FEA AND DESIGN MODIFICATION OF SHREDDER BLADE USED FOR RECYCLING PLASTIC

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ABSTRACT

A paper shredder is considered to be a mechanical machine that is used to cut paper into either strips or fine particles that portray no information that was written on the paper initially. Government establishments, big business holders, and private personalities use these shredders to abolish private, personal, or otherwise delicate and secret pamphlets. Privacy specialists frequently claim that individuals should shred their bills, tax papers, credit cards, and statements of their bank account, and other kinds of stuff that might be used by robs to commit deceit and theft. In this report, we are going to design a shredder blade and perform a FEA Static Structural analysis followed by topology optimization and Material comparison for low weight, durability, strength, span life and other stress factors, so that the blade can handle cutting force applied to it.

Keyword - Shredder Blade, FEA Static Structural Analysis, Topology Optimization, Material Comparison

1. INTRODUCTION

This research work is aimed at solving the problems of plastic wastes management in developing countries. In this study, we designed and constructed a plastic shredding machine. The machine consists of the following main components; hopper assembly, shredding chamber, drive shaft, frame, V-belts, and an electric motor. Although the form of the plastics was vastly different to their equivalent when it comes to mass recycling over large sizes, the energy difference highlights the potential environmental benefit of utilizing re-cyclites plastics where it can be pointed out that shredding machine is a feasible operation for recycling purposes.

Humans have always produced trash and disposed of it in some way so solid waste management is not a new issue. What have changed are the types and amounts of waste produced, the methods of disposal, and the human values and perceptions of what should be done with it. The applications of plastic materials and their composites are still growing rapidly due to their low cost and ease of manufacture. Therefore, high amount of waste plastic being accumulated which create big challenges for their disposal.

This research is motivated by concerns about rising global composite waste. Despite the developing composite recycling technology, environmental aspects, particularly the process energy demand of the recycling methods, has not been thoroughly addressed. This research aims to model energy demand of composite recycling processes while considering the quality and characteristics of the re-cyclites. The outcomes are to establish efficient use of energy demand and to enable assessment of a circular economy for composite materials

1.1 Aim and Purpose

A shredding machine Blade is designed to reduce large solid material objects into a smaller volume, or smaller pieces. Shredding machines are usually used to reduce the size and shape of materials so they can be efficiently used for the purpose intended to. Shredding just like crushing can be defined as the process of transferring a force amplified by mechanical advantage through a material made of molecules that bond together more strongly, and resist deformation more, than those in the material being crushed do. The shredding materials must possess a better strength and toughness than the plastic materials.

This study's aim to investigate the effect of operational parameters on process energy demand and quality of re-cyclates in mechanical recycling of plastics. Three control factors which will be investigated are blade, material thickness and material size. Performance of two different granulator technologies will be also compared. The vision is to develop the knowledge base for selecting optimum parameters to minimise energy footprint and to predict re-cyclites quality

1.2 Problem Statement

Paper shredders are great machines that can help keep everyone's private information under wraps. However, it doesn't matter if you own a personal-sized device or a large departmental one, you probably will experience some problems with it at some point. Here are five common problems and some ideas on how to fix them.

There may be chance of a paper jam. Even if you have a jam free shredder, you might still have to deal with a paper jam at some point, depending on your usage so this need to understand the cutting force for the blade.
Most jams can be cleared up by simply running the machine in reverse and removing the paper.
If uneven force is applied to paper blade then it may loss its balance, shredder makes too much noise. Some of the units out there are noisy from the get-go.

2. Design Process

When a new product or their elements are to be designed, a designer may proceed as follows:

- Make a detailed statement of the problems completely; it should be as clear as possible & also of the purpose for which the machine is to be designed.
- Make selection of the possible mechanism which will give the desire motion.
- Determine the forces acting on it and energy transmitted by each element of the machine.
- Select the material best suited for each element of the machine.
- Determine the allowable or design stress considering all the factors that affect the strength of the machine part.
- Identify the importance and necessary and application of the machine Problems with existing requirement of the machine productivity and demand.

In this research work each critical part of the machine will be conceptually set up and this choice will be based on criteria design criteria which will be used to produce a detailed design of machine.
The quality that makes a good design is based on the developed of a good philosophy of design. The following consideration was adopted in this design:

- Minimum vibration level
- Lower overall cost
- Machine longer and extended product life
- Good and attractive appearance of machine assembly- color and styling.
- Design for easy manufacturing
- Design for easy maintenance and assembly
- Design for high efficiency.

Facilities Required

1. Catia v5
    - Surface modeling
    - Part design

- Drafting

2. FEA Ansys Workbench R1 2020
    a. Static Structural Analysis
    b. Topology Optimization
    c. Vibrational Modal analysis
    d. Material Comparison
3. Process in Ansys
    a. Stress, strain & deformation factors with fatigue assessment on Structural steel.
    b. Mass reduction on topology optimization
    c. Frequency generation Modal analysis on 7 modes
    d. Comparing for different Material grades for strength, weight and etc.

PROPERTY	METRIC	UNITS	ENGLISH	UNITS
General				
Density	952 - 965	kg/m^3	00.344 - 0.0349	lb/ft^3
Mechanical				
Yield Strength	2.62e7 - 3.1e7	Pa	3.8 - 4.5	ksi
Tensile Strength	2.21e7 - 3.1e7	Pa	3.21 - 4.5	ksi
Elongation	11.2 - 12.9	% strain	1.12e3 - 1.29e3	% strain
Hardness (Vickers)	7.75e7 - 9.71e7	Pa	7.9 - 9.9	HV
Impact Strength (un-notched)	1.9e5 - 2e5	J/m^2	90.4 - 95.2	ft.lbf/in^2
Fracture Toughness	1.52e6 - 1.82e6	Pa/m^0.5	1.38 - 1.66	ksi/in^0.5
Young's Modulus	1.07e9 - 1.09e9	Pa	0.155 - 0.158	10^6 psi
Thermal				
Max Service Temperature	113 - 129	°C	235 - 264	°F
Melting Temperature	130 - 137	°C	266 - 279	°F
Insulator or Conductor	Insulator		Insulator	
Specific Heat Capability	1.75e3 - 1.81e3	J/kg °C	0.418 - 0.432	BTU/lb. °F
Thermal Expansion Coefficient	1.06e-4 - 1.98e-4	strain/°C	59 - 110	µstrain/°F

Table 5.1 properties of cutting waste

2.1 Equations required to calculate the cutting force for blade.

Breaking strength can be assumed as the ultimate strength multiplied by a designer factor of safety.
Breaking strength of PET plastic material:
$\tau_{(br)\text{" plastic "}}$=Fos*ultimate strength of material
The cross-sectional area of the material to be cut is
$A = w^{*} ts$
Where: W= Width of cutting blade edge,
t= Thickness of the plastic material
The cutting force required for cutting the plastic.
$F_c = \tau_{(br)plastic} * A$
Fos=2
ultimate strength of material to be cut=45Mpa
Assume thickness to cut as = 1-5 mm = 5mm
$\tau_{(br)\text{" plastic "}} = 2*45 = 90$Mpa
Form a trail & error method select 2cm to be as width of the blade
A = 20*5 = 100 mm^2
$F_c = \tau_{(br)plastic} * A$
= 90*100 = 9000 N
9000 N of force to be applied on the blade and also to optimize the blade for its yield strength.

2.2 Material

1 A2 Tool Steel

Properties	**Metric**	**Imperial**
Density	7.86 g/cm^3	0.284 lb/in^3
Melting point	1424°C	2595°F

Mechanical Properties
The mechanical properties of A2 tool steels are displayed in the following table.

