Weight Reduction of Engine Mounting Bracket of a Passenger Car by using Strain Gauge Method

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Abstract - *Nowadays automotive part designing is totally based on strength & light weight as per as mobility is concerned for performance enhancement. So, for saving of cost in production methods of parts of automobile and with consideration of weight of parts in which a region can be focused having less stresses will be cut within geometry of parts. The main purpose of an engine mounting bracket is to support the power –train system in an Automobile in all conditions of road surfaces including even, uneven road surfaces. It is very difficult to change the supporting locations and the types of support after the engine is built, the mounting brackets must be verified in the design stage. This project contains the study of vibration and Optimization of an engine mounting bracket and comparison between existing and optimized engine mounting bracket. Optimize model will be manufacturing using SS. material with reinforcement of E- glass and carbon fibre layer separately. CAD model has been generated through reverse engineering. The bracket of TATA CUMMINS Engine is to be taken for study. After analyzing the engine mounting bracket of TATA CUMMINS, Optimization is done. Experimental testing will be performed with help of UTM testing*

Keywords - *Engine mounting bracket, Composite material, FEA, UTM.*

1. INTRODUCTION

In this automotive era the need for light weight structural materials is increasing as there is a more focus on fuel consumption reduction and improvement in decreasing the emission. The magnitude of production volumes has traditionally placed severe requirements on the robustness of process used in the manufacturing. The manufacturers have strong importance on the cost has the demand for the component to improve the material performance and to deliver these materials at low cost is the requirement. In automobile sector the extremely competitive automotive business needs manufactures to pay a lot of attention to traveling comfort. Resonant vibration is from unbalanced masses exist within the engine body, this is causing the designers to direct their attention to the event of top quality engine mounting brackets so as to confirm that there is improvement in riding comfort. The demand for higher play acting engine mount brackets should not be offset by arise within the production prices and/or development cycle time.In diesel engine, the engine mounting bracket is the major problem as there is unthrottled condition and higher compression ratio and even there are more speed irregularities at low speed and low load when compared to gasoline engines. So due to this there are more vibration excitation. By this vibration engine mount bracket may fail, so by optimizing the shape and thickness of engine mount bracket we can improve the performance at initial design stages. By some studies it is observed that brackets saved 38% of mass. Structural optimization is an important tool for an optimum design; comparison in terms of weight and component performance structural optimization techniques is effective tool to produce higher quality products at lower cost.

Fig 1: Engine Mounting Bracket

2. LITERATURE SURVEY

Sreekanth Dondapatibet al. [1] in the present work, experimental investigation on the failure of a muffler mounting bracket attached to commercial vehicle is done. Cracks are identified at the welded location of muffler mount which shows that weld joint has better strength than the muffler/bracket body. To understand the possible root causes of the failure, fishbone diagram was used, which helped in determining the major causes of the failure by a graphical representation. Further, the three parameter Weibull distribution was also developed to determine the Mean Time to Failure (MTTF) life which was found to be 15,172 km. In addition, tensile testing of sheet metal was performed on the sheets which was used in the manufacturing of Muffler. Furthermore, a Thermo-Mechanical coupled analysis was carried out using commercial code, ANSYS 16.0, which adapts Finite Element Analysis (FEA) formulation. The thermal loads on muffler were imported to structural analysis along with a static load of 4 g acceleration were imposed on the muffler body to simulate the effects of high impact loads.

Joong Jae Kim et al. [2] in order to obtain an automatically designed shape of engine mount, an optimum shape design process of engine mounting rubber using a parametric approach is introduced. The optimization code is developed to determine the shape to meet the stiffness requirements of engine mounts, coupled with a commercial nonlinear finite element program. A bush type engine mount being used in a passenger car is chosen for an application model. The shape from the result of the parameter optimization is determined as a final model with some modifications. The shape and stiffness of each optimization stage are shown and the stiffness of the optimized model along the principal direction is compared with the design specification of the current model. Finally, an overview of the current status and future works for the engine mount design are discussed.

Liu Qianga et al. [3] because of the launch vibration and shock, magnetically suspended flywheels (MSFWs) are equipped with an additional launch locking protective device (LLPD), and the LLPD performance has great influence on the attitude control precision of the flywheel system. In this paper, a LLPD that takes the carbon fiber bracket as the key clamped and releasable mechanism was presented. And the configuration, operating principle and functional performance requirements were introduced. The locking/unlocking force, maximum stress and contact force of the carbon fiber bracket were analyzed. The dynamic analysis of the single carbon fiber bracket equivalent to the cantilever beam model was carried out. Subsequently, the sensitivity of the constraint variables vs the structural parameters was calculated. The lower and upper parts of the carbon fiber bracket were separately optimized. The result shows that the mass of the carbon fiber bracket can reach to the minimum of 60.5 g when the number of the upper carbon fiber bracket slices is 12. Finally, the LLPD prototype was manufactured and its locking protection for the flywheel system was verified by the swept-sine vibration and the random vibration.

