank.

3.5. Numerical calculation

This program is designed to calculate the structural parameters of bike crank arm due to impact. MATLAB software R2017a (License: Free For All)

MATLAB Command Window

```
>> %Material: Aluminium 6061-T6

%Recorded Deflection in Ansys = 0.042 in

L= 6.69; %Length of the bar (in)

h= 0.9055; %Height of the bar (in)

t= 0.375; %Thickness of the bar (in)

I=1/12*t*h^3; %Moment of inertia (in^4)

E= 10*10^6; %Modulus of Elasticity (psi)

y_max = 0.042; %Maximum deflection (in)

y = h/2; %Distance from centroid to bottom surface (in)

A = h*t; %Cross-section area of the bar (in^2)

Sy = 40000; %Yield Strength of the material (psi)

%Find the applied force using deflection (lbf)

F= y_max*3*E*I/L^3;

V= F; %Maximum shear force (lbf)

M= F*L; %Maximum bending moment (lbf.in)

%Find Maximum bending stress
```

```
bending_x = M*y/I;

bending_y = 0;

%Find Maximum shear stress

shear_xy = 3*V/(2*A);

%Find principle stresses

sig_1 = (bending_x+bending_y)/2 + sqrt(((bending_x+bending_y)/2)^2+shear_xy^2);

sig_2 = (bending_x+bending_y)/2 - sqrt(((bending_x+bending_y)/2)^2+shear_xy^2);

%Find Von-Misses Stress (psi)

von_misses = sqrt(sig_1^2 - sig_1 * sig_2 + sig_2^2);

% Find the factor of safety

n = Sy/von_misses;

%Output results

fprintf('The force due to deflection = %.2f lbf\n\n',round(F))

fprintf('The maximum bending stress = %.2f psi\n\n',round(bending_x))

fprintf('The maximum shear stress = %.2f psi\n\n',round(shear_xy))

fprintf('The principle stress 1 = %.2f psi\n\n',round(sig_1))

fprintf('The principle stress 2 = %.2f psi\n\n',round(sig_2))\

fprintf('The Von-misses stress = %.2f psi\n\n',round(von_misses))

fprintf('The factor of safety = %4.4f\n\n',n) %Press Enter
```

The force due to deflection = 98.00 lbf

The maximum bending stress = 12746.00 psi

The maximum shear stress = 431.00 psi

The principle stress 1 = 12761.00 psi

The principle stress 2 = -15.00 psi

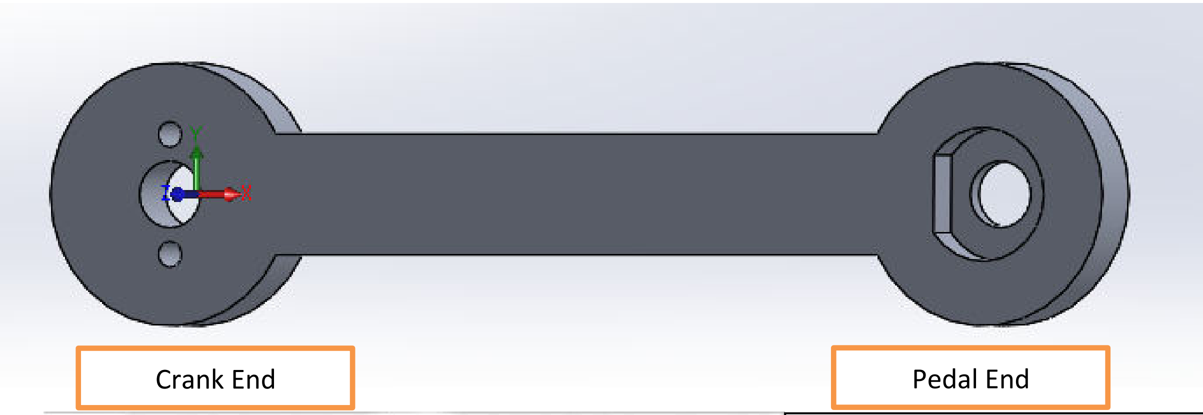
The Von-misses stress = 12768.00 psi

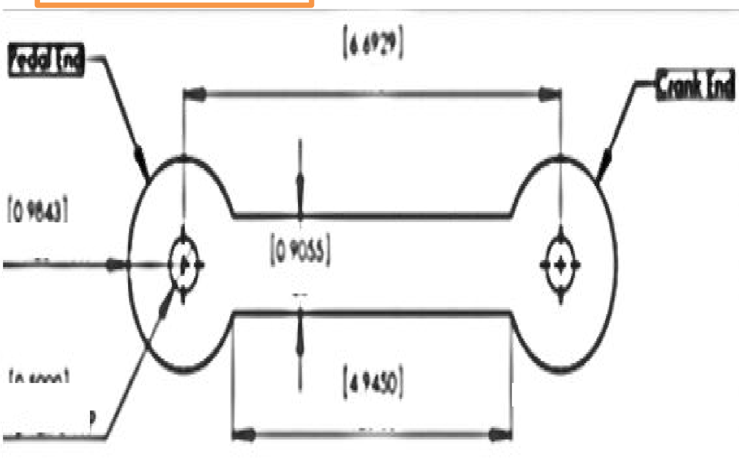
The factor of safety = 3.1328

>>

3.6. Hand Calculation

3-D Model in SolidWorks





Dimentions

1. Centre to Centre Distance = 6.69 in
2. Arm Height =0.9055 in
3. Arm Length =4.9450 in
4. Hole Diameter = 0.5 in
5. Outer arc Radius = 0.9845 in
6. Depth =0.375 in
7. Force at pedal end = 100lbf (=444.48N)

1.Stress in x plane in x direction (σ_{xx})

M=F*L= (100 lbs)*(6.69 in) = 669 lb-inches

I=1/12(thickness)(height)^3 =1/12(0.375)(0.9055)^3 = 0.02320 in^4

σ_{xx} =(M*y)/I =(669 lbs-ins*0.45275 in)/0.02320 in^4=13055.59psi

2.Stress in y plane in y direction (σ_{xx})

$\sigma_{yy} = 0$

3.Stress in x plane in y direction (τ_{xy})

τ_{xy} =(3F/2A)

F=100 lbf t=0.375 in A=(thickness)(height)=.375*.9055=.339in^2

τ_{xy} =(3*100 lbf)/(2*0.339 in^2)=442.47 psi

4. Hooke's Law

$$\begin{bmatrix}\epsilon_{xx}\\ \epsilon_{yy}\\ \epsilon_{xy}\end{bmatrix} = \frac{1}{E}\begin{bmatrix}1 & -v & 0\\ -v & 1 & 0\\ 0 & 0 & 1+v\end{bmatrix}\begin{bmatrix}\sigma_{xx}\\ \sigma_{yy}\\ \tau_{xy}\end{bmatrix}$$

$$\mathbf{E = 10{,}000\ ksi}$$
$$\mathbf{v = .33}$$

$$\begin{bmatrix}\sigma_{xx}\\ \sigma_{yy}\\ \tau_{xy}\end{bmatrix} = \begin{bmatrix}13055.59\\ 0\\ 442.47\end{bmatrix}$$

$$\frac{1}{E}\begin{bmatrix}1 & -v & 0\\ -v & 1 & 0\\ 0 & 0 & 1+v\end{bmatrix} = 1\times10^{-6}\begin{bmatrix}.1 & -.033 & 0\\ -.033 & .1 & 0\\ 0 & 0 & .133\end{bmatrix}$$

$$\begin{bmatrix}\epsilon_{xx}\\ \epsilon_{yy}\\ \epsilon_{xy}\end{bmatrix} = 1\times10^{-6}\begin{bmatrix}.1 & -.033 & 0\\ -.033 & .1 & 0\\ 0 & 0 & .133\end{bmatrix}\begin{bmatrix}13055.5\\ 0\\ 442.47\end{bmatrix} = \begin{bmatrix} \\ \\ \end{bmatrix}\mu\epsilon$$

ϵ_{xx}=1305.559με, ϵ_{yy}=-4308.34με, ϵ_{zz}=58.2098με

<u>CHAPTER</u> <u>4</u>

DESIGNING

IN

CATIA AND ANSYS DESIGN MODELER

CHAPTER 4

Designing in CATIA and ANSYS Design Modeler

4.1 INTRODUCTION

Geometric design (**GD**) is a branch of computational geometry. It deals with the construction and representation of free-form curves, surfaces, or volumes and is closely related to geometric modeling. Core problems are curve and surface modelling and representation. GD studies especially the construction and manipulation of curves and surfaces given by a set of points using polynomial, rational, piecewise polynomial, or piecewise rational methods. The most important instruments here are parametric curves and parametric surfaces, such as curves, spline curves and surfaces. An important non-parametric approach is the level set method. Geometric models can be built for objects of any dimension in any geometric space. Both 2D and 3D geometric models are extensively used in computer graphics. 2D models are important in computer typography and technical drawing. 3D models are central to computer-aided design and manufacturing. Geometric problems originating in architecture can lead to interesting research and results in geometry processing, computer-aided geometric design, and discrete differential geometry.

Solid modeling (or modelling) is a consistent set of principles for mathematical and computer modeling of three-dimensional solids. Solid modeling is distinguished from related areas of geometric modeling and computer graphics by its emphasis on physical fidelity. Together, the principles of geometric and solid modeling form the foundation of 3D-computer-aided design and in general support the creation, exchange, visualization, animation, interrogation, and annotation of digital models of physical objects.

4.2 CATIA V5

CATIA offers a solution to shape design, styling, surfacing workflow and visualization to create, modify, and validate complex innovative shapes from industrial design to Class-A surfacing with the ICEM surfacing technologies. CATIA supports multiple stages of product design whether started from scratch or from 2D sketches(blueprints).

CATIA enables the creation of 3D parts, from 2D sketches, Sheetmetal, composites, molded, forged or tooling parts up to the definition of mechanical assemblies. The software provides advanced technologies for mechanical surfacing & BIW. It provides tools to complete product definition, including functional tolerances as well as kinematics definition. CATIA provides a wide range of applications for tooling design, for both generic tooling and mold & die. In the case of Aerospace engineering an additional module named the aerospace Sheetmetal design offers the user combine the capabilities of generative Sheetmetal design and generative surface design.

4.3. ANSYS Design Modeler

The ANSYS Design Modeler create and modify their geometry in preparation for their analysis in ANSYS Workbench, this modeler is intended for both FEA and CFD. All engineering simulation starts with geometry to represent the design, be it a solid component for a structural analysis or the air volume for a fluid or electromagnetic study. The engineer either has geometry that has been produced in a computer-aided design (CAD) system or builds the geometry from scratch. ANSYS Design Modeler software is the gateway to geometry handling for an ANSYS analysis.

4.4 SOLID MODELING

The crank itself was made available to be measured and a CAD model produced to enable a Finite Element Analysis. Dassault's CATIA V5 CAD package was selected for solid modelling stage. This was chosen because it have all the functionality required to produce the model as well as the ability to Export A STEP file which would be needed to allow the model exchange to ANSYS's Workbench Package.

