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FAILURE ANALYSIS AND OPTIMIZATION OF ROCKER ARM

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ABSTRACT

Reduction of weight and improving the performance of components is on the top priorities in the manufacturing sector. In this view, this paper presents the study on the use of an advanced S-Glass Fiber material for the rocker arm. The theoretical and finite element analysis of rocker arm of diesel engine was performed. The percentage of variation in maximum deformation, maximum Von-Mises stress and maximum shear stress is observed to be 4.71%, 1.78% and 3.12%, which is acceptable. The obtained values of maximum deformation, maximum Von-Mises stress and maximum shear stress for the S-Glass Fiber were comparatively smaller than the corresponding values for the existing material of alloy steel for the rocker arm. Therefore, the proposed material of S-Glass Fiber provides more strength to the rocker arm and hence this material is suggested as a future material for the rocker arm of diesel engines.

KEYWORDS: S-Glass Fiber, Failure Analysis, Rocker Arm, Stress, Deformation.

1. INTRODUCTION

Rocker arm is a mechanically advantaged lever which is pivoted at pivot point so that it can transmit camshaft motion to the valve. Jonathan Rundle Bacon created Rocker arms in the 19th century [1]. In the early phase of development, the pivot point was based on less efficient theories which led to wear of valve tips, valve guides and other valve train components. Throughout the history of the rocker arm, its function has been studied and improved upon. Some designs can actually use two rocker arms per valve, while others utilize a roller bearing to depress the valve. These variations in design result in rocker arms that look physically different from each other, though every rocker arm still performs the same basic function. Design engineers always aim at improvement in each and every part of automobile system. Automobile industry, since many years, is conducting constant efforts for the purpose of modification of the mechanical parts of vehicles in order to improve the performance. In addition, redesigning the mechanical parts play an important role in improving the sustainability of the system against the resultant stresses and strains, therefore, significant consideration should be taken for this when parts are designed by engineers. Failure of rocker arm is a measure concern as it is one of the important components of push rod IC engines. Present work is carried out stress analysis of rocker arm under extreme load condition. The failure of rocker arm makes engine useless also requires costly replacement designers are facing a lot of problems especially, stress concentration and effect of loads and forces and other factors. The finite element analysis (FEA) method is the most popular approach and found commonly used for analyzing fracture mechanics problems [2]. Therefore it needs to carry out a detailed FEA analysis work to study and calculate deformations and stresses in rocker arm to understand the failure modes.

Sai Krishna et al. [1] have studied different types of rocker arm from last few years. The rocker arm concern points such as new method to found the problems and their solution to control the failure. And the detailed history of rocker arms such as different types of rocker arm used in different engine along with various materials used to manufacture the rocker arm. Explanations behind Failure of rocker arm are additionally examined. V. R. Magdum et al. [2] are observing that how the stresses are varying by changing different materials of the rocker arm. Increase strength and durability of rocker arm is still subject of research and investigation. For this, they have modeled the arm using design software like CATIA and the stressed regions are found out using ANSYS software. Using different materials for construction of rocker arms is one of the best methods to increase its durability and strength. Tawanda Mushiri et al. [3] have been considered High Density Polyethylene (HDPE) composite rocker arm has for analysis owing to its light weight, higher strength and good frictional characteristics. A 3-D finite element analysis was carried out to find out the maximum stresses developed in the

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rocker arms made of cast iron and composite. With this it may be concluded that the stresses developed in the composite is well within the limits without failure. D. Vinay kumar et al. [7] have studied the structural analysis of the component with various material along with different forces acting on the system is considered. Where in this product, redesign the product along with it can be tested in different situation of forces with different boundary condition applied on the system. In this they covered all the theoretical calculation analysis in different condition with lots of load can be applied on the body and last one is experimentation of the whole system under different loads. And for the extra information they were going through different material analysis. All the analysis has been done in workbench 14 from it calculate the value of stress and compare with manual estimation. Sachin Bacha, P. Swaminadhan, D. Deshpande [9] have studied the analysis of bending stresses, shear stresses and deformations level in rocker arm body are carried out as per the ASME section code and it can be concluded that Finite Element analysis is required to match the results of analytical calculations. This study is not exhaustive and conducted by Finite Element analysis of a Rocker Arm using 3D modeling and post-processing for analysis of rocker arm with rocker shaft according to ASME codes.

1.1 Background

From the study it tends to be seen that the design of rocker arm has been a decent research subject for some analysts, because of its significant job in pressure examination in rocker arm. The scientists began from creating hypotheses identified with exhaustion disappointment of rocker arm and further moving to discovering different parameters as per their application [3]. We found that the rocker arm fails due to poor design of the rocker and due to heavy weight of the arm. Poor design of rocker gets fail due to maximum forces acting on the inlet or exhaust of the valve, which cannot hold the pressure of the force cause failure in structure of the rocker. High weight of the rocker arm need maximum force to lift the arm which affects on the speed of the system, speed can be low due to heavy weight of rocker. Here we have the information of cast iron, structural steel and different material has been used but the major problem of the rocker failure is the weight of the rocker where in all these materials has the same problem [7]. In all this condition, due to heavy weight of the rocker arm, it need maximum load to lift up the arm that means sudden loading cause the failure of the rocker arm. The cyclic loads on the fulcrum pin cause the failure at the circular section of the rocker where it produces maximum shear stress on the arm. So we need to improve a rocker arm with low weight which must have the high strength along with good life [11]. For that we need to use the composite material to reduce the weight of the rocker arm and to have maximum life of the rocker arm.