Maryam Hajizadehet al. [4] the bonding strength of bracket–adhesive tooth system should be high enough to withstand different loads applied either for treatment purpose or by patient. Different parameters affect the bond strength of bracket–adhesive-tooth system; however, only a few studies have reviewed the effect of orthodontic bracket base on bond strength of bracket–adhesive-tooth system. In this study, optimization of the bracket base geometry for teeth with planar enamel surface was investigated in order to increase the shear, tensile and torsional bond strength of bracket–adhesive-tooth system. Materials and methods: Rectangular bracket was primarily bonded on maxilla central tooth to measure stress distribution of bracket–adhesive-tooth system with applying shear and tensile forces and torsional moment. Trapezoidal, hexagonal and elliptical brackets were then modeled for this planar enamel surface tooth. All of these brackets were bonded to tooth separately and similar loading conditions were applied on the bracket of each system. Stress distributions of bracket–adhesive-tooth systems were calculated and compared to each other. Results: It was observed that for hexagonal bracket–adhesive-tooth system, adhesive layer and enamel, and for elliptical bracket the bracket and enamel layer were of more symmetric and appropriate pattern of stress distribution and lower maximum stress. Therefore, these shapes of bracket are more proper than the other two shapes for a planar enamel surface tooth. Conclusion: Bracket base geometry was confirmed to crucially affect the bond strength of bracket–adhesive-tooth system through finite element analysis approach.

B. Vijaya Ramnathaet. Al [5] A well-designed runner and gating system is very important to produce good quality die castings by providing a homogenous mould filling pattern. Flow analysis of the component is done in order to visibly analyse the cavity filling process. In this study, a Commutator End (CE) bracket, a cold chamber die casted product was chosen. Initially when the component was casted numerous defects such as Cold shuts, Misrun, Shrinkage porosity and Gas porosity were found. This in turn led to rejection of number of components. In order to improve the quality of the castings produced, the gating system was changed from the existing flat gate to modified spoon fed gate. The component was designed using Pro-Engineer and flow analysis was carried out using Rotork Flow 3D Software.

3. PROBLEM STATEMENT

Automotive parts like mount brackets are having weight which leads to increase in total weight of automobile with less performance for mobility, as it is directly affects the mileage & cost. To overcome this problem modelling of bracket will be done in CAD & analyzing in CAE for induced stresses & deformation. Excess material will be search by region having less stresses & that region will be cut & again optimized model is analyzed for comparison with experimental test results.

4 OBJECTIVES

1. Modelling Engine mounting bracket in CATIA V5 software.
2. Analyzing for stresses and deformation in engine mounting bracket of vehicle
3. Topological optimization for the model.
4. Optimize model will be manufacturing using structure steel material with reinforcement of glass carbon fiber layer.
5. To perform Model analysis of both engine mounting bracket with help of ANSYS 19 software

6. To perform experimental testing of new, optimize engine mounting bracket on UTM for measuring strain.

5. FEA ANALYSIS

CAD Model:

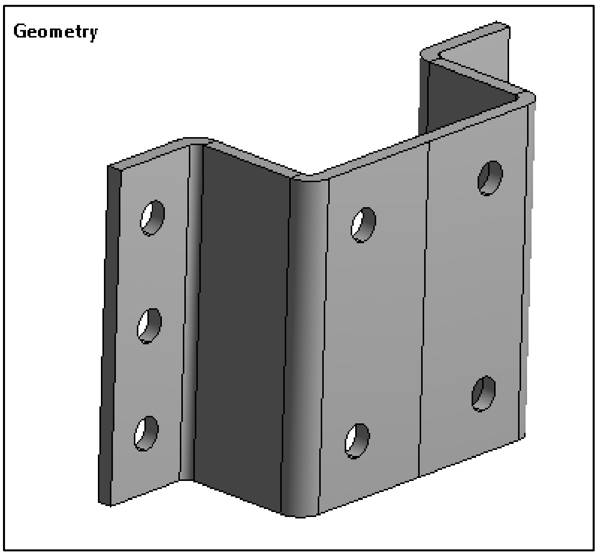


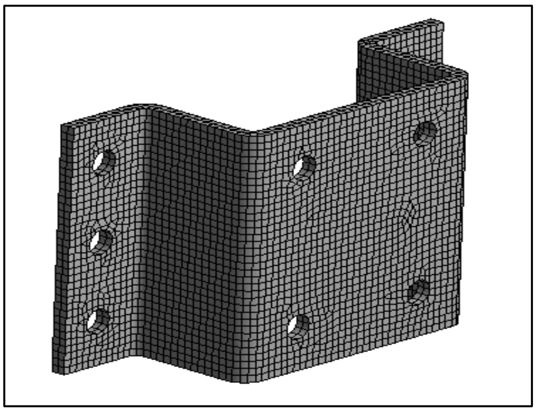
Fig 2: CAD Model of Engine mounting bracket

Table. Material Properties

	A	B	C
1	Property	Value	Unit
2	Material Field Variables	Table	
3	Density	7850	kg m^-3
4	Isotropic Secant Coefficient of Thermal Expansion		
6	Isotropic Elasticity		
7	Derive from	Young's Modulus and Pois...	
8	Young's Modulus	2E+11	Pa
9	Poisson's Ratio	0.3	
10	Bulk Modulus	1.6667E+11	Pa
11	Shear Modulus	7.6923E+10	Pa

MESHING:

In ANSYS meshing is performed as similar to discretization process in FEA procedure in which it breaks whole components in small elements and nodes. So, in analysis boundary condition equation are solved at this elements and nodes. ANSYS Meshing is a general-purpose, intelligent, automated high-performance product. A mesh well suited for a specific analysis can be generated with a single mouse click for all parts in a model. The power of parallel processing is automatically used to reduce the time you have to wait for mesh generation.



Statistics	
Nodes	5565
Elements	5348

Fig 3: Details of meshing of mounting bracket

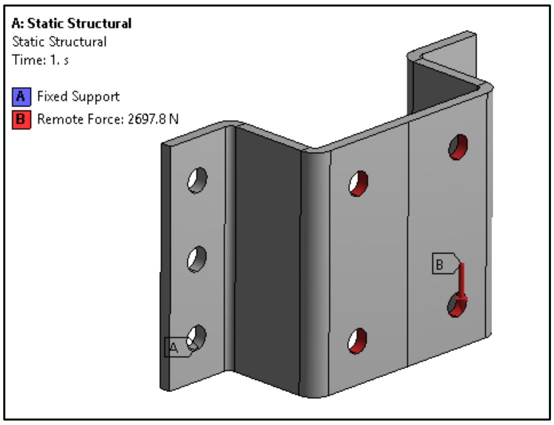


Fig 4: Boundary condition

Boundary condition are applied as per research paper to determine stress and deformation with and without reinforced composite material.

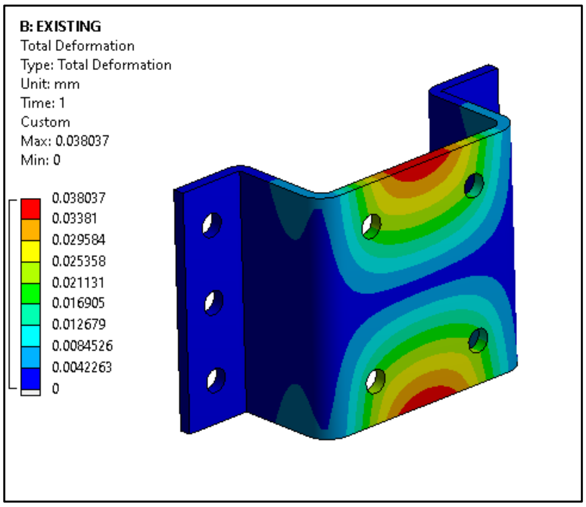


Fig 5: Deformation results

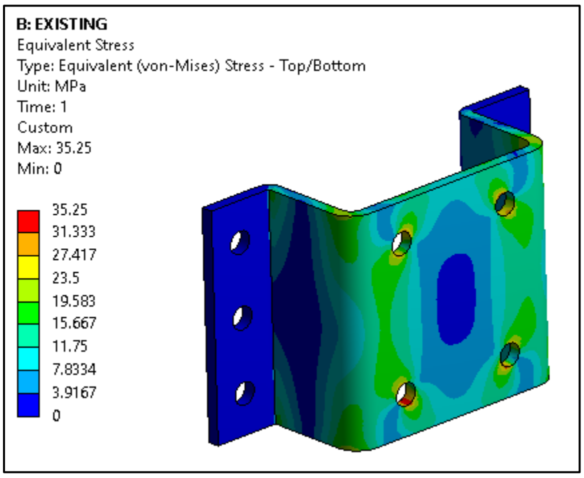


Fig 6: Equivalent stress results

MODAL ANALYSIS:

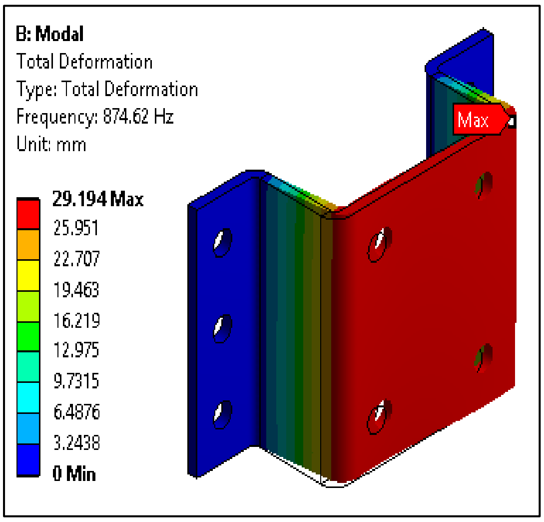


Fig 7: Mode shape 1

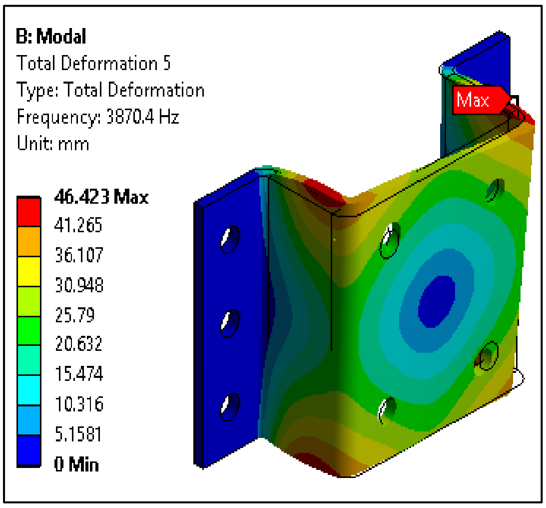


Fig 8: Mode shape 5

6. FEA OF ENGINE MOUNTING BRACKET GLASS FIBER REINFORCEMENT

Geometry

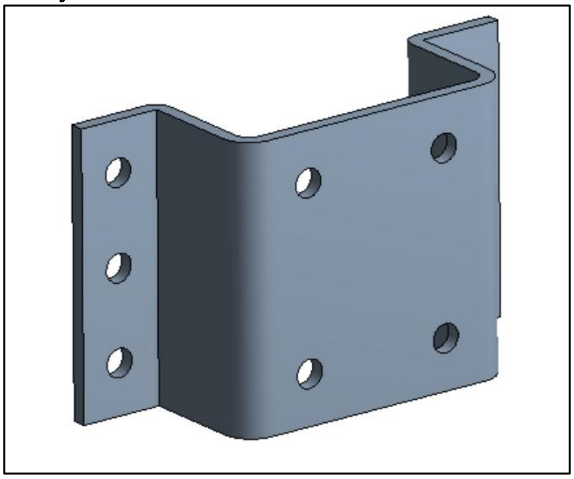


Fig 9: Geometry of engine mounting bracket glass fiber reinforcement

Material properties

Properties of Outline Row 9: Epoxy E-Glass UD

	A	B	
1	Property	Value	
2	Density	2E-09	mm^-3 t
3	Orthotropic Elasticity		
4	Young's Modulus X direction	45000	MPa
5	Young's Modulus Y direction	10000	MPa
6	Young's Modulus Z direction	10000	MPa
7	Poisson's Ratio XY	0.3	
8	Poisson's Ratio YZ	0.4	
9	Poisson's Ratio XZ	0.3	
10	Shear Modulus XY	5000	MPa
11	Shear Modulus YZ	3846.2	MPa
12	Shear Modulus XZ	5000	MPa
13	Orthotropic Stress Limits		
14	Tensile X direction	1100	MPa
15	Tensile Y direction	35	MPa
16	Tensile Z direction	35	MPa
17	Compressive X direction	-675	MPa
18	Compressive Y direction	-120	MPa
19	Compressive Z direction	-120	MPa
20	Shear XY	80	MPa
21	Shear YZ	46.154	MPa
22	Shear XZ	80	MPa