4.4.1 BIKE CRANK ARM MODEL (Ref. CORNELL UNIVERSITY)

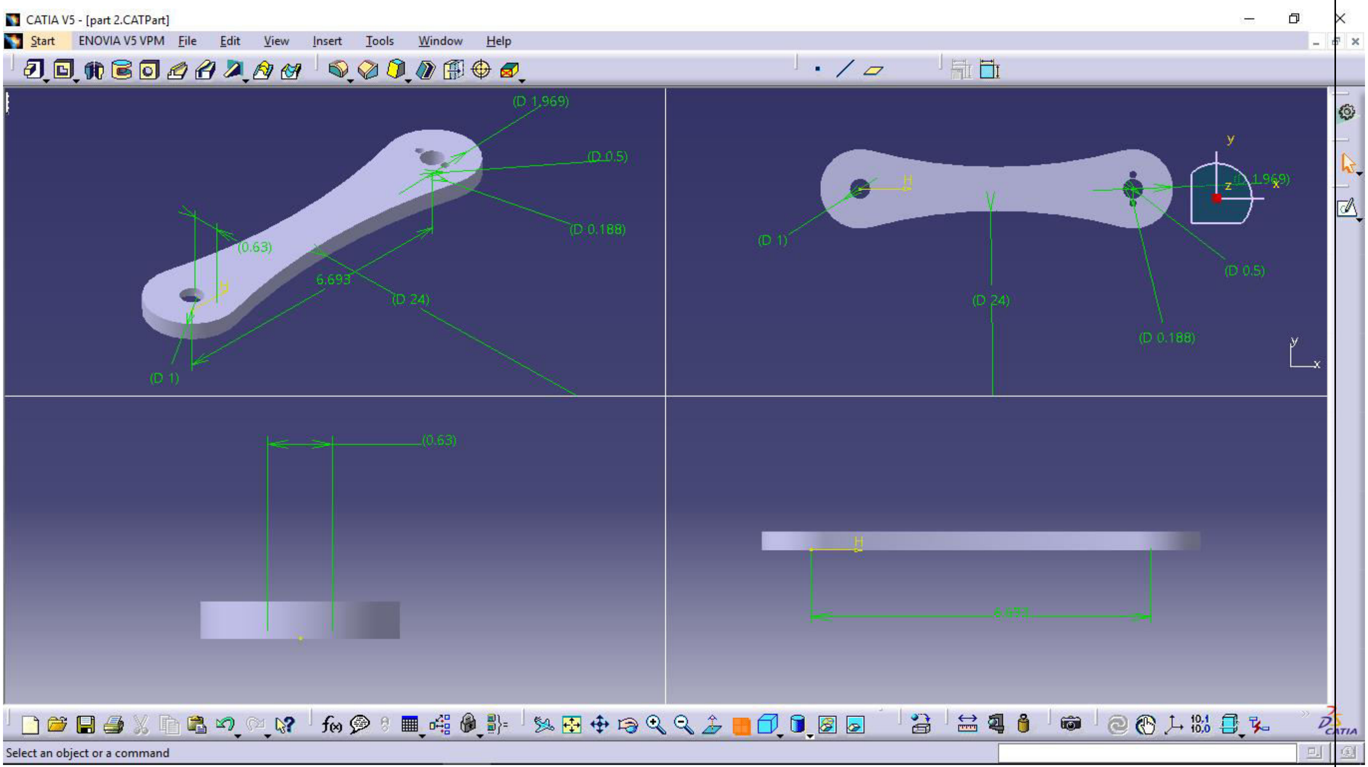


Fig. - Different orientation of model

1. Top-Left Corner ORTHOGONAL VIEW
2. Top-Right Corner TOP VIEW
3. Bottom-Left Corner FRONT VIEW
4. Bottom-Right Corner SIDE VIEW

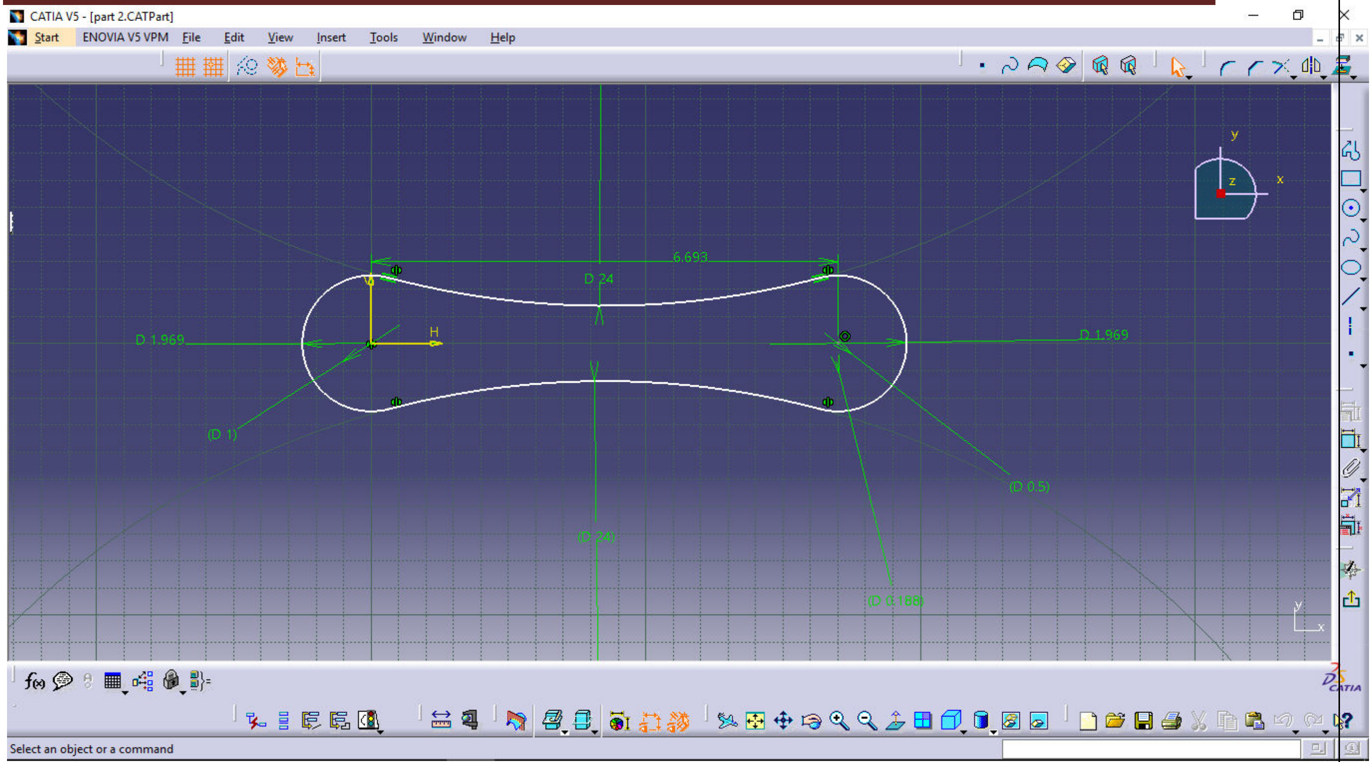
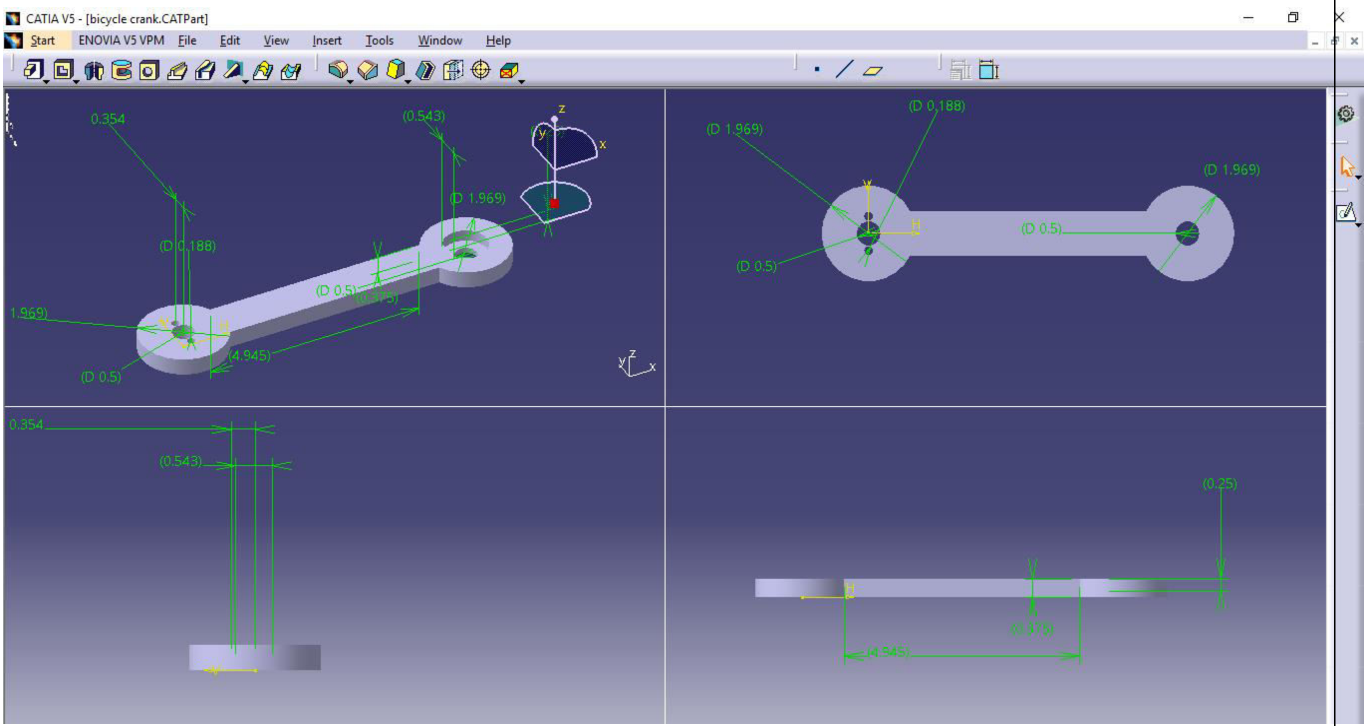