Properties	**Metric**	**Imperial**
Hardness, Rockwell C (as air-hardened (63-65 HRC average), 60-62 HRC at 205°C, 59-61 HRC at 260°C, 58-60 HRC at 315°C, 57-59 HRC at 370°C and 425°C and 480°C, 56-58 HRC at 540°C, 50-52 HRC at 595°C, 42-44 HRC at 650°C)	64	64
Bulk modulus (typical for steels)	140 GPa	20300 ksi
Machinability (based on carbon tool steel)	65%	65%

Shear modulus	78.0 GPa	11300 ksi
Poisson's ratio	0.27-0.30	0.27-0.30
Elastic modulus	190-210 GPa	27557-30457 ksi

Table 5.2 Properties of A2 tool Steel

Material 2 Stainless steel

Mechanical properties for 304 stainless steel alloys - plate from 8 - 75 mm thick

Grade	304	304L	304H
Tensile Strength (MPa)	520 - 720	500 - 700	-
Proof Stress (MPa)	210 Min	200 Min	-
Elongation A5	45 Min %	45 Min %	

Property	Value
Density	8.00 g/cm^3
Melting Point	1450 °C
Modulus of Elasticity	193 GPa
Electrical Resistivity	0.72 x 10^{-6} Ω.m
Thermal Conductivity	16.2 W/m.K
Thermal Expansion	17.2 x 10^{-6}/K

Table 5.3 Material properties of 304 Stainless Steel

4. Analysis

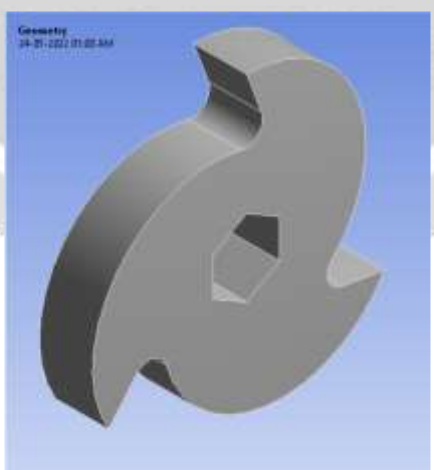


Figure 6.2 Geometry Importation

Bounding Box	
Length X	20. mm
Length Y	112.5 mm

Length Z	109.95 mm
Properties	
Volume	1.422e+005 mm³
Mass	1.1163 kg
Scale Factor Value	1.

Table geometry properties

Material – Structural Steel

Structural Steel

Fatigue Data at zero mean stress comes from 1998 ASME BPV Code, Section 8, Div 2, Table 5-110.1

Density	7.85e-06 kg/mm³

Structural

Isotropic Elasticity

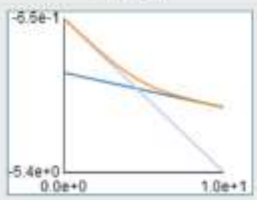
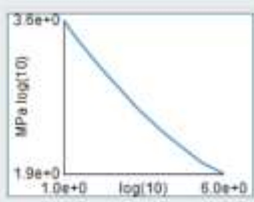
Derive from	Young's Modulus and Poisson's Ratio
Young's Modulus	2e+05 MPa
Poisson's Ratio	0.3
Bulk Modulus	1.6667e+05 MPa
Shear Modulus	76923 MPa
Isotropic Secant Coefficient of Thermal Expansion	1.2e-05 1/°C
Compressive Ultimate Strength	0 MPa
Compressive Yield Strength	250 MPa
Strain-Life Parameters	
S-N Curve	
Tensile Ultimate Strength	460 MPa
Tensile Yield Strength	250 MPa

Table 6.1 Material Properties Structural steel

Figure 6.3 Mesh Creation on 3D part body

Definition		
Suppressed	No	
Method	Automatic	
Element Order	Use Global Setting	
Type		Element Size
Element Size		2.0 mm
Advanced		
Defeature Size		Default
Behavior		Soft

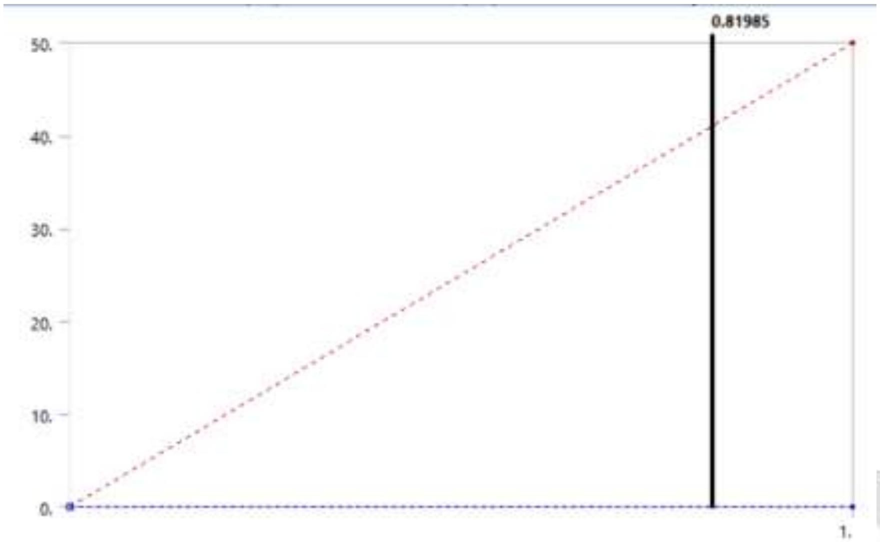
Statistics	
Nodes	107051
Elements	24010

Table 6.2 Mesh Configuration

Boundary Condition

Object Name	*Rotational Velocity*
State	Fully Defined
Scope	
Scoping Method	Geometry Selection
Geometry	All Bodies
Definition	
Define By	Components
Coordinate System	Global Coordinate System
X Component	50. RPM (ramped)
Y Component	0. RPM (ramped)
Z Component	0. RPM (ramped

Table 6.3 boundary Condition



Graph Rotational velocity

Definition		
Type	Fixed Support	Force
Suppressed	No	
Define By		Vector
Applied By		Surface Effect
Magnitude		9000. N (ramped)
Direction		Defined