Layer	Material	Thickness (mm)
(+Z)		
4	Epoxy E-Glass UD	1
3	Epoxy E-Glass UD	1
2	Epoxy E-Glass UD	1
1	Structural Steel	4
(-Z)		

Composite of each layer with structural steel of 4 mm thickness and composite layer of each 1 mm is provided

Mesh

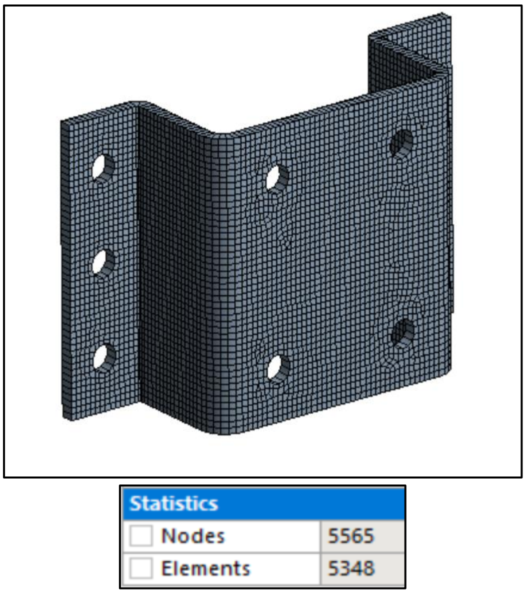


Fig 10: Meshing of engine mounting bracket glass fiber reinforcement

Boundary condition

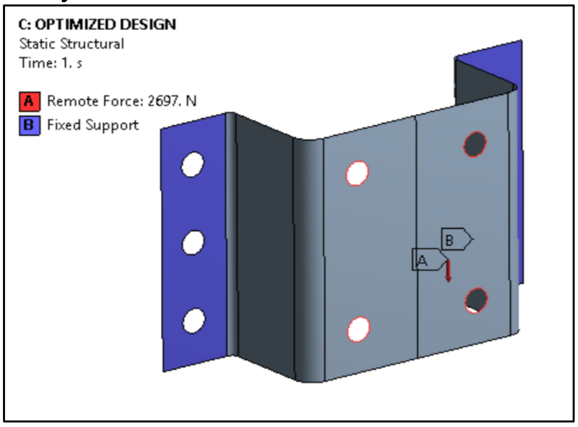


Fig 11: Boundary condition of engine mounting bracket glass fiber reinforcement

Total deformation

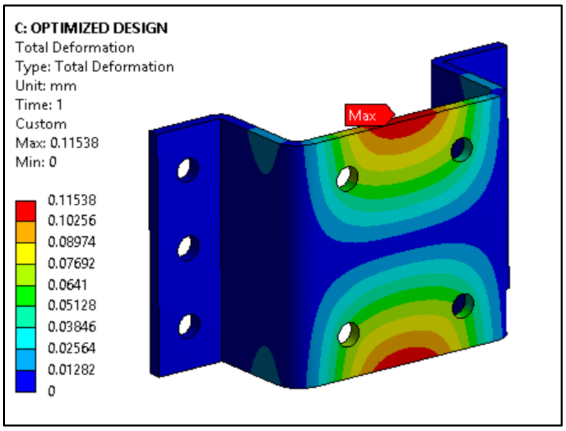


Fig 12: Total deformation of engine mounting bracket glass fiber reinforcement.

LAYER WISE EQUIVALENT STRESS ON GLASS FIBRE

AT LAYER 1

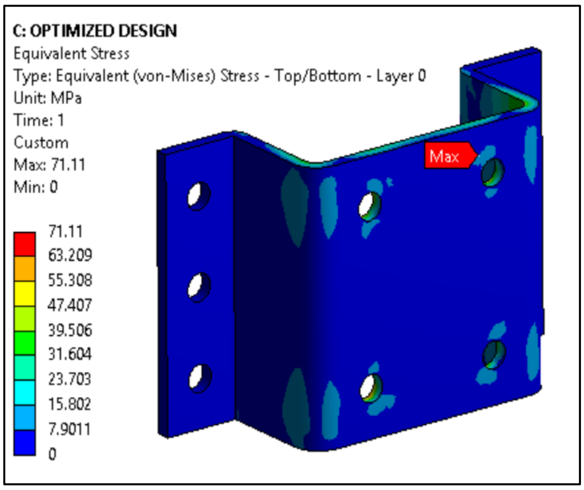


Fig 13: Equivalent stress at layer 1 of engine mounting bracket glass fiber reinforcement