Fig. – Sketch of bike crank arm

4.4.2 BIKE CRANK ARM MODEL (Self Designed Model A)



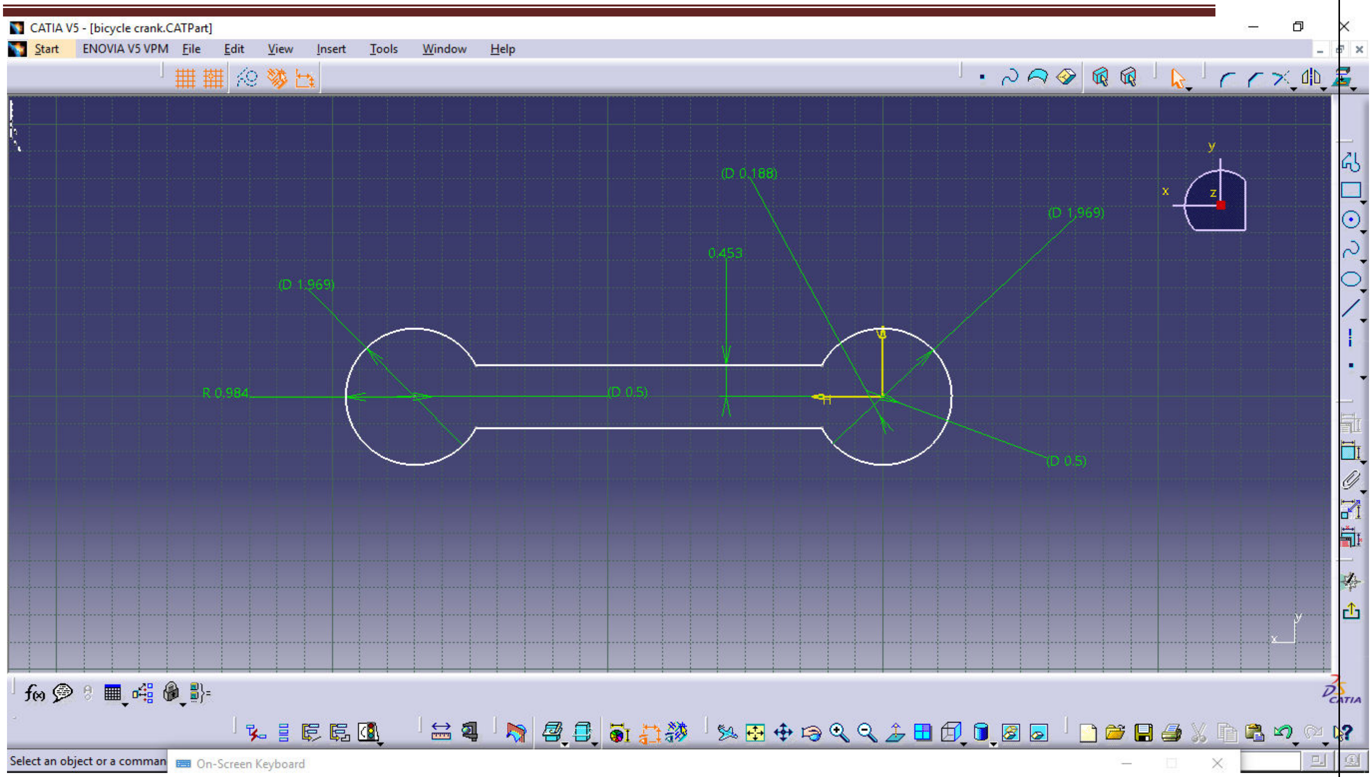
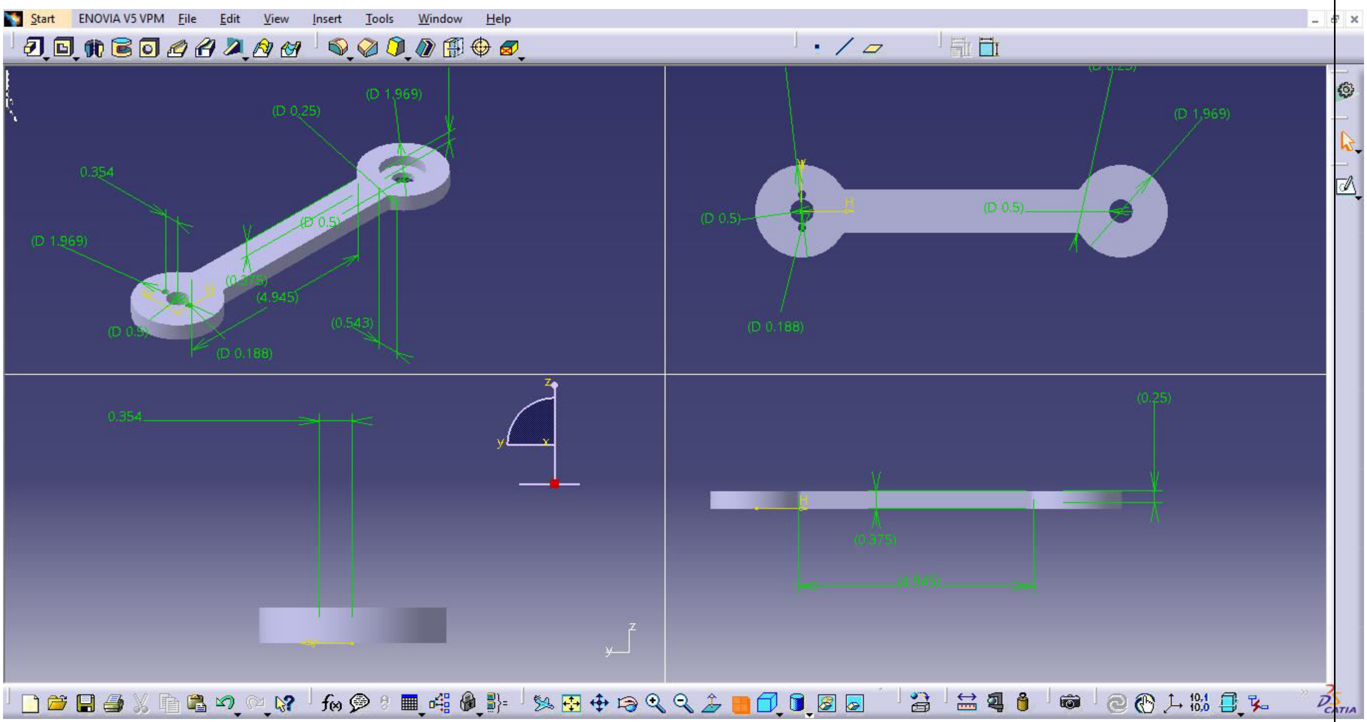


Fig. – Sketch of bike crank arm(Model A)

4.4.3 BIKE CRANK ARM MODEL (Self Designed Model B)



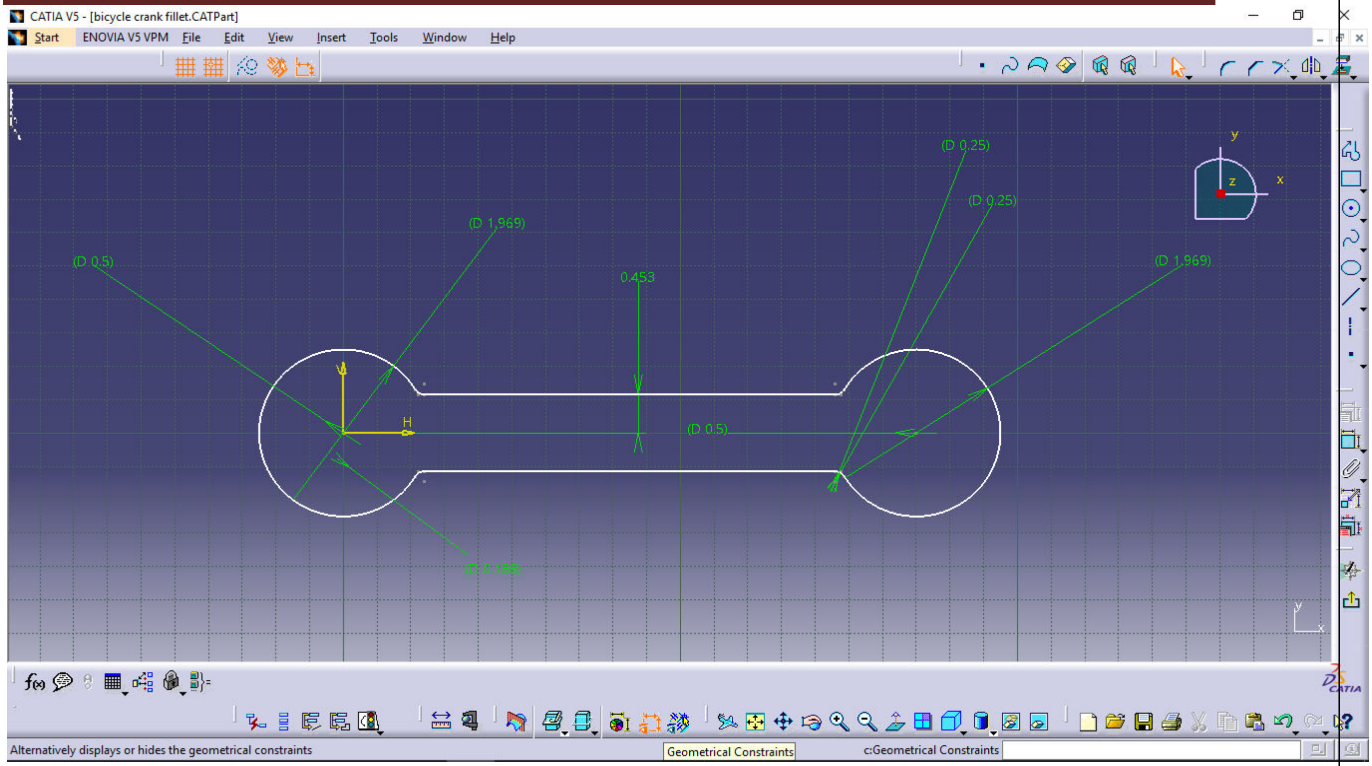


Fig. – Sketch of bike crank arm(Model B)

Model A(Without Fillet)		**Model B**(Fillet R.125 in)	
Parameters	Values	Parameters	Values
Crank arm Length	8.661 in	Crank arm Length	8.661 in
Crank arm Width	1.968 in	Crank arm Width	1.968 in
Crank arm Thickness	0.375 in	Crank arm Thickness	0.375 in
Middle Part Length	4.945 in	Middle Part Length	4.945 in
Inner hole diameter	0.5 in	Inner hole diameter	0.5 in
Support holes diameter	0.188 in	Support holes diameter	0.188 in
Outer arc section ϕ	1.969 in	Outer arc section ϕ	1.969 in
D section edge	0.543 in	D section edge	0.543 in
D section depth	0.25 in	D section depth	0.25 in
Circle inscribing D section ϕ	1 in	Circle inscribing D section ϕ	1 in