1.2 Objective

- To find the value of shear stress, von-mises stress, deformation parameter by using theoretical calculations for the proposed material (S-glass fiber).
- To find the value of shear stress, von-mises stress, deformation parameter by using finite element analysis software for the proposed material (S-glass fiber).
- Compare the finite element Analysis results with theoretical calculation.

2. MATERIALS AND PROPERTIES

S-implies structural glass and it is known for its mechanical properties. Glass fiber is comprised of silica sand which softens at 1720 degree Celsius; SiO_2 is additionally the essential component quartz normally happening stone [2].
Composition of S Glass fiber: carbon fiber (graphite), quartz polyester.

There are different types of glass fiber from that by comparing the existing material we select the S-Glass fiber the other information is as follows,

- S-glass fiber is the unidirectional mat that all the fiber runs in the same direction. But S- Glass fiber gets three dimension strength and stiffness.
- It has a particular resistance more prominent than the steel, so it is utilized to make high performance.
- It has the properties of low thermal conductivity making it profoundly utilized in building businesses.
- It can ready to keep up high strength over the wide scope of condition.
- It has the ability of high stiffness compare with other material.
- It has the low density than the existing material which helpful in the weight of the component.

The properties of S-Glass Fiber material appeared in Table-1 is as follows,

Table-1: Properties of S-Glass Fiber Material

Material	Youngs Modulus (GPa)	Poissons Ratio	Density kg/m^3
S-Glass Fibre	88.9	0.23	2500

3. METHODOLOGY
Analytical Design calculations of the rocker arm.
- Creating of 3-D Model by using Solid work Software.
- Finite Element Analysis (FEA).
- Comparison of Theoretical Analysis and FEA results.

4. ENGINE SPECIFICATION

Table-2: Engine Specification

Type	Air cooled, diesel engine
No. of Cylinder	1
Bore / Stroke	97mm × 100mm
Max. Engine Output	5.07 kW at 3000 rpm
Max. Torque	15 Nm at 2500 rpm

5. THEORETICAL ANALYSIS OF ROCKER ARM
The various forces act on the rocker arms during the operation. These forces can be mainly categorized into:
- Gas load on valve, which comes into play when valve opens.
- Inertia force, which oppose the downward movement of the valve.
- Initial Spring force, which hold the valve on its seat against the suction.

Let m = Mass of the valve,
d_v = Diameter of the valve head,
h = Lift of the valve,
a = Acceleration of the valve,
P_c = Cylinder pressure or back pressure when the exhaust valve opens, and
P_s= Most extreme suction pressure,
We know that speed of engine 3000 RPM,
The speed of camshaft = N/2
= 3000/2
= 1500 RPM.
Time taken for the valve to open and close,
t = Angle of action of cam
Angle turned by camshaft of 110,
= 110/9000
t = 0.012 s.
We have, maximum acceleration of the valve,
a =1780.2 m/s2.
And force due to valve acceleration,
F_{va}= Mass of valve × Acceleration of valve
= m_v x a
= 0.09 × 1780.2 + 0.882
F_a = 161.1 N.
∴ Maximum load on the rocker arm for exhaust valve,

$F_e = P + F_s + F_{va}$
$= 503.282 + 24.238 + 161.1$
Fe = 688.62 N.
It may be noted that maximum load on the rocker arm for inlet valve is,
$F_i = F_s + F_{va}$
Since the maximum load on the rocker arm for exhaust valve is more than that of inlet valve, therefore, the rocker arm must be designed on the basis of maximum load on the rocker arm for exhaust valve.
Design of the cross-section (I-section) of the rocker arm is as follows,
The thickness of the web is calculated from the equation,
$F \times l = \sigma \times Z$
F=Force due to exhaust gases, N.
l= effective length of a rocker arm, mm.
σ = allowable stress, N/mm^2
Z= section modulus
The thickness of the web is (t) = 8mm.
Width of flange= 2.5t= 2.5 × 8=20mm.
Depth of web = 4 t= 4 × 8 = 32 mm.
And depth of the section = 6t = 6×8= 48 mm.
The force due to valve acceleration because of valve speeding up (Fa) might be gotten as talked about underneath,
Valve diameter = dv = 40 mm.
Lift of valve = h = 13 mm.
Cylinder pressure = Pc = 0.6 N/mm^2.
Max. Suction pressure = Ps = 0.02 N/mm^2
Diameter of fulcrum pin = d1 = 24 mm.
Boss diameter = D1 = 35 mm.
Speed of engine = N = 3000 rpm.
Angle of action of cam = Φ = 110°
Angle between two arms = ϴ = 176°
Volume of rocker arm =V = $3.71 \times 10^{-5} m^3$
Distance between shaft centre to pin centre = L = 27.37 mm.
Width of rocker arm = b = 25 mm.
Distance from neutral axis = d = 6.57 mm.
Vertical distance from neutral axis = y = 6.5 mm.
Density of material = ρ in kg/ m^3
Mass of material = m in kg.
Weight of rocker arm= W in N.
Acceleration due to gravity = g = 9.81 m/ s^2