AT LAYER 2

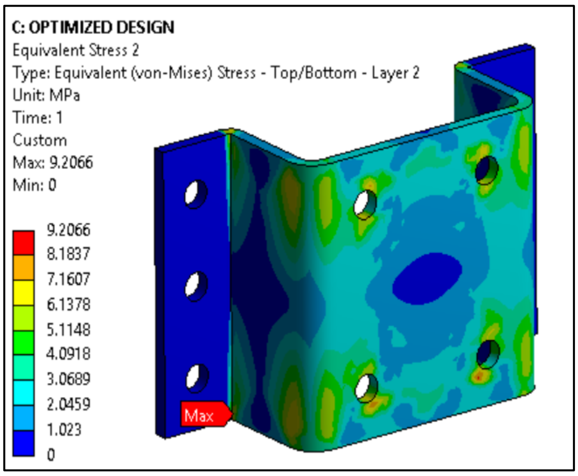


Fig 14: Equivalent stress at layer 2 of engine mounting bracket glass fiber reinforcement

AT LAYER 3

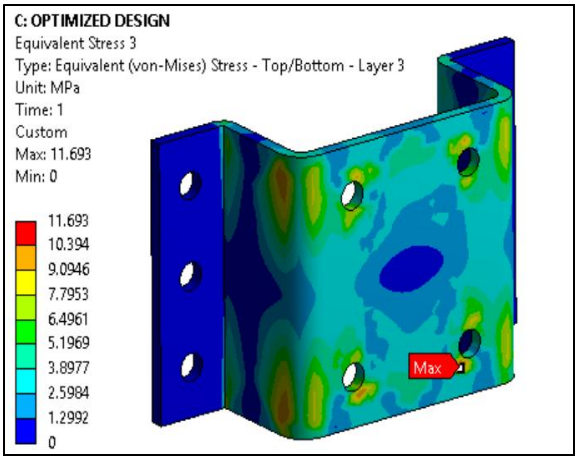


Fig 15: Equivalent stress at layer 3 of engine mounting bracket glass fiber reinforcement.

7. FEA OF ENGINE MOUNTING BRACKET CARBON FIBER REINFORCEMENT

Material properties of carbon fibre

Properties of Outline Row 4: Epoxy Carbon UD (230 GPa) Prepreg

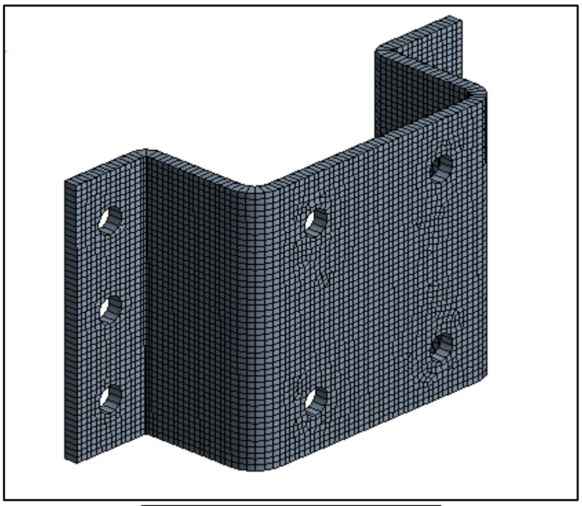
	A	B	C
1	Property	Value	Unit
2	Density	1.49E-09	mm^-3 t
3	Orthotropic Secant Coefficient of Thermal Expansion		
8	Orthotropic Elasticity		
9	Young's Modulus X direction	1.21E+05	MPa
10	Young's Modulus Y direction	8600	MPa
11	Young's Modulus Z direction	8600	MPa
12	Poisson's Ratio XY	0.27	
13	Poisson's Ratio YZ	0.4	
14	Poisson's Ratio XZ	0.27	
15	Shear Modulus XY	4700	MPa
16	Shear Modulus YZ	3100	MPa
17	Shear Modulus XZ	4700	MPa
18	Orthotropic Stress Limits		
19	Tensile X direction	2231	MPa
20	Tensile Y direction	29	MPa
21	Tensile Z direction	29	MPa
22	Compressive X direction	-1082	MPa
23	Compressive Y direction	-100	MPa
24	Compressive Z direction	-100	MPa
25	Shear XY	60	MPa
26	Shear YZ	32	MPa
27	Shear XZ	60	MPa

Layer information

Layer	Material	Thickness (mm)	Angle (°)
(+Z)			
4	Epoxy Carbon UD (230 GPa) Prepreg	1	0
3	Epoxy Carbon UD (230 GPa) Prepreg	1	0
2	Epoxy Carbon UD (230 GPa) Prepreg	1	0
1	Structural Steel	4	0
(-Z)			