MODEL DEVELOPMENT IN CATIA SHOWING BOUNDARY CONDITION

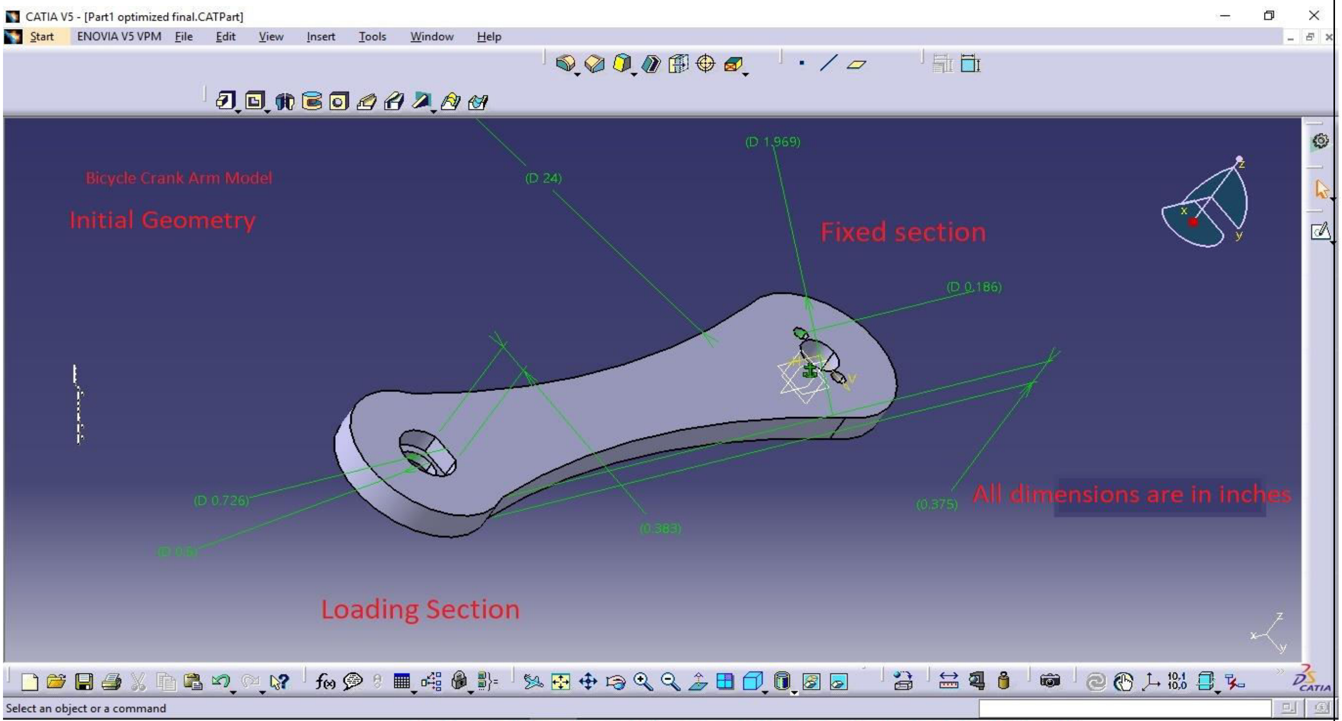


Figure 1. Bicycle crank arm model development in catia V5

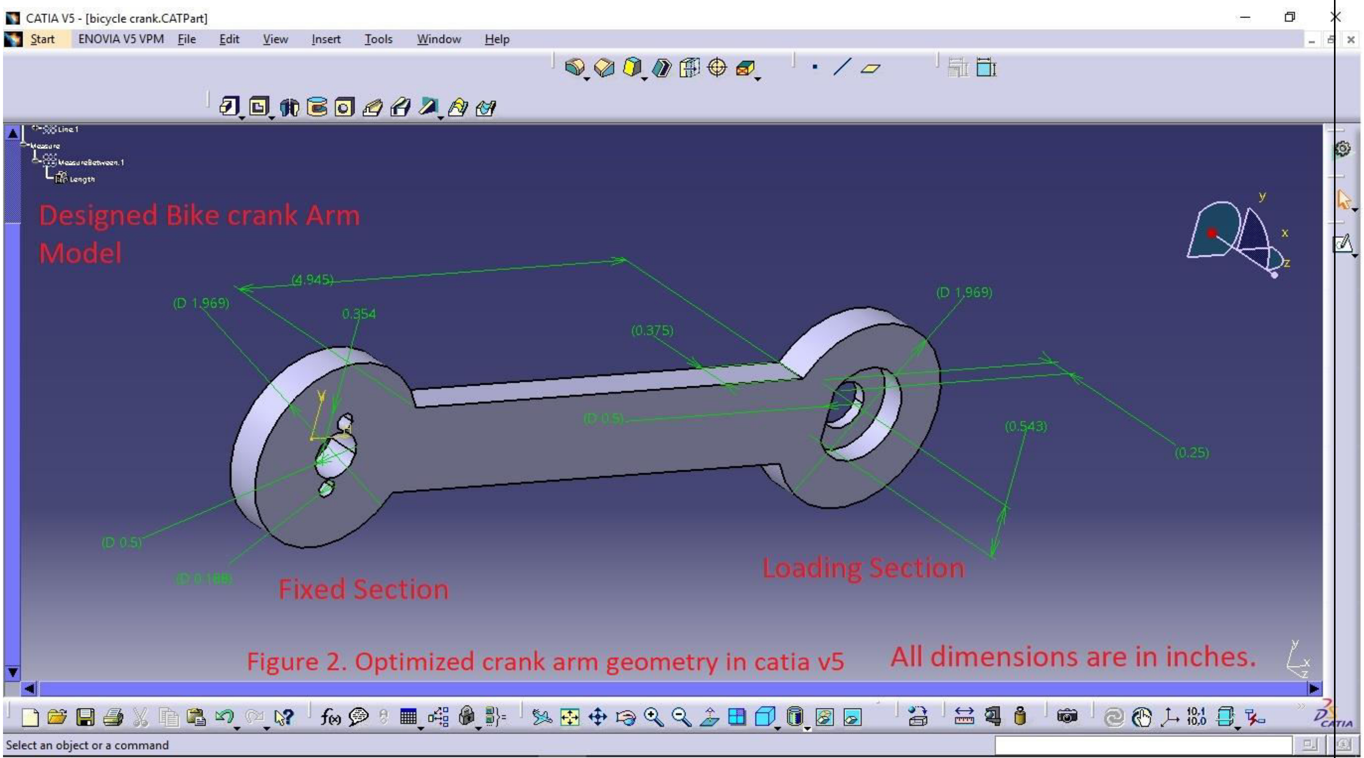


Figure 2. Optimized crank arm geometry in catia v5

CHAPTER 5

STATIC STRUCTURAL ANALYSIS IN ANSYS WORKBENCH

CHAPTER 5

Static Structural Analysis in ANSYS WORKBENCH

5.1 INTRODUCTION

Structural analysis is the determination of the effects of loads on physical structures and their components. Structural analysis employs the fields of applied mechanics, materials science and applied mathematics to compute a structure's deformations, internal forces, stresses, support reactions, accelerations, and stability. The results of the analysis are used to verify a structure's fitness for use, often precluding physical tests. Structural analysis is thus a key part of the engineering design of structures.

A static structural analysis determines the displacements, stresses, strains, and forces in structures or components caused by loads that do not induce significant inertia and damping effects. *Steady* loading and response conditions are assumed; that is, the loads and the structure's response are assumed to vary slowly with respect to time. A static analysis can be either linear or nonlinear.

The kinds of loading that can be applied in a static analysis include:

- Externally applied forces and pressures
- Steady-state inertial forces (such as gravity or rotational velocity)
- Imposed (non-zero) displacements

The procedure for a static analysis consists of three main steps:

1. Build the model.

2. Apply loads and obtain the solution.

3. Review the results.

5.2 MATERIAL ASSIGNMENT

In the mathematical model, the only equation that contains the material property is the Hooke's Law. The specific material properties that appear in the Hooke's law are the Young's modulus and Poisson's ratio.

Material	Minimum safety factor
Aluminium 6061	0.77
Mild Steel	0.54
Carbon Fibre (CFRP)	0.94
Titanium	0.81

Outline of Schematic A2: Engineering Data

	A	B	C	D
1	Contents of Engineering Data		Source	Description
2	Material			
3	AL 6061-T6			"Equation of State and Strength Properties of Selected Materials". Steinberg D.J. LLNL. Feb 1991
4	Epoxy_Carbon_UD_395GPa_Prepreg			
5	Structural Steel			Fatigue Data at zero mean stress comes from 1998 ASME BPV Code, Section 8, Div 2, Table 5-110.1
6	TI 6%AL4%V			"Equation of State and Strength Properties of Selected Materials". Steinberg D.J. LLNL. Feb 1991

1. Aluminium 6061-T6 Alloy

   6061 is a precipitation-hardened aluminium alloy, containing magnesium and silicon (solutionized and artificially aged) Temperate grade - T6. Aluminium cranks may be cast, hot forged or cold forged. Cold forging gives the metal additional strength, and the cranks can therefore be made lighter without increasing the risk of breakage. Also, some aluminium cranks are made by forging the main arms around a hard steel insert which is then withdrawn, leaving an internal void to save weight. They are then welded up before final machining.

Youngs modulus	68.9 GPa
Tensile Strength	310 MPa
Yield Strength	275 MPa

Properties of Outline Row 3: AL 6061-T6

	A	B	C
1	Property	Value	Unit
2	Density	2703	kg m^-3
3	Isotropic Elasticity		
4	Derive from	Young's Modulus and Poisson's Ratio	
5	Young's Modulus	1E+07	psi
6	Poisson's Ratio	0.33	
7	Bulk Modulus	6.7596E+10	Pa
8	Shear Modulus	2.592E+10	Pa
9	Field Variables		
10	Temperature	Yes	
11	Shear Angle	No	
12	Degradation Factor	No	
13	Tensile Yield Strength	40000	psi
14	Tensile Ultimate Strength	45000	psi

2. Titanium (TI 6%Al 4%V)

A denser material which is costlier and difficult to machine.

an alpha-beta titanium alloy featuring high strength, low weight ratio and excellent corrosion resistance.

Properties of Outline Row 6: TI 6%AL4%V

	A	B	C
1	Property	Value	Unit
2	Density	4419	kg m^-3
3	Isotropic Elasticity		
4	Derive from	Young's Modulus and Poisson's Ratio	
5	Young's Modulus	1.65E+07	psi
6	Poisson's Ratio	0.342	
7	Bulk Modulus	1.2E+11	Pa
8	Shear Modulus	4.2386E+10	Pa
9	Field Variables		
10	Temperature	Yes	
11	Shear Angle	No	
12	Degradation Factor	No	
13	Tensile Yield Strength	1.28E+05	psi
14	Tensile Ultimate Strength	1.38E+05	psi

3. Carbon fibre

Carbon fibre is the best material choice for this part however there are two main reasons why carbon fibre was not chosen.

- Non-Linear material
- Very brittle material

Properties of Outline Row 4: Epoxy_Carbon_UD_395GPa_Prepreg

	A	B	C
1	Property	Value	Unit
2	Density	1540	kg m^-3
3	Orthotropic Elasticity		
4	Young's Modulus X direction	2.09E+05	MPa
5	Young's Modulus Y direction	9450	MPa
6	Young's Modulus Z direction	9450	MPa
7	Poisson's Ratio XY	0.27	
8	Poisson's Ratio YZ	0.4	
9	Poisson's Ratio XZ	0.27	
10	Shear Modulus XY	5500	MPa
11	Shear Modulus YZ	3900	MPa
12	Shear Modulus XZ	5500	MPa
13	Field Variables		
17	Orthotropic Stress Limits		
18	Tensile X direction	1979	MPa
19	Tensile Y direction	26	MPa
20	Tensile Z direction	26	MPa
21	Compressive X direction	-893	MPa
22	Compressive Y direction	-139	MPa
23	Compressive Z direction	-139	MPa
24	Shear XY	100	MPa
25	Shear YZ	50	MPa
26	Shear XZ	100	MPa

4. Structural steel

Properties of Outline Row 5: Structural Steel

	A	B	C
1	Property	Value	Unit
2	Density	7850	kg m^-3
3	Isotropic Elasticity		
4	Derive from	Young's Modulus and Poisson's Ratio	
5	Young's Modulus	2E+05	MPa
6	Poisson's Ratio	0.3	
7	Bulk Modulus	1.6667E+11	Pa
8	Shear Modulus	7.6923E+10	Pa
9	Field Variables		
10	Temperature	Yes	
11	Shear Angle	No	
12	Degradation Factor	No	
13	Alternating Stress Mean Stress	Tabular	
17	Strain-Life Parameters		
25	Tensile Yield Strength	250	MPa
26	Compressive Yield Strength	250	MPa
27	Tensile Ultimate Strength	460	MPa
28	Compressive Ultimate Strength	0	Pa

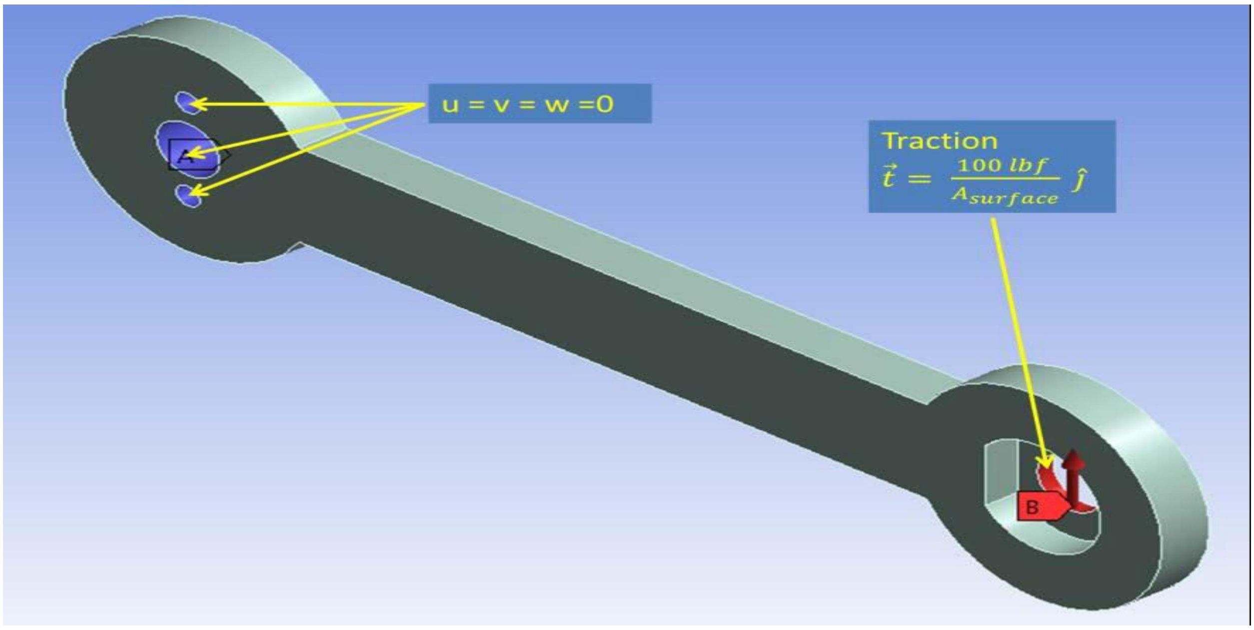
STRUCTURAL ANALYSIS OF CRANK ARM

Table: Material Properties of Different Materials used in crank arm

PROPERTY \ MATERIAL	Al 6061-T6	TI (6%Al 4%V)	Epoxy carbon (CFRP)-x	Structural steel
Young modulus	68.9GPa	113.76GPa	209GPa	200GPa
Poisson's ratio	0.33	0.342	0.27	0.3
Density	2.703g/cm^3	4.419 g/cm^3	1.54 g/cm^3	7.850 g/cm^3
Shear Modulus	26 GPa	42.683 GPa	5.5GPa	76.92GPa
Tensile Strength, Ultimate	310 MPa	951.48 MPa	1979MPa	250MPa

5.3 BOUNDARY CONDITION

- Essential boundary conditions are applied at the three holes on the crank where displacements are set to zero as shown in the figure below.
- Natural boundary conditions are applied at all other boundary surfaces. The traction is non-zero for the surface at which the force is applied (see figure below). It is zero for the remaining surfaces.