Table 6.4 Force at blade face

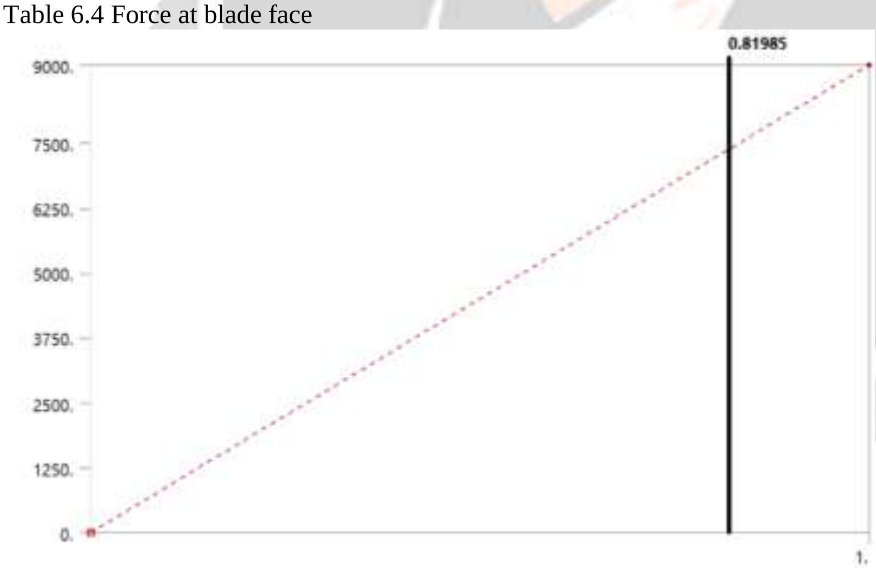


Figure Force

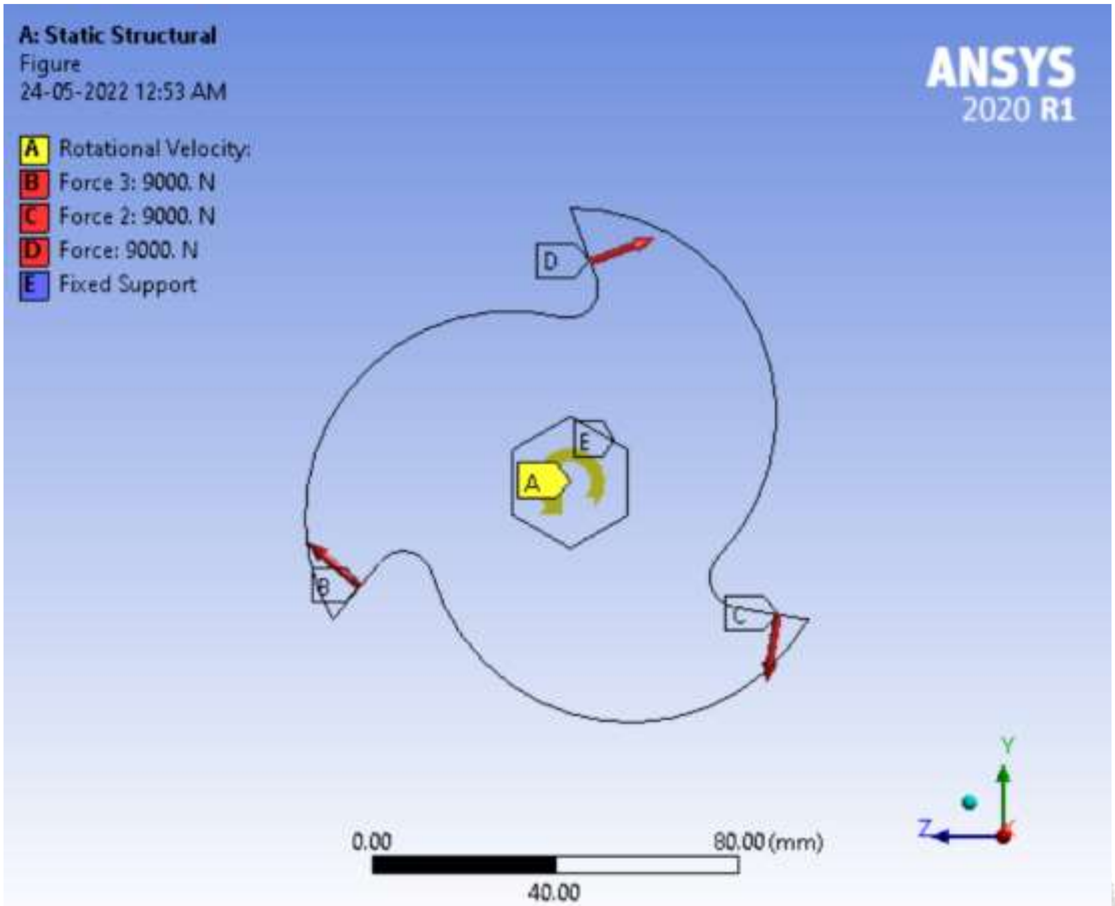


Figure 6.4 Boundary Condition

Results

Results			
Minimum	0.mm Deformation	1.028 MPa Stress	5.6068e-006 mm/mm Strain
Maximum	4.4824e-002 mm	184.59 MPa	1.0785e-003 mm/mm
Average	1.2839e-002 mm	31.124 MPa	2.3453e-001 mm/mm

Table 6.5 Over all result

Result

Total Deformation

Time [s]	Minimum [mm]	Maximum [mm]	Average [mm]
1.	0.	4.4824e-002	1.2839e-002

Table Result Total Deformation

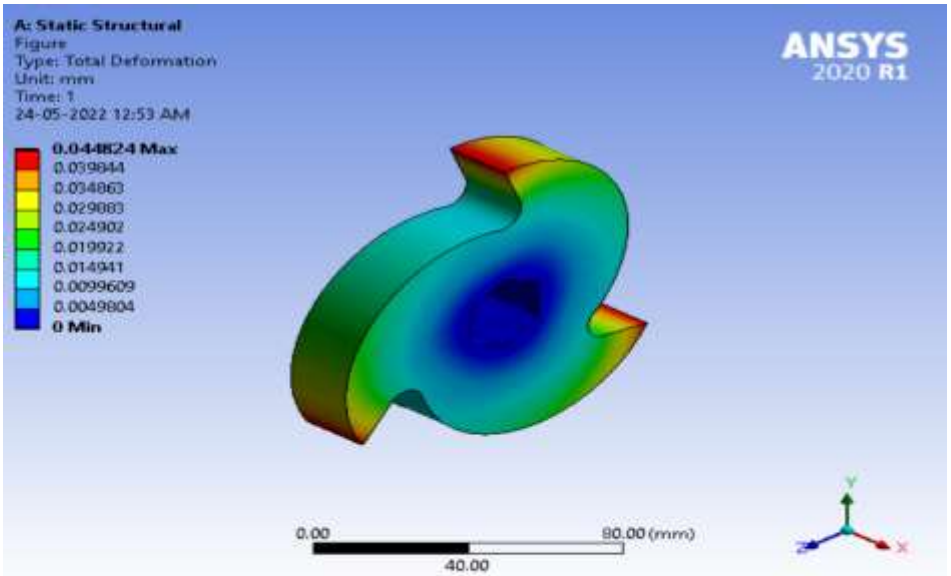


Figure 6.5 Total Deformation

Result Von-Misses Stress

Time [s]	Minimum [MPa]	Maximum [MPa]	Average [MPa]
1.	1.028	184.59	31.124

Table 6.6 Result Von-Misses Stress

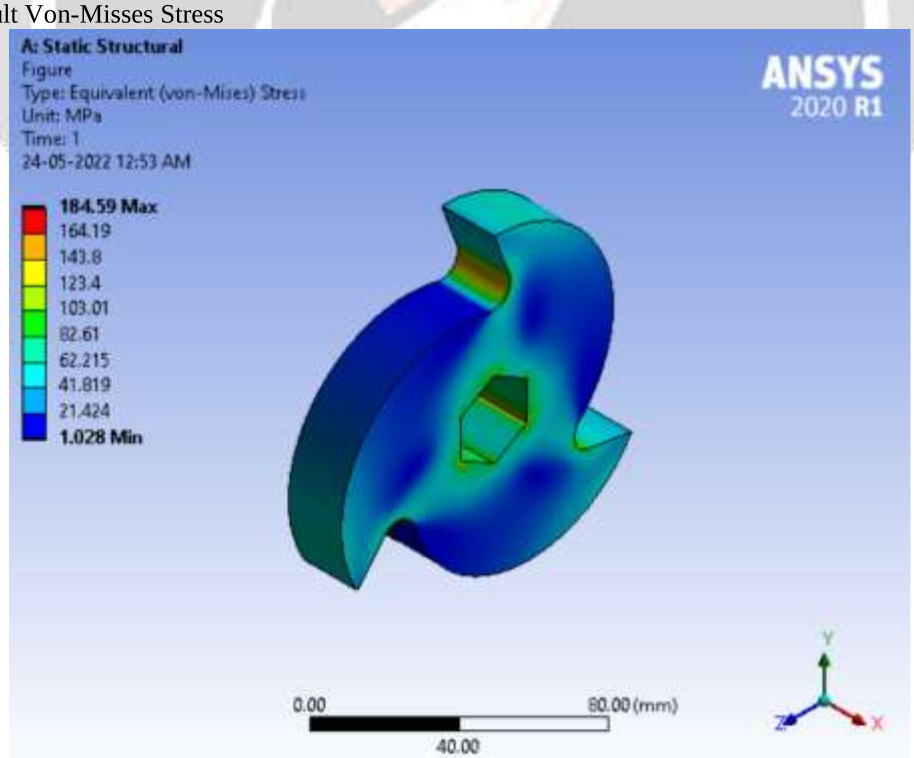


Figure 6.6 Result Von-Misses Stress

Result Von-Misses Strain

Time [s]	Minimum [mm/mm]	Maximum [mm/mm]	Average [mm/mm]
1.	5.6068e-006	1.0785e-003	1.5637e-004