Composite of each layer with structure steel of 4 mm thickness and composite layer of each 1 mm is provided

Mesh



Statistics	
Nodes	5565
Elements	5348

Fig 16: Meshing of engine mounting bracket carbon fiber reinforcement

Boundary condition

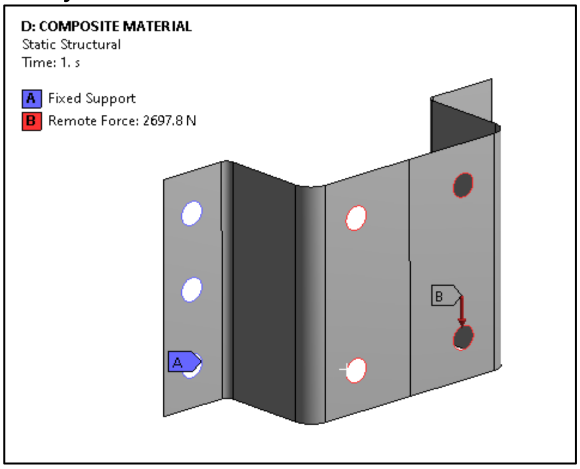


Fig 17: Boundary condition of engine mounting bracket carbon fiber reinforcement

Total deformation

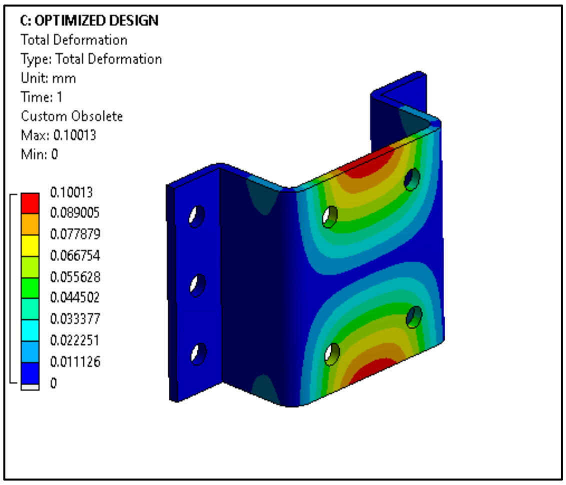


Fig 18: Total deformation of engine mounting bracket carbon fiber reinforcement

LAYER WISE EQUIVALENT STRESS ON CARBON FIBRE AT LAYER 1

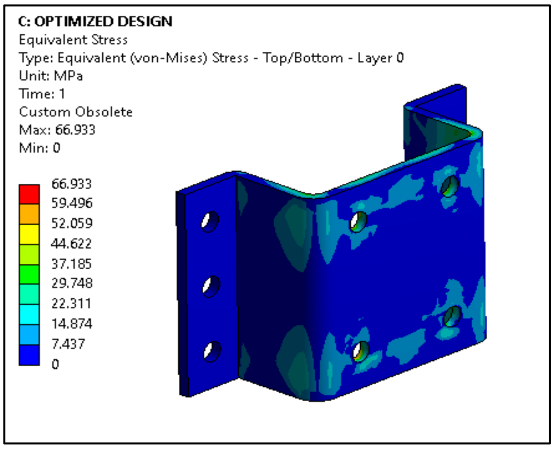


Fig 19: Equivalent stress at layer 1 of engine mounting bracket carbon fiber reinforcement

AT LAYER 2

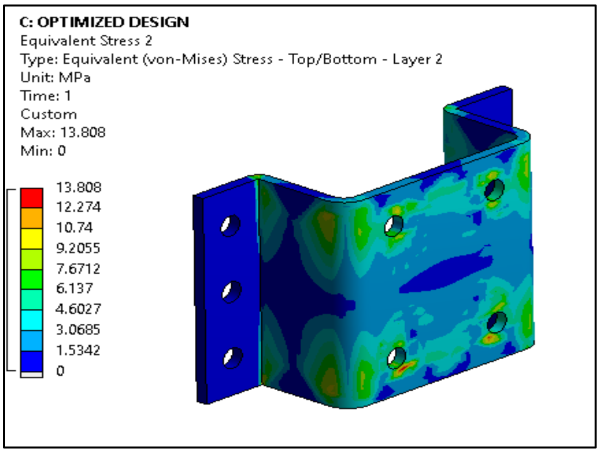


Fig 20: Equivalent stress at layer 2 of engine mounting bracket carbon fiber reinforcement