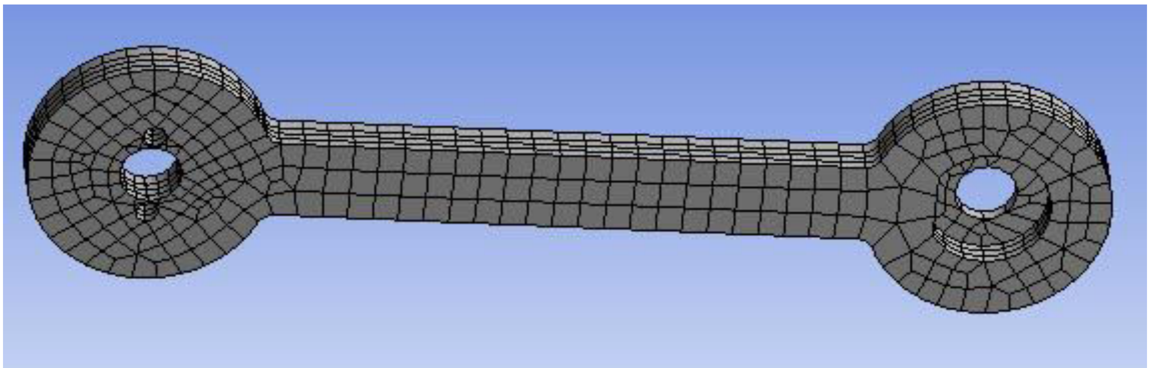


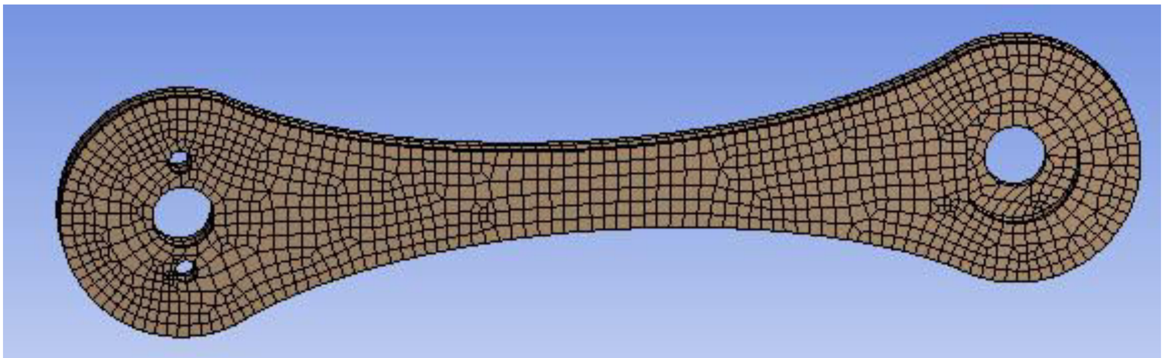
5.4 MESH AND MODAL SETUP

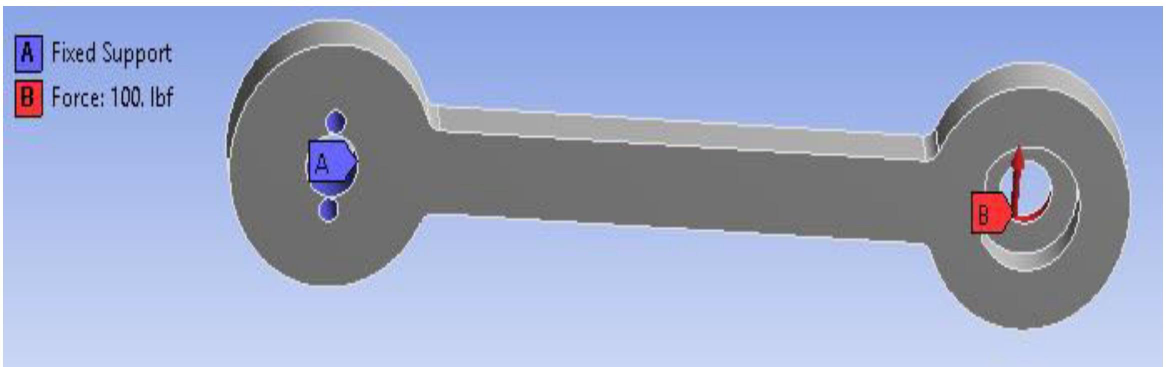
We have created a mesh over our geometry by following procedure:

Initially we came across the default coarse mesh which has to be refined. In mesh controls, we have set the value of body sizing to 0.2 in for whole crank, and redefined around the fixed holes to 0.1 in. Changing the method from default to multizone method to get hexahedral element which gives few nodes as at one node we have three displacement component to determine.

Statistically our mesh has 6852 nodes and 1226 hexahedra elements so we are solving for a total of around 20556 displacement values to be determined directly by ANSYS solver.



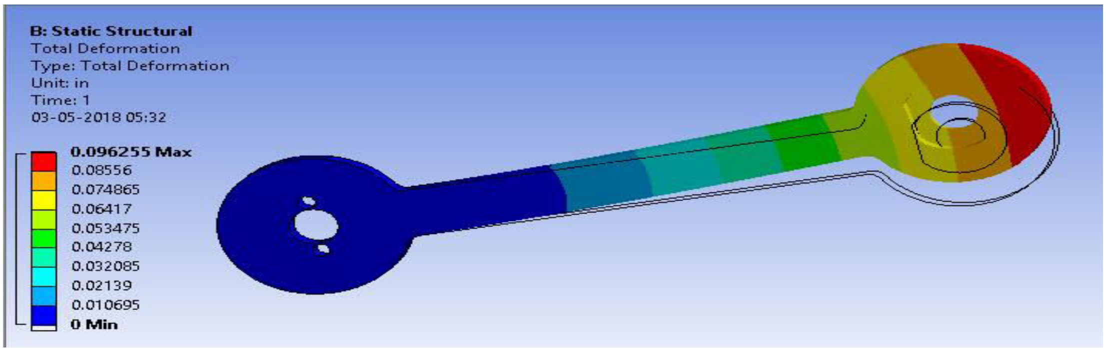


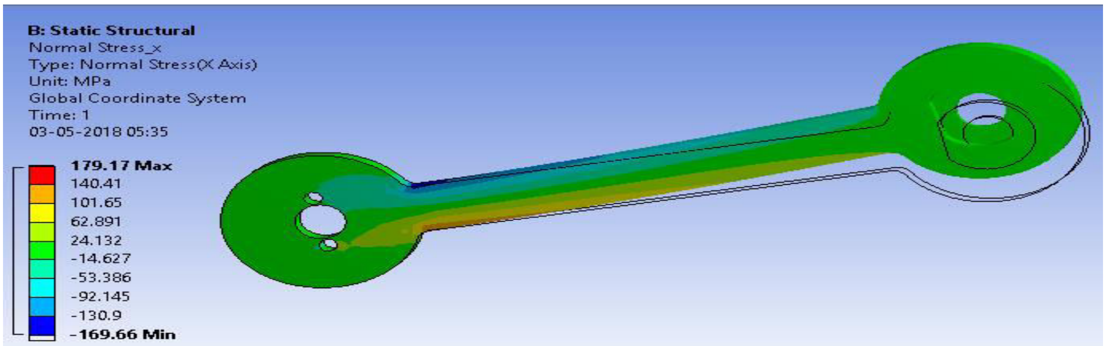


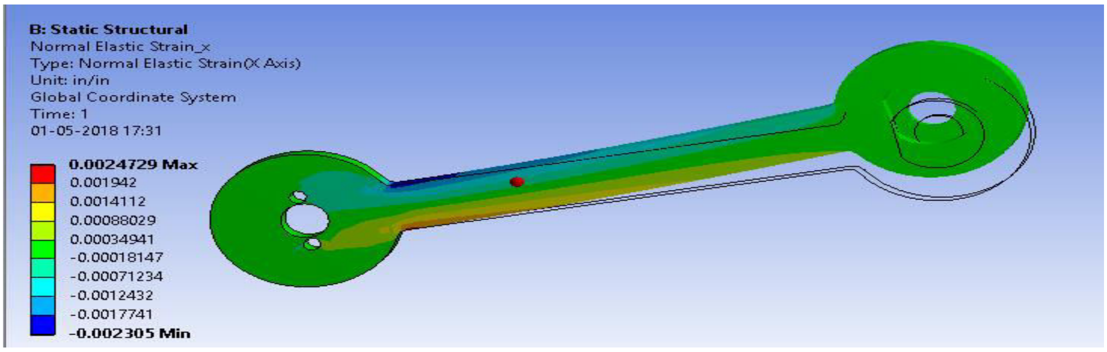
5.5 NUMERICAL RESULTS

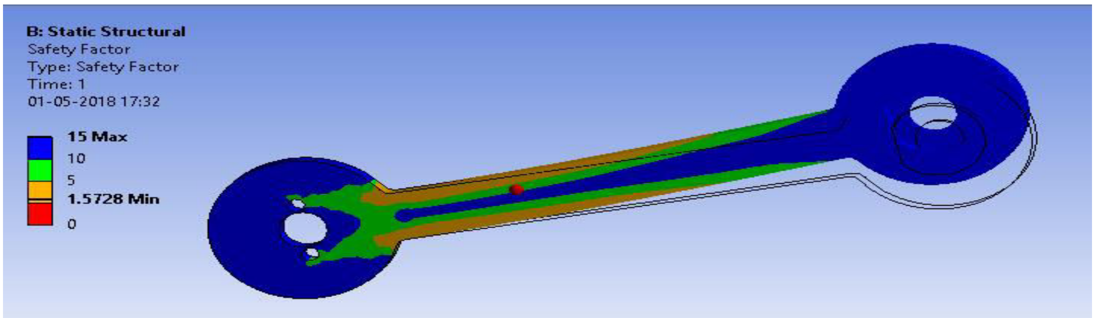
MODEL -B (FILLETS AT EIGHT CORNERS OF MODEL-A)