Table 6.7 Result Von-Misses Strain

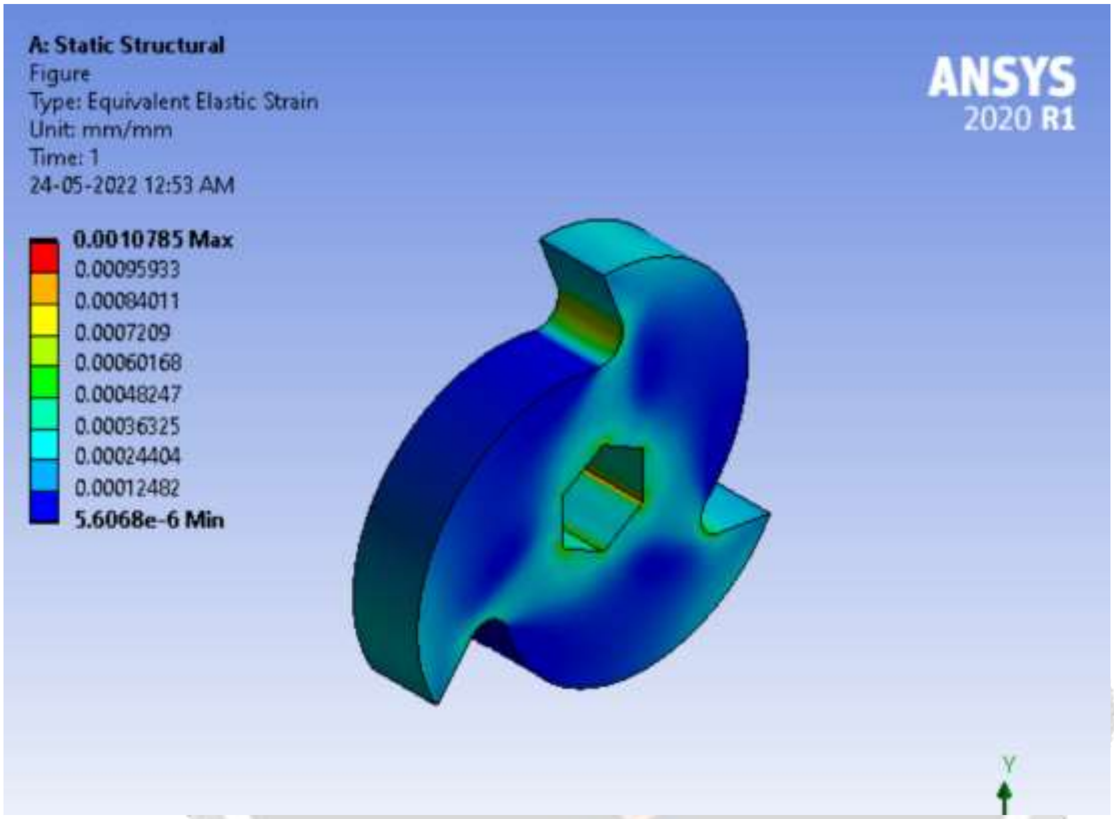
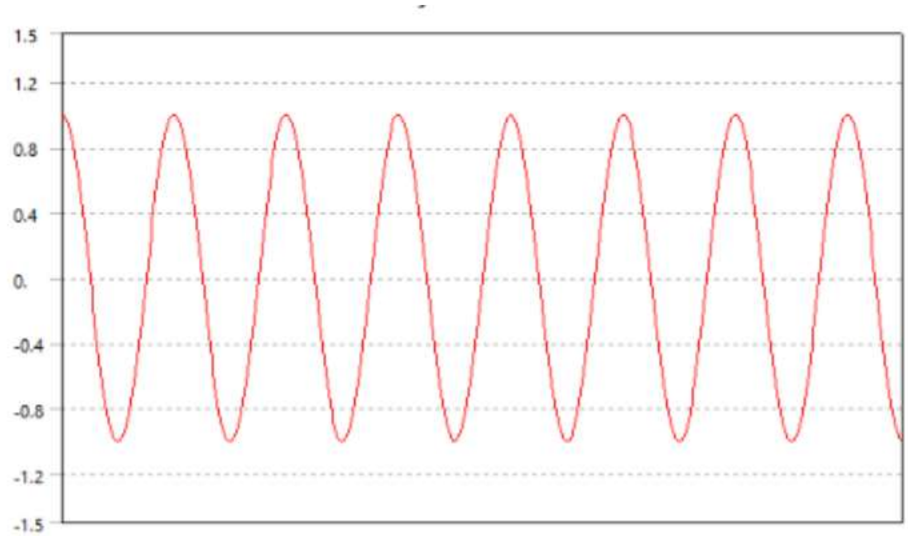


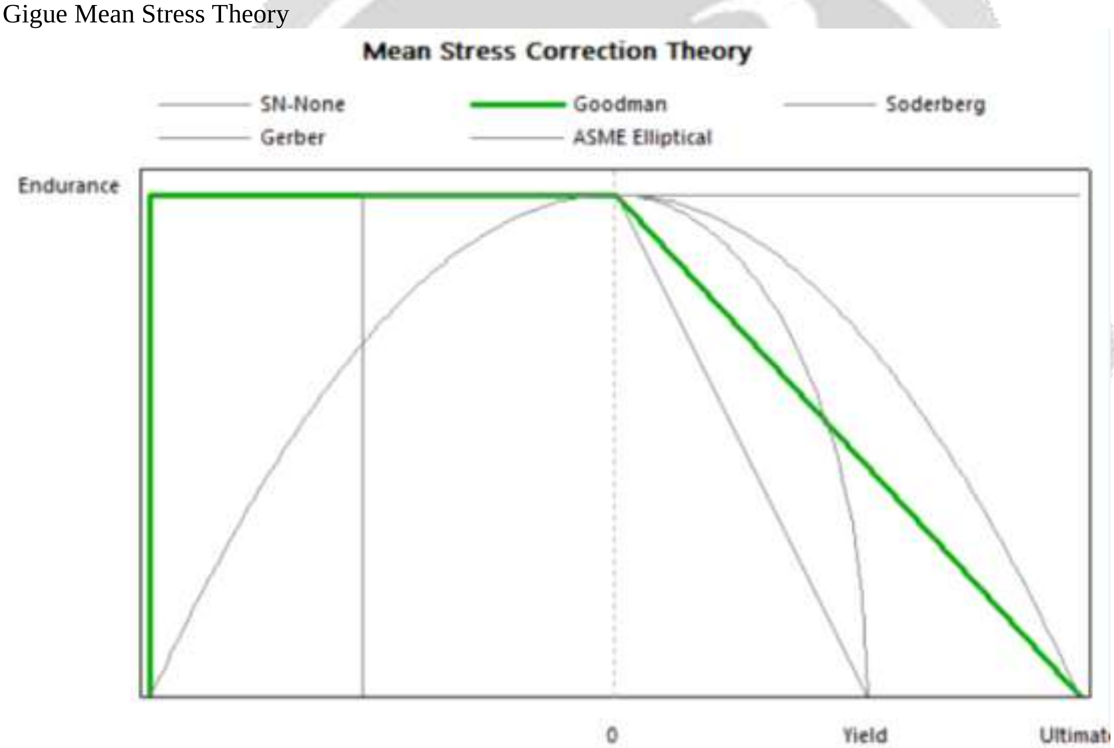
Figure 6.7 Result Von-Misses Strain

Table 6.8 Fatigue Factor

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	cycles
1 cycle is equal to	1. cycles



Gigue Mean Stress Theory



Results

Object Name	*Life*	*Damage*	*Safety Factor*	*Biaxiality Indication*
State	Solved			
Scope				

Scoping Method	Geometry Selection			
Geometry	All Bodies			
Definition				
Type	Life	Damage	Safety Factor	Biaxiality Indication
Identifier				
Suppressed	No			
Design Life		1.e+009 cycles		
Integration Point Results				
Average Across Bodies	No			
Results				
Minimum	34402 cycles		0.46698	-1.
Minimum Occurs On	Blade design-FreeParts|PartBody		Blade design-FreeParts|PartBody	
Maximum		29068		0.96613
Maximum Occurs On		Blade design-FreeParts|PartBody		Blade design-FreeParts|PartBody
Average				-0.40553