AT LAYER 3

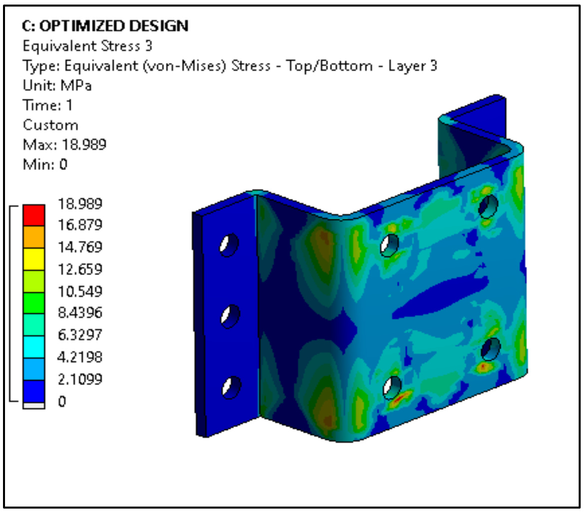


Fig 21: Equivalent stress at layer 3 of engine mounting bracket carbon fiber reinforcement

Equivalent Elastic Strain

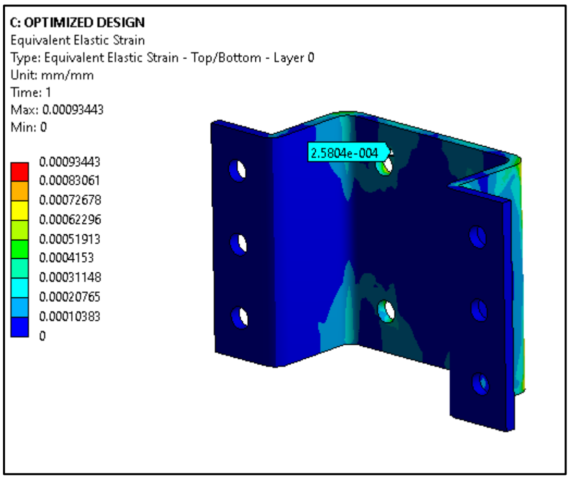


Fig 23: Equivalent Elastic Strain of engine mounting bracket carbon fiber reinforcement

8. EXPERIMENTAL SETUP

A universal testing machine (UTM), also known as a universal tester, materials testing machine or materials test frame, is used to test the tensile strength and compressive strength of materials. An earlier name for a tensile testing machine is a tensometer. The "universal" part of the name reflects that it can perform many standard tensile and compression tests on materials, components, and structures (in other words, that it is versatile). The set-up and usage are detailed in a test method, often published by a standards organization. This specifies the sample preparation, fixturing, gauge length (the length which is under study or observation), analysis, etc. The specimen is placed in the machine between the grips and an extensometer if required can automatically record the change in gauge length during the test. If an extensometer is not fitted, the machine itself can record the displacement between its cross heads on which the specimen is held. However, this method not only records the change in length of the specimen but also all other extending / elastic components of the testing machine and its drive systems including any slipping of the specimen in the grips. Once the machine is started it begins to apply an increasing load on specimen. Throughout the tests the control system and its associated software record the load and extension or compression of the specimen.

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 (L*W*H)	2100*800*2060
10	Weight	2300Kg

8.1 EXPERIMENTAL PROCEDURE

- 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.

Fig.23: Experimental result

From FEA results

Equivalent stress	Engine mounting bracket glass fiber reinforcement	Engine mounting bracket carbon fiber reinforcement
At Layer 1	71.11	66.93
At Layer 2	9.20	13.80
At Layer 3	11.69	18.98

Material and existing weight of model

Properties	
Volume	3.2549e+005 mm^3
Mass	2.5551 kg

S.S material with carbon fiber layer and optimized weight of model

Properties	
Total Thickness	7. mm
Total Mass	1.6679 kg

9. CONCLUSIONS

- From FEA result conclude that After applying 2697.8 N force on single engine mounting bracket, Equivalent stress on Engine Bracket was 39.414MPa
- Modal analysis of engine mounting bracket is performed to obtain different mode shapes and natural frequency of existing engine mounting bracket and also it is observed that maximum frequency is around 3870.4 Hz.
- From FEA analysis of Eengine mounting bracket glass fiber reinforcement and Engine mounting bracket carbon fiber reinforcement it concluded that equivalent stress induced in carbon fibre are less than glass fibre.
- Due to use of carbon fibre reinforcement engine mounting bracket total weight reduced up to 34.75%.
- Strain measurement of 258 microns and 281 microns by numerical and experimental testing respectively.

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