Displacement and Stress on Crank Arm of Al6061

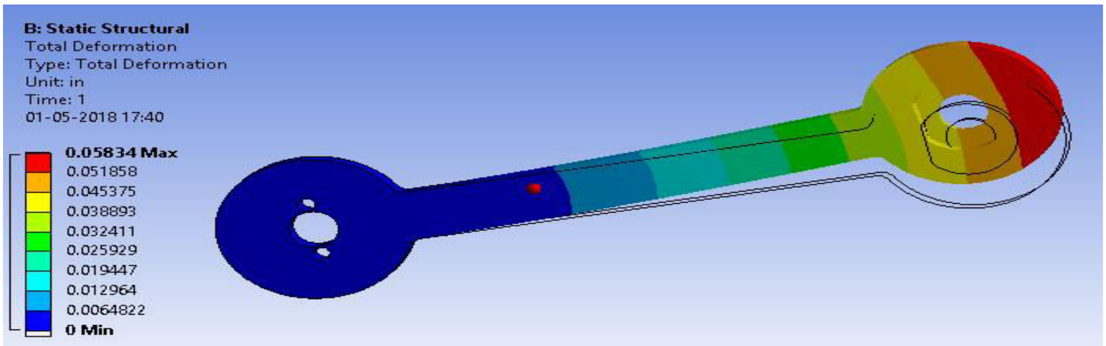


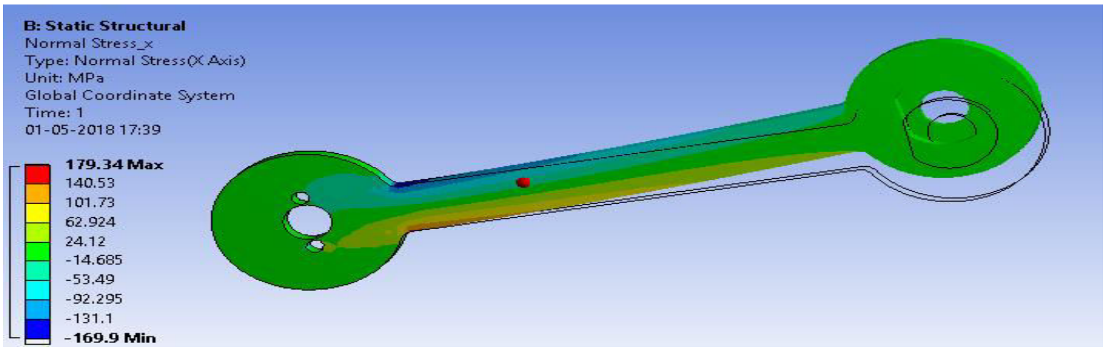


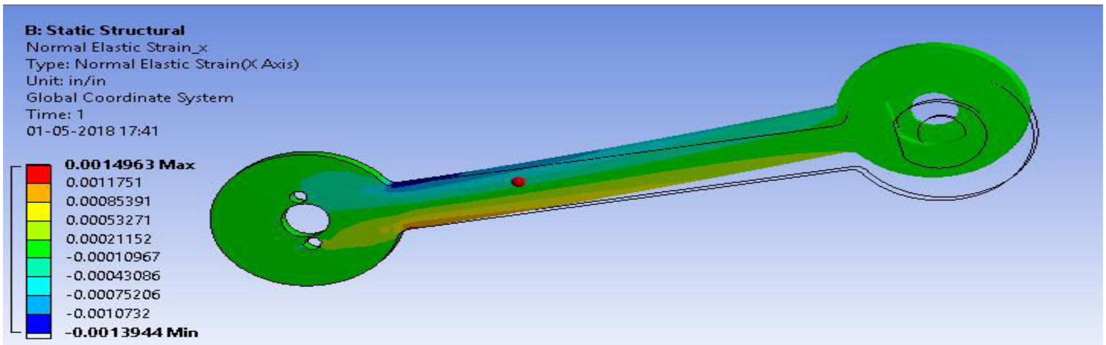


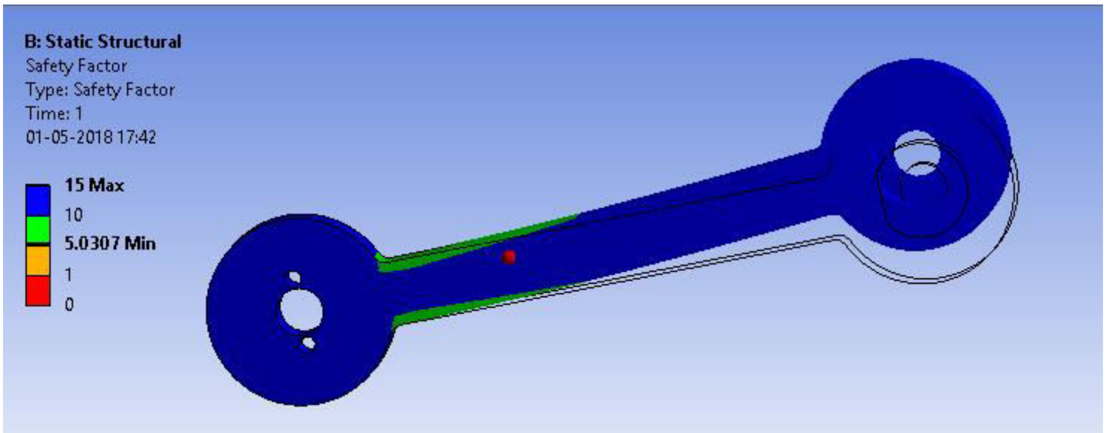


Displacement and Stress on Crank Arm of Ti

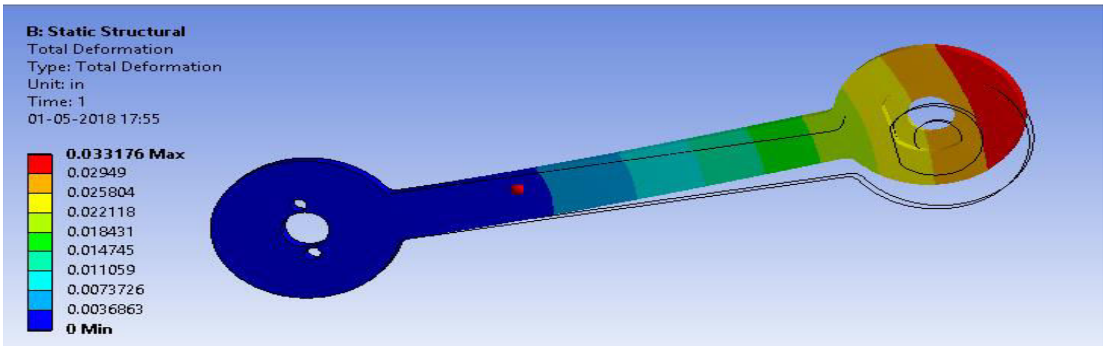


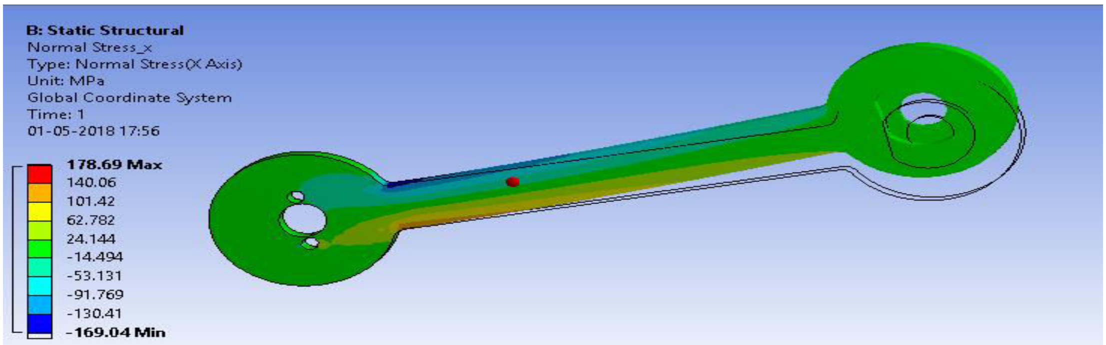


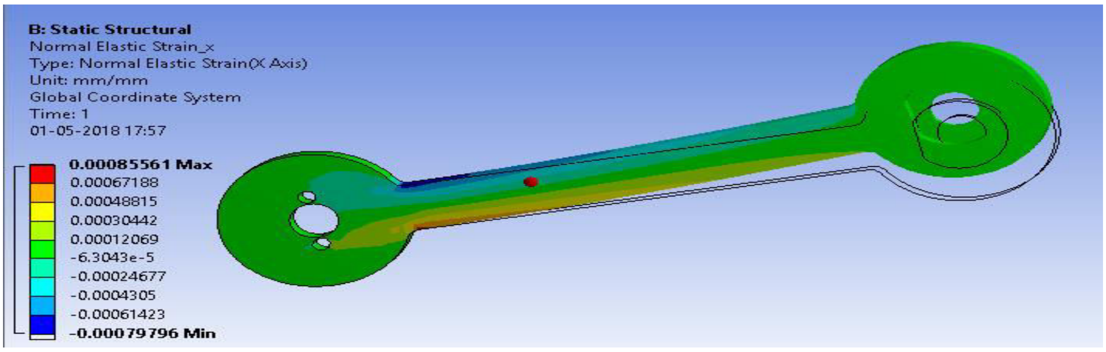


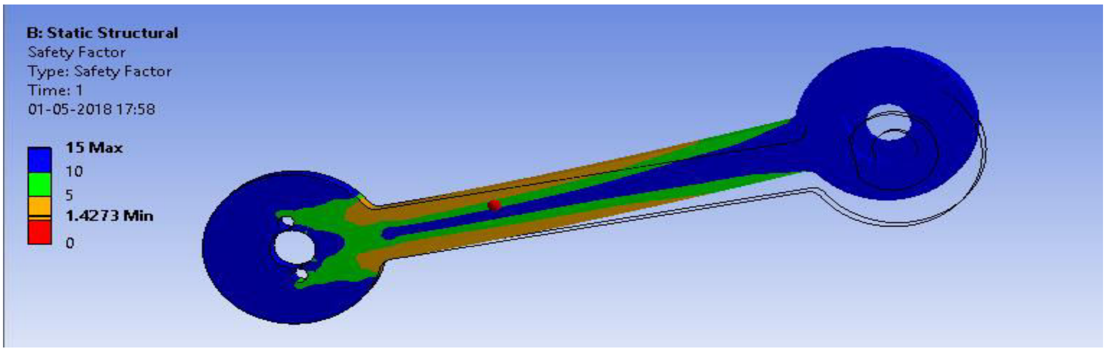


Displacement and Stress on Crank Arm Of SS

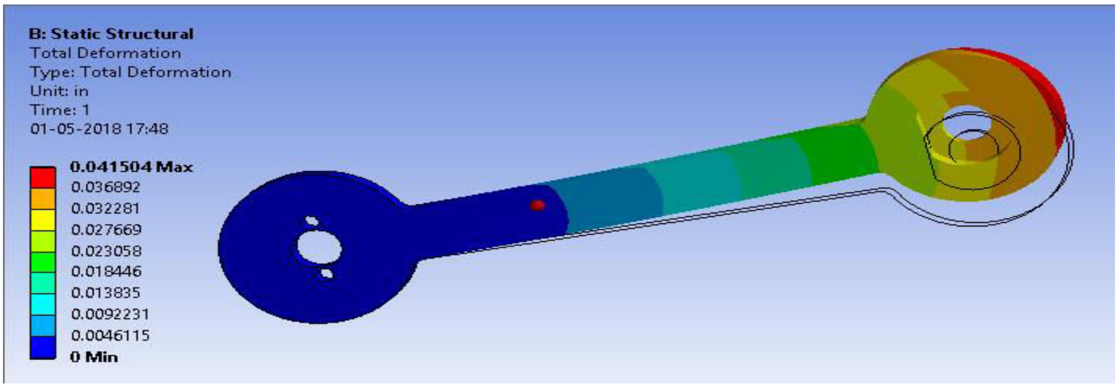


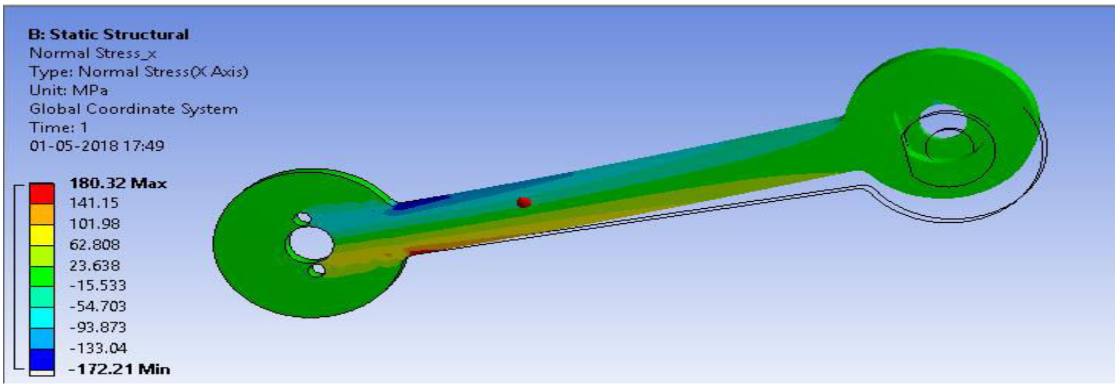


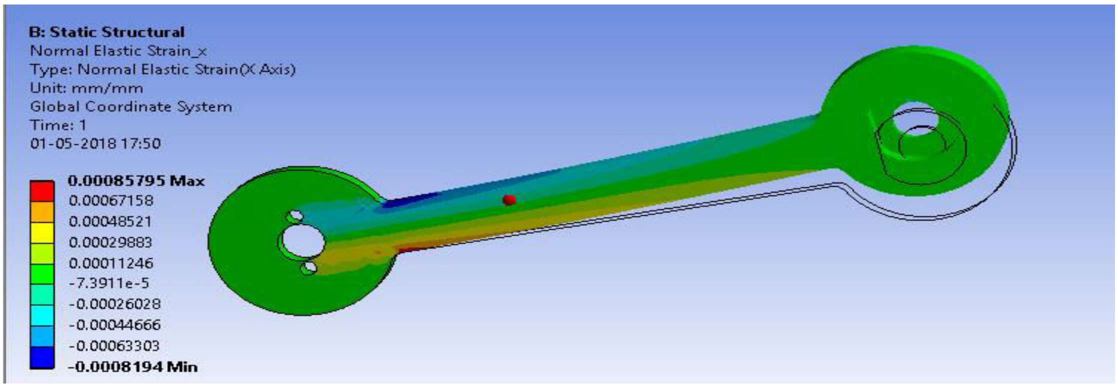




Displacement and Stress on Crank Arm Of Carbon fiber







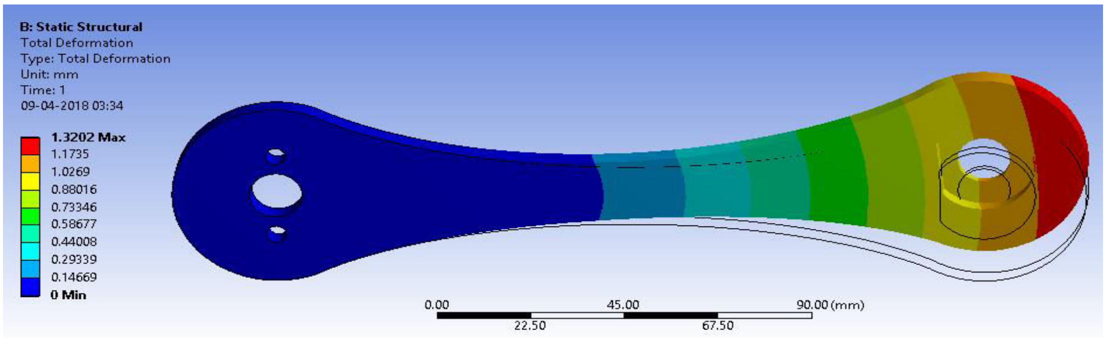
NOTE : The forces where applied on the pedal cavity circular face and the effect of it on the crank arm was studied in this analysis.