Table 6.9 Overall Fatigue Results

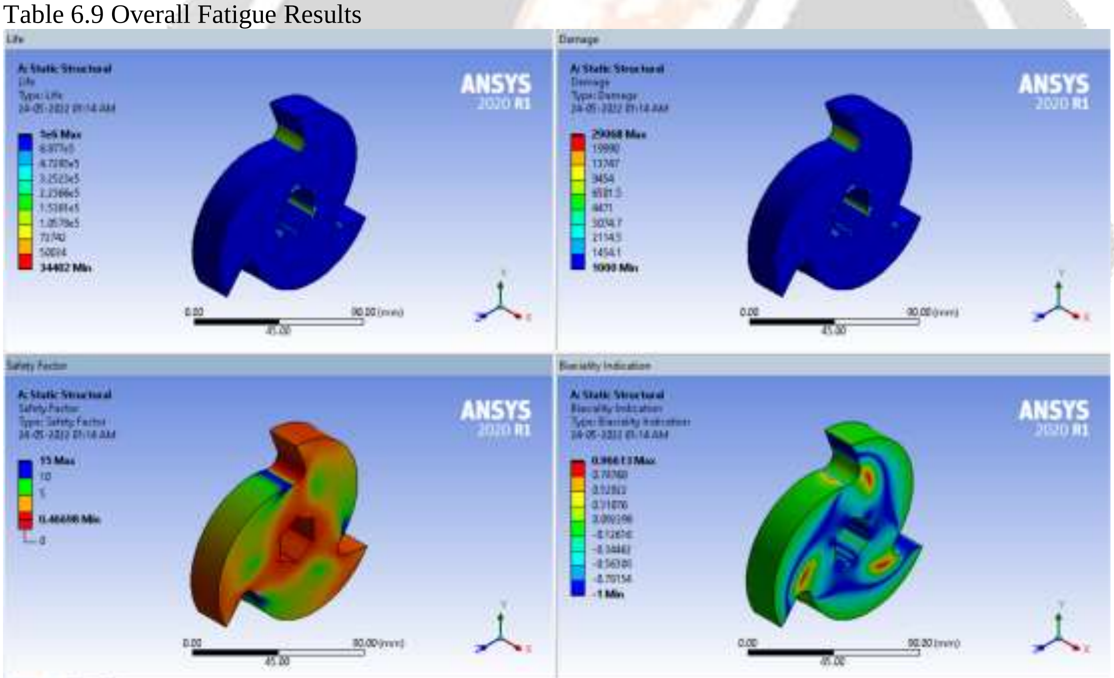
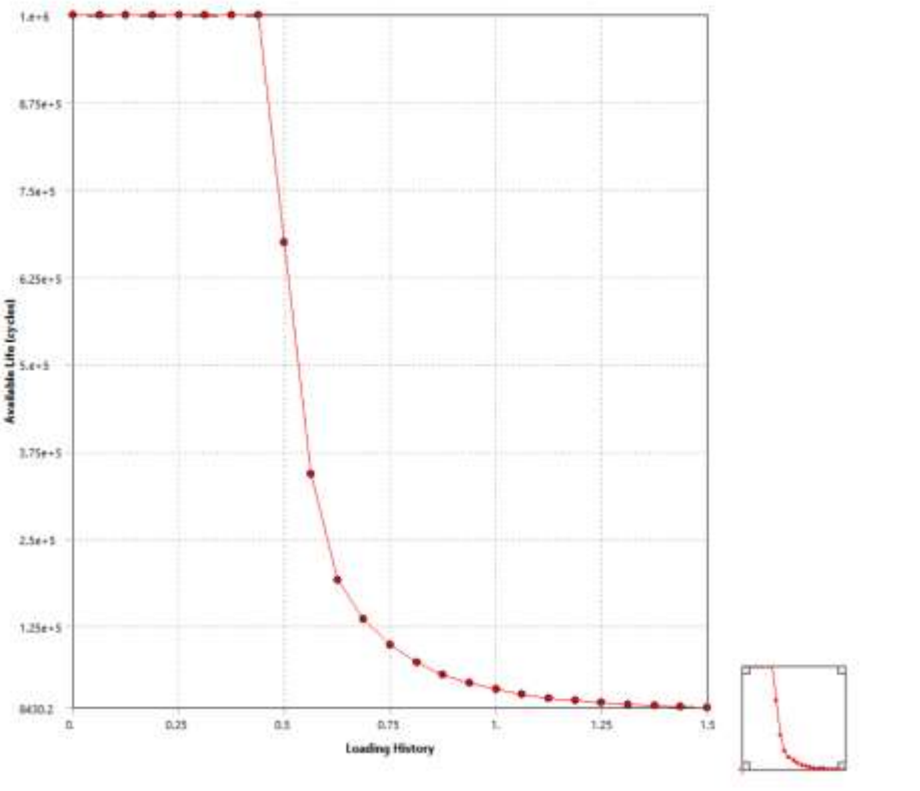


Figure 6.8 a. Life b. Damage c. Safety Factor d. Biaxiality Indication.



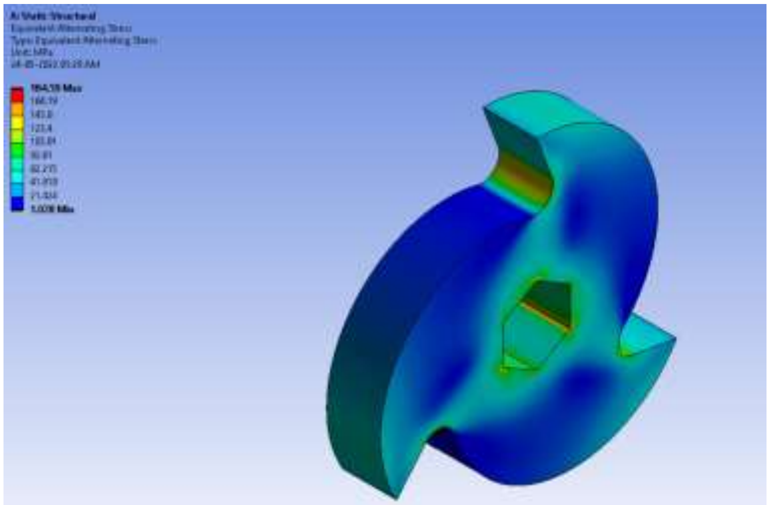
Graph Fatigue Sensitivity

Alternating Stress MPa	Cycles	Mean Stress MPa
3999	10	0
2827	20	0
1896	50	0
1413	100	0
1069	200	0
441	2000	0
262	10000	0
214	20000	0
138	1.e+005	0
114	2.e+005	0
86.2	1.e+006	0

TABLE 33
Table Sn-Curve of Material

214	**20000**	**0**
138	**1.e+005**	**0**

Table Oure alternative stress



Fatigue alternative stress

Discussion

The designed shredder blade was analyzed using Static Structural analysis for calculated boundary condition 9000 N load on blade face to cut the plastic of 45 Mpa tensile strength with an factor of safety of 2.

But the obtained safety is less then the proposed safety so, in next procedure optimization will be carried out to reduce stress factors from the blade by redesigning the part to the absolute one by trying to maintain a minimum mass increment in the part body.

2nd iteration Topology Optimization

Topology optimization generates the optimal shape of a mechanical structure. Given a predefined domain in the 2D/3D space with boundary conditions and external loads, the intention is to distribute a percentage of the initial mass on the given domain such that a global measure takes a minimum. Without any further decisions and guidance of the user, the method will form the structural shape thus providing a first idea of an efficient geometry. The design space is discretized by the finite element method to represent the material distribution and at the same time the structural behavior. Therefore, lesser deflections are produced by more material. So, the optimization constraint is the volume of the material. Integration of the selection field over the volume can be done to obtain the total utilized material volume.

Topology optimization can be implemented through the use of finite element methods for the analysis and optimization techniques based on Homogenization method, Optimality criteria method, level set, Moving asymptotes, Genetic algorithms. A brief discussion on these methods is given below.

Procedure

1. To simulate a part under topology formation, it must be simulated with one of the main modules of system like static, transient, Dynamic, CFD, Model or IC engines etc.
2. After the main module boundary processing a topology optimization module or scope is combined with the static structural analysis, results section from static are targeted into the optimization and upon the requirement we can optimize the part for required constraints mode like percentage of reduction of material from part stress based, strain based, vibrational based and mass based.

Details of "Analysis Settings"

Definition	
Maximum Number Of Iterations	500.
Minimum Normalized Density	1.e-003
Convergence Accuracy	0.1 %
Penalty Factor (Stiffness)	3.
Region of Manufacturing Constraint	Include Exclusions
Region of Min Member Size	Exclude Exclusions
Region of AM Overhang Constraint	Exclude Exclusions
Solver Controls	
Solver Type	Program Controlled
Output Controls	
Store Results At	All Iterations
Analysis Data Management	
Solver Files Directory	D:\Mech p\sudarshan sir blad analysis\Analysi...
Future Analysis	None
Scratch Solver Files Directory	
Save MAPDL db	No
Delete Unneeded Files	Yes
Solver Units	Active System
Solver Unit System	nmm
Max Num Of Intermediate Files	All Iterations

Figure number of iterations, Convergency accuracy & density of solution for optimization

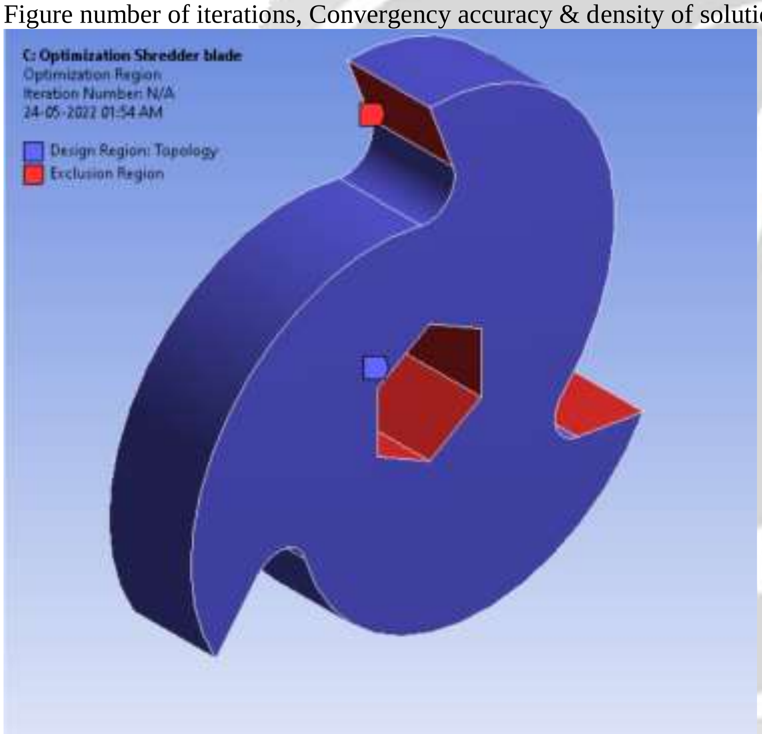


Figure 6.9 Region of optimization

Object Name	*Response Constraint*
State	Fully Defined
Scope	
Scoping Method	Optimization Region
Optimization Region Selection	Optimization Region
Definition	
Type	Response Constraint
Response	Global von-Mises Stress
Maximum	100. MPa
Environment Selection	All Static Structural
Suppressed	No

Table 6.10 Stress Reduction Response Constraint

Response Type	Goal	Criterion	Formulation	Environment Name	Weight	Multiple Sets	Start Step	End Step	Step	Start Mode	End Mode	Mode
Compliance	Minimize	N/A	Program Controlled	Static Structural	N/A	Enabled	1	1	1	N/A	N/A	N/A

Table Objective

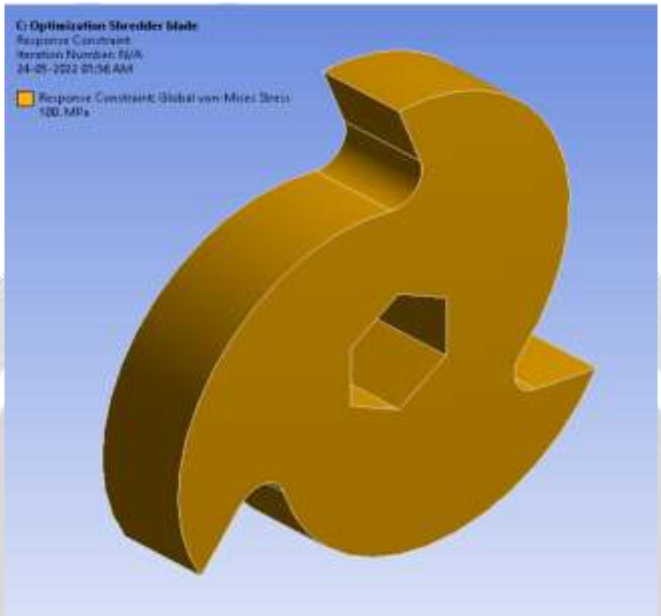
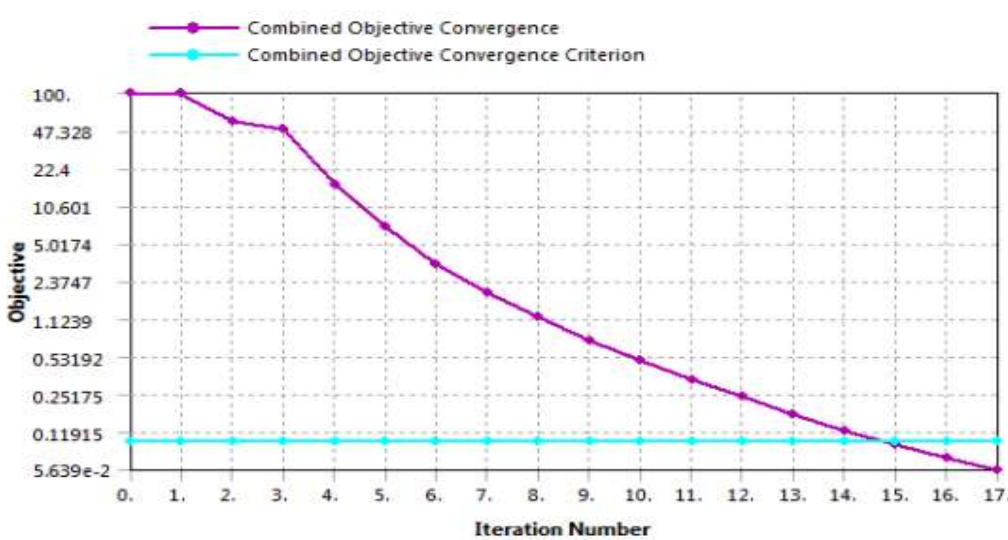


Figure 6.10 Response Constraint

Result



Graph Minimize compliance /Vs No of iterations performed

After optimization

Redesigned part with minimized material condition & equalized strength condition.

Figure 6.11 redesigned part

Weight of the geometry after Topology optimization at 100 MPa Stress Retention

Bounding Box	
Length X	25. mm
Length Y	112.5 mm
Length Z	109.95 mm
Properties	
Volume	1.5664e+005 mm^3
Mass	1.2296 kg

Table 6.11 geometry Parameter after optimization

Discussion

finally, the optimization helped in reducing the mass.

Before optimization mass of the part body = 1.1163 Kg

After Optimization mass of the part body = 1.2296 Kg

Average = 0.113grms of weight has been Increased &

In next iteration let us see how much stress has been reduced from the blade part body with mass increment of optimized part.

Iteration 3

Geometry, Mesh & Boundary Condition.

In this Iteration Same Boundary Condition is Applied to know the difference after the optimization for stress reduction.

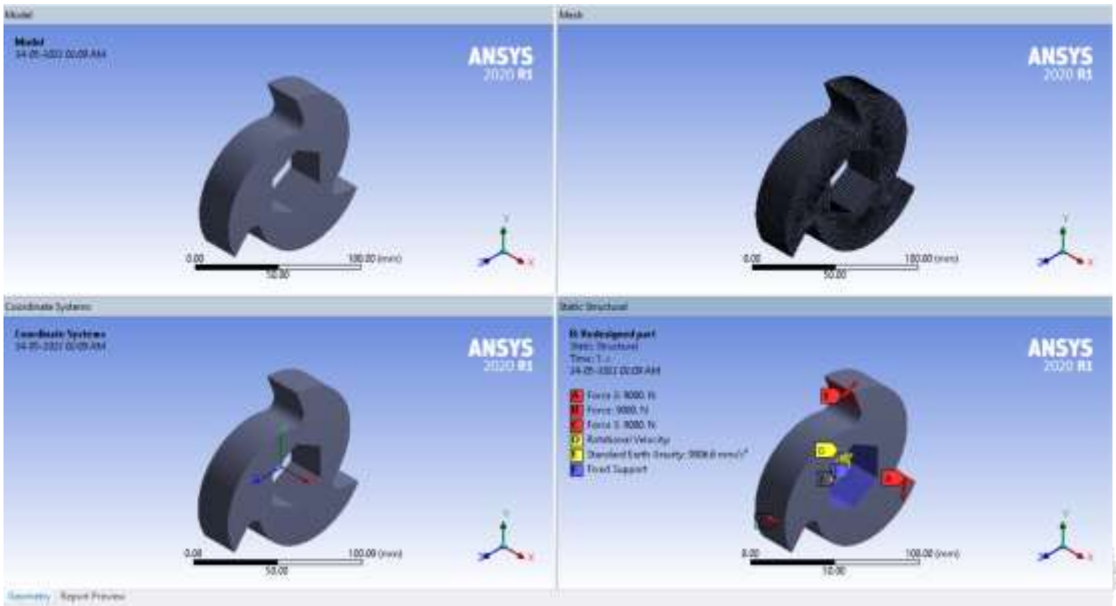


Figure 6.12 a. Geometry Importation b. Mesh Generation c. Co-Ordinate System d. Boundary Condition.