Table of Results for Stress and Strain

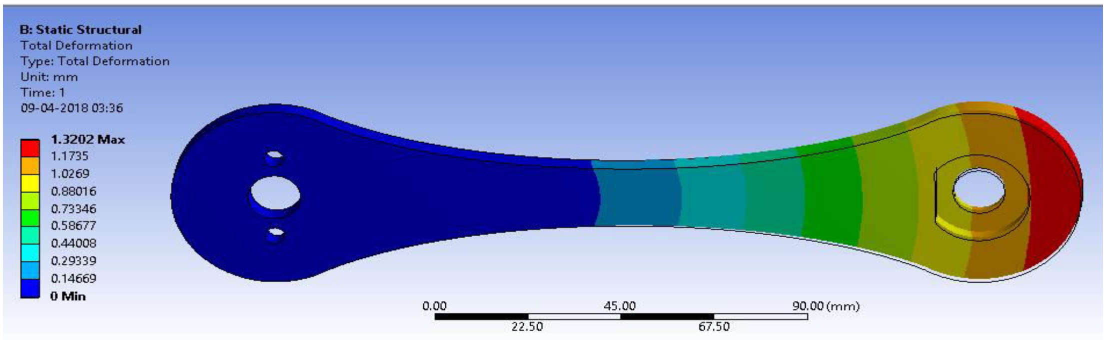
	Al 6061-T6	TI	SS	Carbon Fiber
Displacement (mm)	0.0962in	0.05834in	0.033176in	0.0415in
Stress (N/mm2)	179.17MPa	179.34MPa	178.69MPa	180.32MPa
Strain	0.0024	0.00149	0.00085	0.00085
Factor of Safety	1.57	5.03	1.42	
Weight (Kg)	0.14078kg	0.23016kg	0.4088kg	0.08021kg
Volume(mm^3)	52084	52084	52084	0.00005208

Imported model (ref . Cornell university)

Deformation - Total deformation = 1.32 mm (0.05 in)

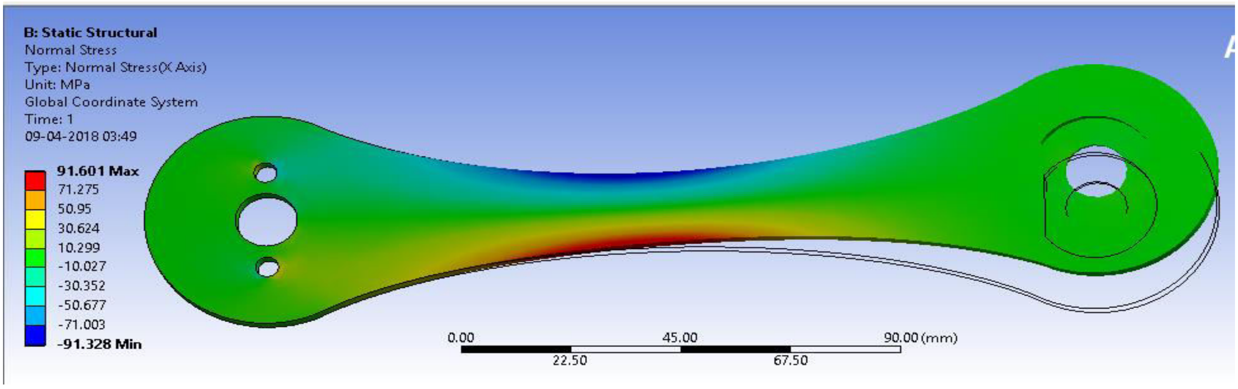


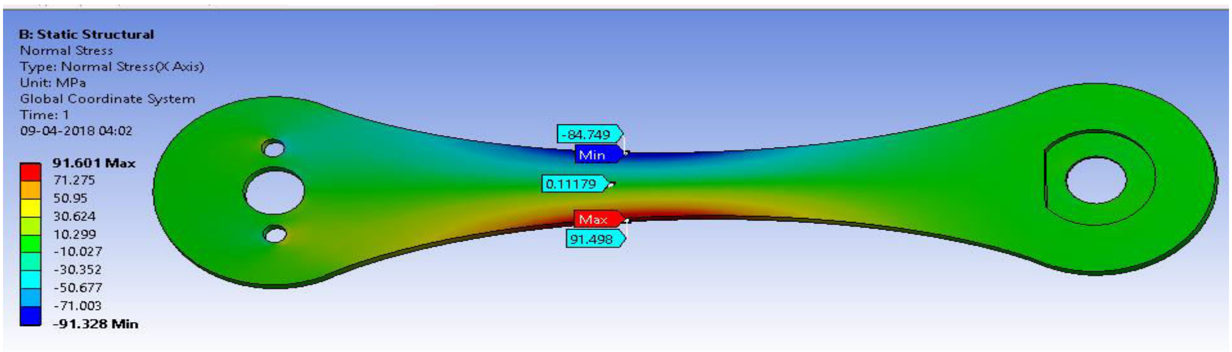
Result is Scaled by default.



True scale: the deformation is much smaller.

<u>Normal stress-</u> Maximum nominal stress = 91.601 MPa (13286 psi)

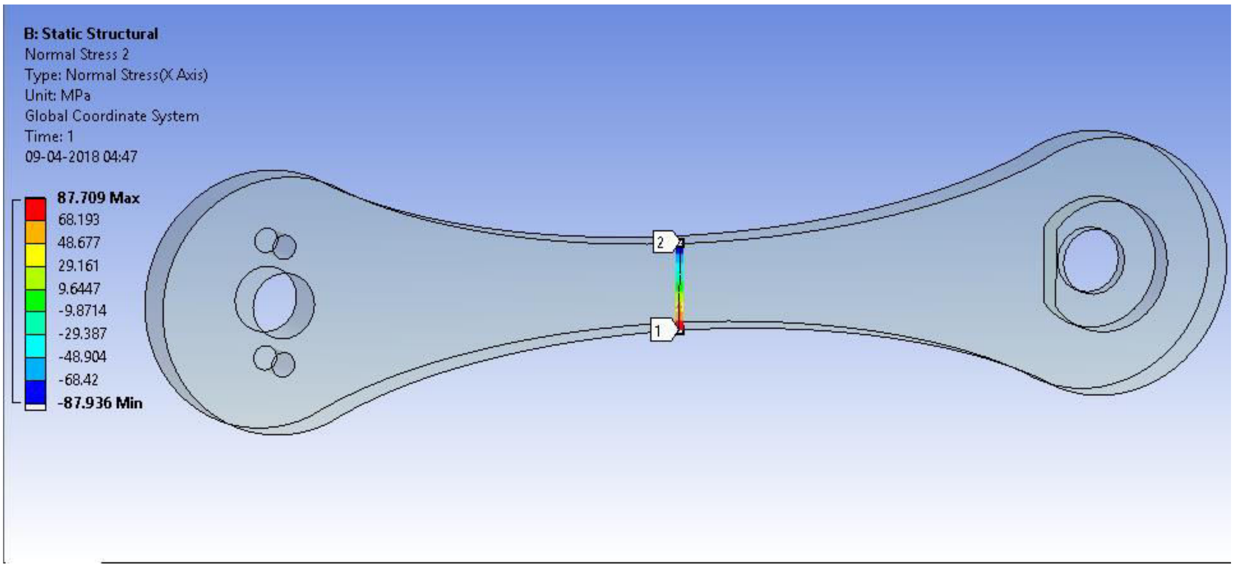




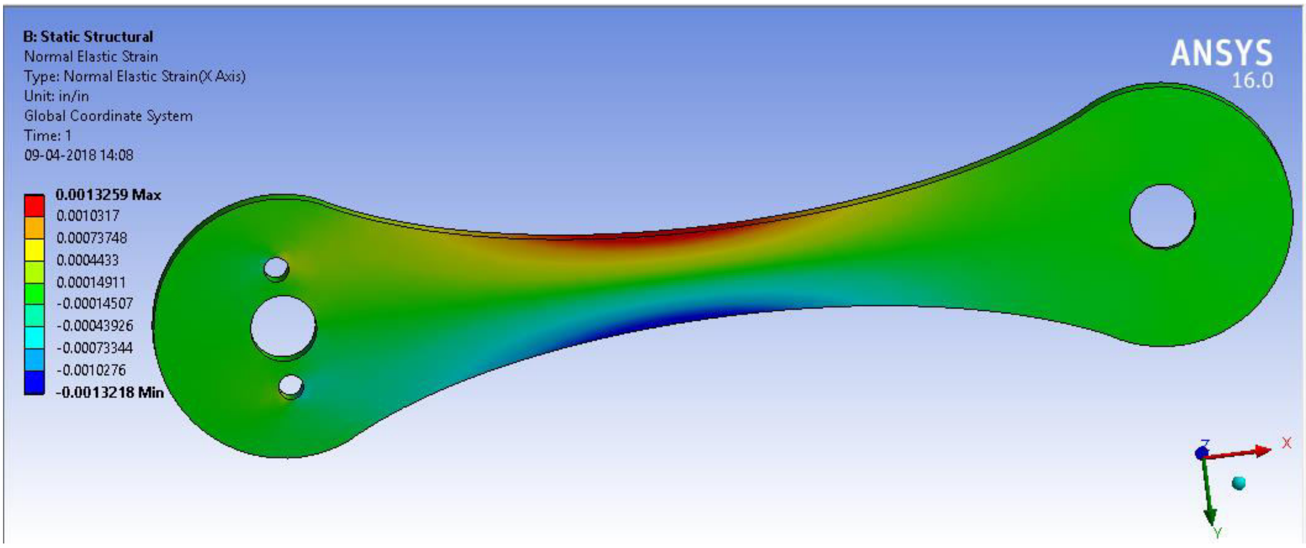
Sigma_x goes from high tension to high compression in the middle of the crank.