Results

Results			
Minimum	0. mm Total Deformation	0.39824 MPa Stress	2.4609e-006 mm/mm Strain
Maximum	2.5015e-002 mm	113.28 MPa	5.6643e-004 mm/mm
Average	4.2803e-003 mm	21.148 MPa	1.0614e-004 mm/mm

Table Overall Results

Fatigue Result

Object Name	*Safety Factor*	*Life*	*Damage*	*Biaxiality Indication*	*Equivalent Alternating Stress*
State	Solved				
Scope					
Scoping Method	Geometry Selection				
Geometry	All Bodies				
Definition					
Design Life	1.e+009 cycles		1.e+009 cycles		
Type	Safety Factor	Life	Damage	Biaxiality Indication	Equivalent Alternating Stress
Identifier					
Suppressed	No				

Integration Point Results					
Average Across Bodies	No				
Results					
Minimum	0.76091	2.0738e+005 cycles		-0.99999	0.39824 MPa
Maximum			4822.1	0.99054	113.28 MPa
Average				-0.37431	21.148 MPa

Table 6.12 Fatigue Overall Results

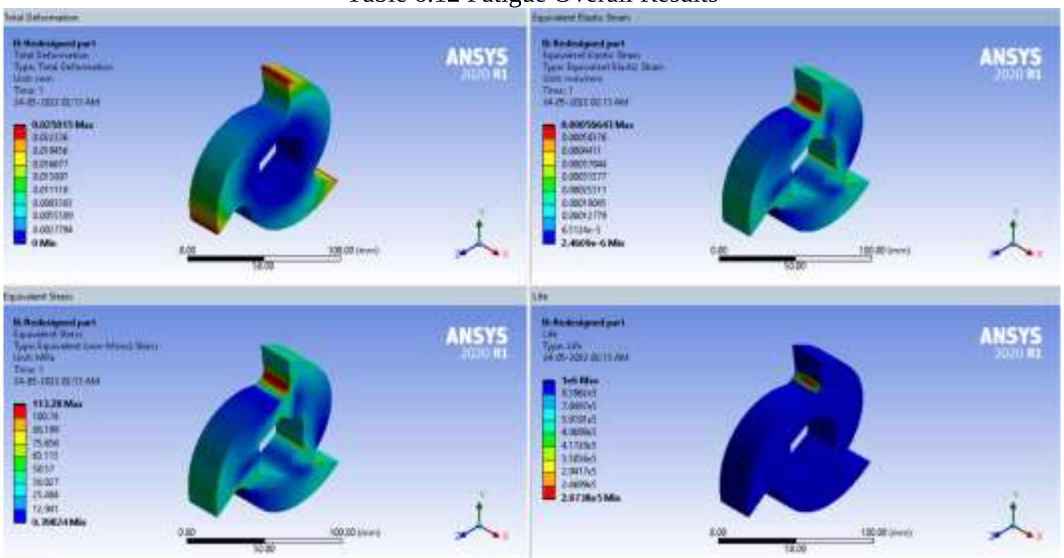


Figure a. Total Deformation. b. Strain c. Stress d. Life

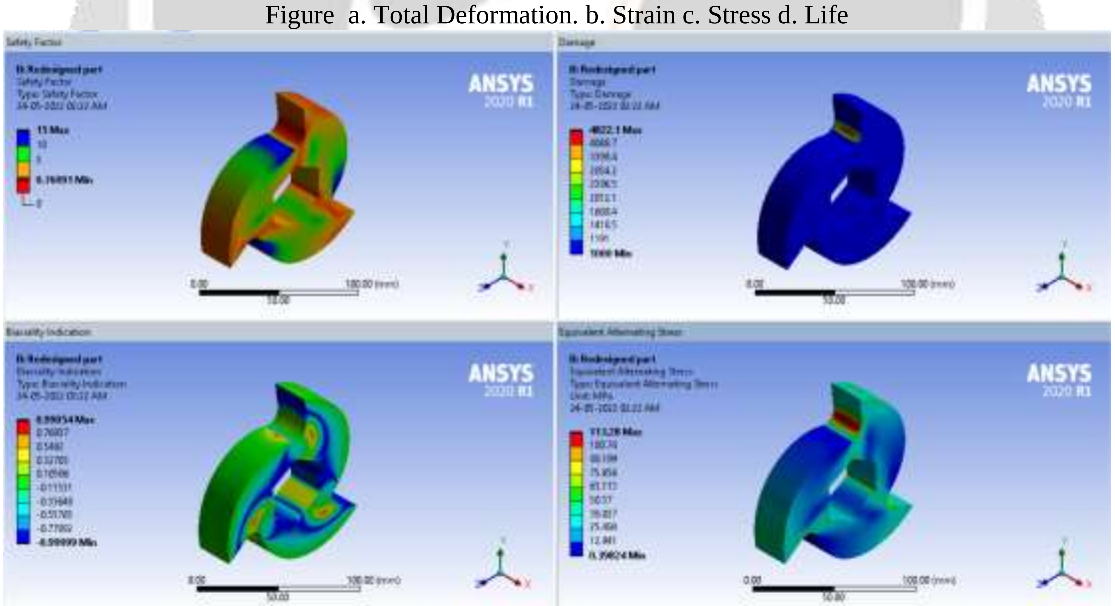
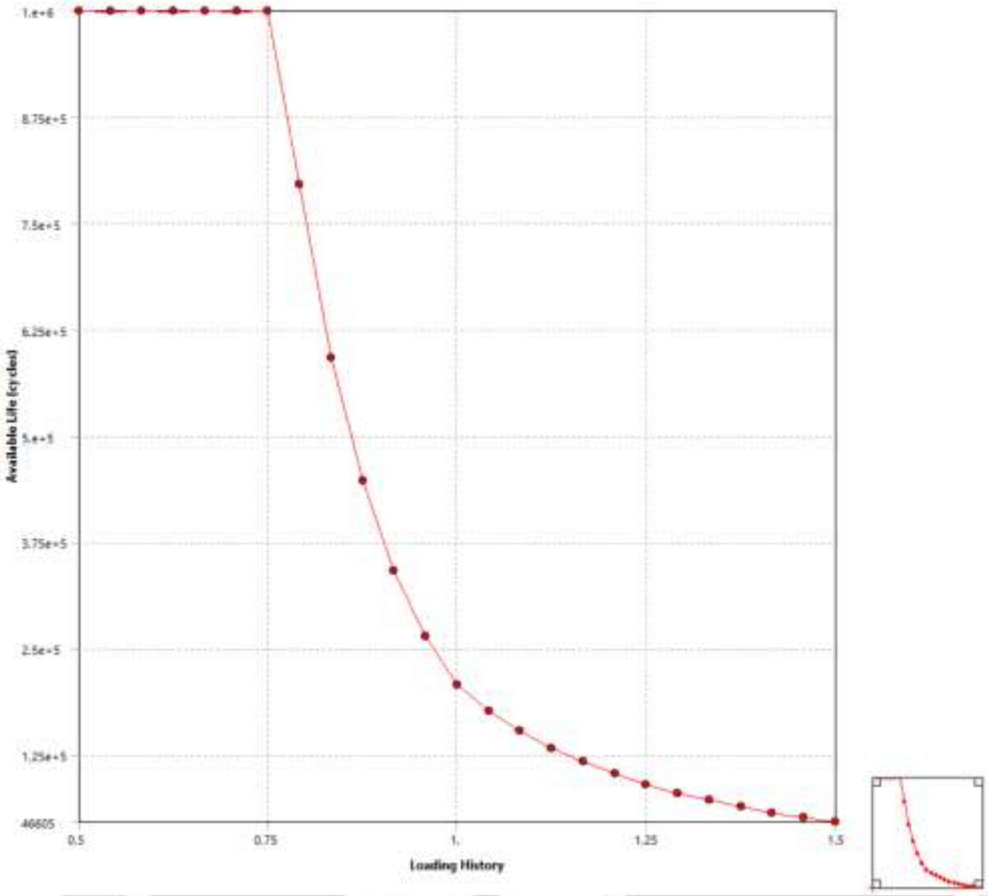


Figure a. Safety Factor b. Damage c. Biaxiality Indication d. alternative stress



Graph Fatigue Sensitivity

Alternating Stress MPa	Cycles	Mean Stress MPa
3999	10	0
2827	20	0
1896	50	0
1413	100	0
1069	200	0
441	2000	0
262	10000	0
214	20000	0
138	1.e+005	0
114	2.e+005	0
86.2	1.e+006	

Table SN- Curve

114	2.e+005	0
86.2	1.e+006	0

Our Alternative strength

Discussion

Before optimization mass of the part body = 1.1163 Kg
After Optimization mass of the part body = 1.2296 Kg

Average = 0.113grms of weight has been Increased &
Alternative stress Before optimization mass of the part body = 184.59 MPa
Alternative stress After optimization mass of the part body = 113.28 MPa

- Hence the part body optimized was successfully designed to reduce the stress factor only by increasing the mass to a 113 grms.
- The part body with optimized parameter will be feasible to fabricate then the parent section.
- In next iteration two materials will be compared with the optimized part body, to know variation of strength with respect to material physical property.

Iteration 4 Material Comparison
Material 1 A2 Tool Steel

Bounding Box	
Length X	25. mm
Length Y	112.5 mm
Length Z	109.95 mm
Properties	
Volume	1.5664e+005 mm³
Mass	1.2312 kg
Scale Factor Value	1.

Table geometry property

Results			
Minimum	0. mm Deformation	0.4477 MPa Stress	2.3553e-006 mm/mm Strain
Maximum	2.3744e-002 mm	112.86 MPa	5.3744e-004 mm/mm
Average	4.0316e-003 mm	21.159 MPa	1.0114e-004 mm/mm

Table Overall Result

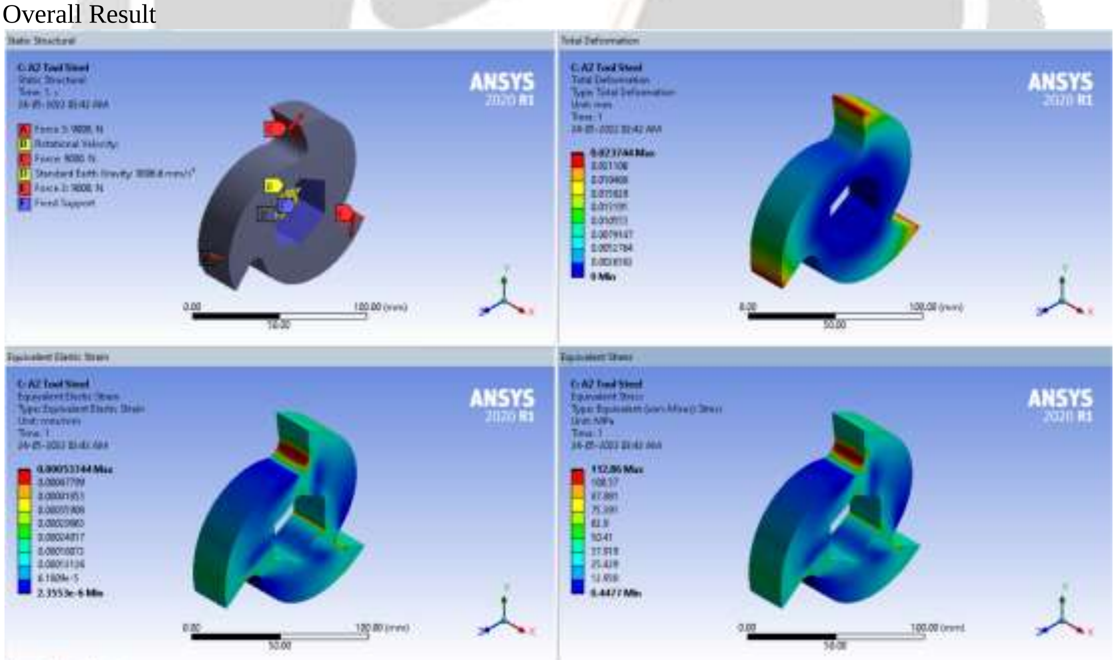


Figure a. Boundary Condition b. Total Deformation c. Strain d. Stress
Alternative stress = 112.86

114	2.e+005	0
86.2	1.e+006	0

Our Alternative strength

Material 2 Stainless steel

Bounding Box	
Length X	25. mm
Length Y	112.5 mm
Length Z	109.95 mm
Properties	
Volume	1.5664e+005 mm³
Mass	1.2531 kg
Scale Factor Value	1.

Table geometry property

Results			
Minimum	0. mm Deformation	0.46241 MPa Stress	2.8672e-006 mm/mm Strain
Maximum	2.5949e-002 mm	113.45 MPa	5.8781e-004 mm/mm
Average	4.4514e-003 mm	21.144 MPa	1.0998e-004 mm/mm

Table Overall Result

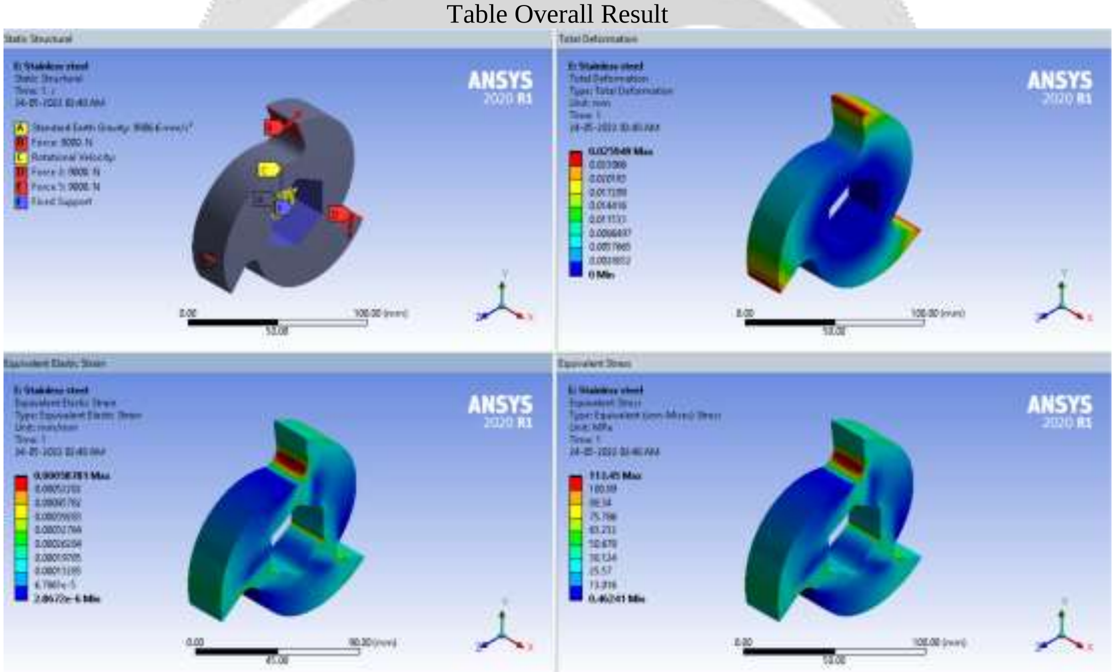


Figure a. Boundary Condition b. Total Deformation c. Strain d. Stress

Alternative stress = 113.45 Mpa

114	2.e+005	0
86.2	1.e+006	0

Our Alternative strength

4. CONCLUSIONS

FEA Static Structural Analysis had been successfully conducted on the Engine Mount bracket for the self-load condition, to investigate the stress concentration factor & vibrational modes of frequency for a defined boundary condition. Finally, all the results were observed and noted down.

In first iteration the proposed modal was solved for the Static condition, stress and deformation factors were more so on the Blade for an applied boundary Condition, so optimization strategy was used to reduce the Stress and also to maintain equalized Mass.

After 1st optimization redesign was made, by Editing the geometry and then solved for the same. This time deformation and stress factor were brought to minimum by conducting topology method.

Material Comparison For final designed part of Blade, Material comparison was Made to investigate the stress factors for A2 Tool steel & Stainless-steel Alloy, Hence the solution was optimum as expected.

The following result table explains the FEA modulation for designed, optimized part of engine mount bracket.

Table of Result

Sl No	Material	Type of State	Deformation In mm	Strain	Stress in MPa	Mass of part body in Kg
1.	Structural Steel	Static Structural Analysis	0.00482	1.07e-3	184.59	1.1163
2.	Structural steel	After Redesign part	0.002501	5.66e-4	113.28	1.2296
3.	A2 tool Steel	After Redesign part	0.002374	5.373-4	112.86	1.2312
4.	Stain-less steel	After Redesign part	0.002549	5.87e-4	113.45	1.2531

Table 7.1 overall result column

Structural steel is low cost, high strength material for cutting plastics with the shredder machine.

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