(Min tension =-84.749 MPa; Neutral axis = 0.111 MPa; Max tension = 91.498 MPa)

<u>Path Plot of Normal Stress</u>

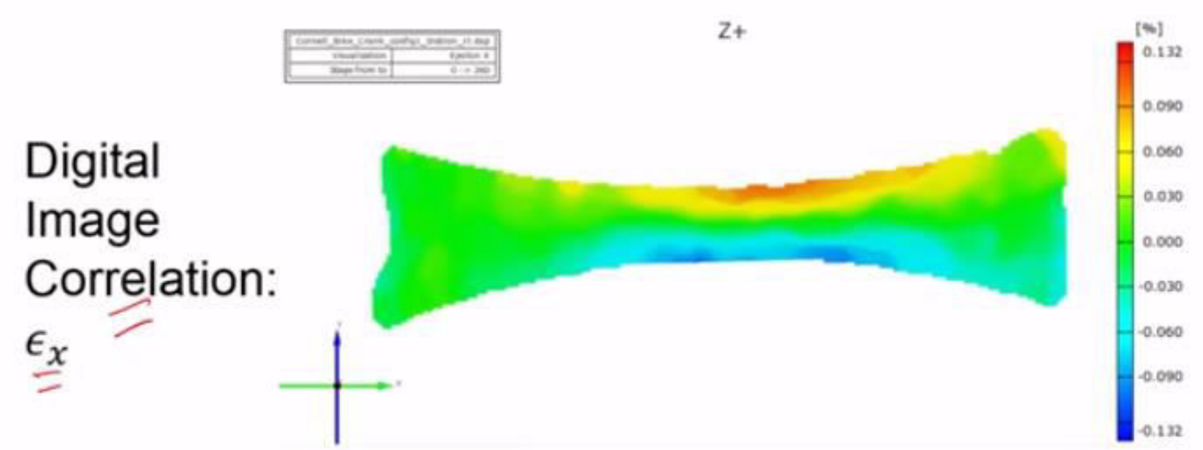


VALIDATION (based on Normal elastic strain)



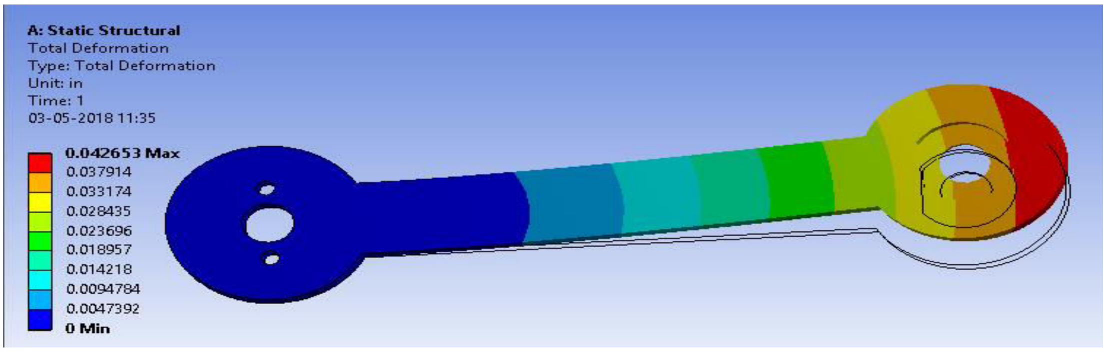
DIGITAL IMAGE CORELATION

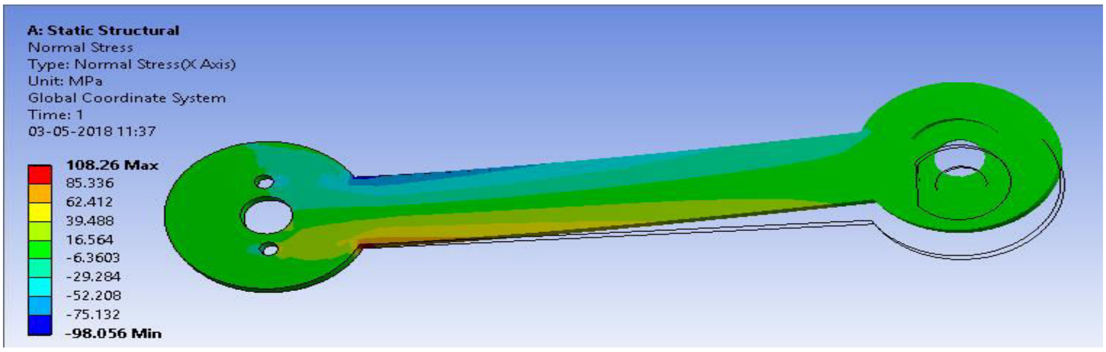
Digital Image Correlation (DIC) is a full-field image analysis method, based on grey value digital images, that can determine the contour and the displacements of an object under load in three dimensions. Due to rapid new developments in high resolution digital cameras for static as well as dynamic applications, and computer technology, the applications for this measurement method has broadened and DIC techniques have proven to be a flexible and useful tool for deformation analysis.

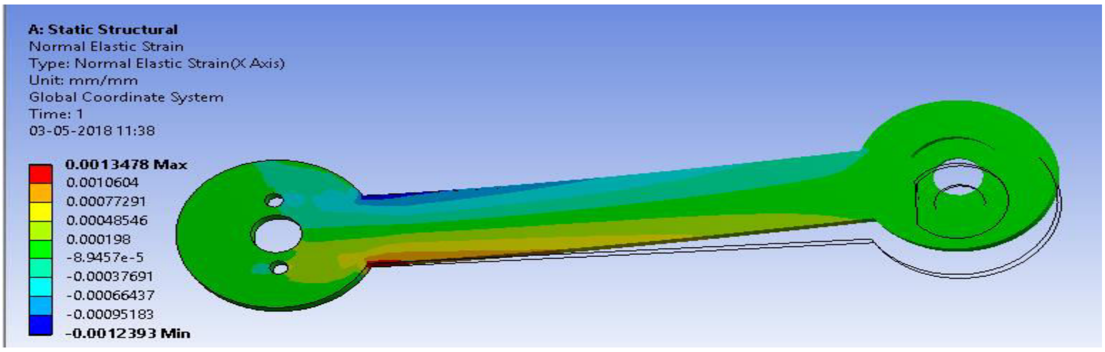


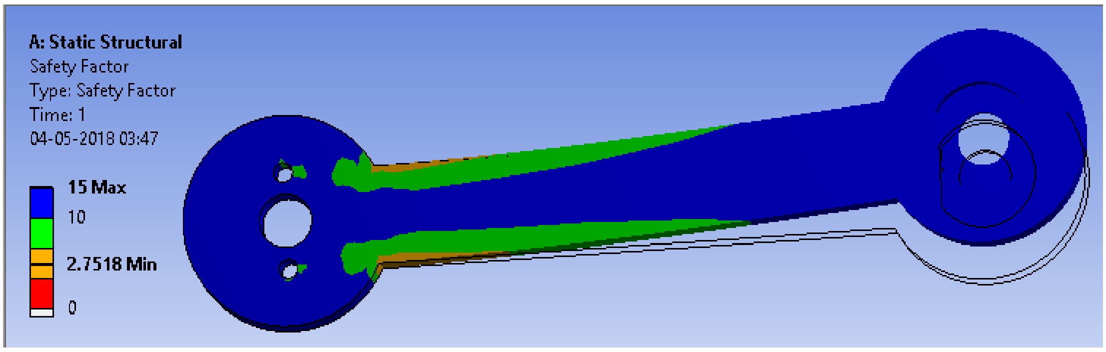
Result courtesy Datapoint labs

Designed model A









Material Al 6061-T6

The percentage change in maximum stress is greater than the percentage change in maximum deformation.

Solutions(Maximum)	Geometry 1 (CU)	Geometry 2(model A)
Total Deformation(δ)	0.05 in	0.0426 in
Normal Stress (σx)	13286 psi	15702 psi
Solutions(Minimum)	Geometry 1 (CU)	Geometry 2(model A)
Total Deformation(δ)	0	0
Normal Stress (σx)	-13291 psi	-14222 psi

Table of Results for Stress and Strain

Imported Model	Aluminum 6061	Model A	Aluminum 6061
Displacement (in)	0.05 in	Displacement (in)	0.0426in
Stress (MPa)	91.601 MPa	Stress (MPa)	108.26MPa
Strain	0.00132	Strain	0.00134
Factor of safety	3.37	Factor of safety	2.75
Weight (g)	160.01g	Weight (g)	159.69g
Volume(mm^3)	59199mm^3	Volume(mm^3)	59080mm^3

Analysis has been done on ANSYS Workbench R.19 student version Academic Edition on bike crank arm with the use of material optimization and force analysis. All in all three geometric model were analyzed which were developed by modelling software CATIA V5 to create solid geometry.

Meshing was refined to get more accurate results and element were chosen to fill the maximum volume by finite elements using optimization.

CHAPTER 6

RESULTS AND DISCUSSION

CHAPTER 6

Result and Discussion

6.1 INTRODUCTION

From the analysis result generated from ANSYS

For the force of 100lbf and for same material Aluminum 6061-T6 our designed model have more bending strength than the reference model along with weight of model is also reduced from 160.014g to 159.69g however stress is increased from 91.601MPa to 108.26MPa but this is not a problem at all.

For Aluminum 6061-T6 value of Ultimate tensile strength is 310MPa and for manufacturing purposes Factor of safety is taken above 1.5 and in both the cases value of factor of safety we are getting is more than 1.5 so that our designed model is better than reference model in all the aspects.

FOS = Ultimate tensile strength / Maximum stress

Reference Model

FOS = 310 / 91.601

= 3.34

Designed Model

FOS = 310 / 108.06

= 2.86

6.2 Results

From the analysis by ANSYS for force 100 lbf and for same material Al 6061-T6

Parameters	**Reference Model**	**Designed Model -A**
Displacement (max)	0.05 in	0.0426 in
Stress (max)	91.601 MPa	108.26 MPa
Weight	160.014 g	159.69 g

6.3 Discussion

- The percentage change in maximum stress is greater than the percentage change in maximum deformation
- Stress is more sensitive than displacement to the mesh and has a larger relative error. Stress are calculated from these nodal values through differentiation which typically magnifies numerical errors.
- The essential boundary conditions set the displacement values directly and are unaffected by material properties.
- Variation in the values of deformation of mathematical model and finite element analysis model as one is based on Beam theory and another on a set of 3-D Equlibrium equation + Boundary Condition + Hooke's Law + Strain Displacement Equation.
- The present work was an attempt to know about the mechanics of material under static condition.

CHAPTER 7

CONCLUSIONS AND FUTURE SCOPE

CHAPTER 7

Conclusions and Future scope

7.1 Conclusion

In conclusion, the report suggest that the majority of results obtained can be trusted. The numerical analysis revealed results that could be confirmed by finite element analysis when these figures were followed through the FEA could be confirmed using the results from the theoretical experimental procedure.

As a part of project work, we have created 2D and 3D model of existing Bicycle Crank component using design tools. Calculated Peddling Force at bicycle rider at uphill position and applied boundary condition on crank component. FEA results shows stresses are well within the yield stress, and deformation is much less. So, there is a scope for optimization. We will further do optimization by removing material by slots from bicycle crank.

In future it's recommended, were the analysis to be carried out again, that the numerical analysis take into account the modified geometrical dimensions, and it's suggested that the finite element analysis be based on the combination of 3D scanning techniques confirmed by measurement.in addition to these, the analysis could be done on number of occasion by slightly altering mesh to establish average values. Any major variation in these values would suggest faults in crank and cause sliding in its fixing and any external deflections.

Fatigue is the progressive structural damage that occurs when materials are subjected to cyclic loading. Stress due to load on the crank was increased to maximum and decreasing to minimum. Equivalent stress should be reduced and need to keep it in an average value for durability.

7.2 FUTURE SCOPE

The above-described model has been tested successfully in ANSYS simulation, so there lies the opportunity to implement the above-described model in hardware and study the impact of the approach taken in this thesis report. Moreover, in this report we have analyzed only the impact of the approach on independent bike crank so there lies the scope to extend the study to various model of bike crank. Also, here we have done the safety factor calculation so analysis can also be extended to study the dynamics for above the rated model using material optimization.

7.3 REFERENCES

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