- **Calculations for S-Glass Fiber material of rocker arm**

We have to find different stress acted on rocker arm. For that we use material S-Glass Fiber which has density, ρ = 2500 kg/ m^3

a. Mass of rocker arm,

$m = V \times \rho = 3.71 \times 10^{-5} \times 2500 = 0.09275$ kg.
Weight of rocker arm,
$W = V \times \rho \times g$
$= 3.71 \times 10^{-5} \times 2500 \times 9.81$
W = 0.90 N. …(a)

b. Calculating forces,

Gas load on valve,
$P1 = \frac{\pi}{4} \times (dv)^2 \times Pc$
$= \frac{\pi}{4} \times (40)^2 \times 0.6$
P1 = 753.98 N. …(b)

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Total load on valve,
P = P1 + W
= 753.98 + 0.90
P = 754.88 N. … (c)
Initial spring forces (considering weight of valve),
$Fs = \frac{\pi}{4} \times (dv)^2 \times Ps - W$
$= [\frac{\pi}{4} \times (40)^2 \times 0.02] - 0.90$
Fs = 24.22 N. …(d)
We have,
Speed of camshaft, S= 1500 rpm.
Angle turned by camshaft per second, A = 9000 degree/ sec
Time for valve to open & close, t = 0.012 sec.
Max. Acceleration of valve, a = 1782.01 m/s^2
Force due to valve acceleration,
$F_a = (m \times a) + W$
$= (0.292 \times 1782.01) + 0.90$
F_a = 166.181 N. …(f)
Max. load on Rocker arm for exhaust valve is,
$Fe = P + F_s + F_a$
= 754.88 + 24.22 + 166.181
Fe = 945.281 N. …(g)
Two arms of rocker arm is equal, so forces on them are also equal i.e. Fe = Fc

a. Calculating stresses,

Bending stress near critical limit is,
$\sigma_b = ((F_e) \text{ x } (L-(D_1/2) \text{ x } (y))/ ((b \text{ x } d^3)/12))$
$= ((945.281) \text{ x } (27.37-(35/2) \text{ x } (6.5))/ ((25 \text{ x } 6.57^3)/12))$
σ_b = 43.590 N/mm^2
Shear stress at critical limit is,
$\tau = \sqrt{(\sigma_b/2)^2 + (F_e/b \text{ x } d^2)}$
$= \sqrt{(43.590/2)^2 + (945.281/25 \text{ x } 6.57^2)}$
τ = 61.540 N/mm^2

b. Von-misses stress

Von Misses stress is,
Forces acted only in X direction so,
$\sigma_b = \sigma_x$ & $\sigma_y=0$
$\sigma_1, \sigma_2 = ((\sigma_x + \sigma_y)/2) \pm \sqrt{((\sigma_x + \sigma_y/2)^2 + (\tau)^2)}$
$= ((43.590 + 0)/2) \pm \sqrt{((43.590 + 0/2)^2 + (22.540)^2)}$
σ_1 = 53.140 N/mm^2
σ_2= -9.460 N/mm^2
Von-misses stress is given by,
$\sigma_v = \sqrt{\{[\sigma_1]^2 - [(\sigma_1) \times (\sigma_2)] + [\sigma_2]^2 + [(3) \times (\tau)^2]\}}$
$\sigma_v = \sqrt{\{[53.140]^2 - [(53.140) \times (-9.46)] + [-9.46]^2 + [(3) \times (61.540)^2]\}}$
σ_v = 121.562 N/mm^2
We have,
Inertia of section

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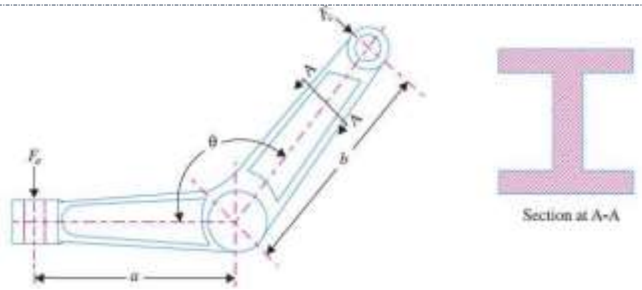

Fig-1: I-section of the rocker arm

Inertia of section A-A,
$I = [(B_1 \times H_1^3) + (B_2 \times H_2^3)]/12$
$= [(9.956 \times 25^3) + (20.18 \times 5^3)]/12$
$I = 12658.12\ mm^4$

a. Max. Deformation

Max. Deformation is,
$\delta max = F_e \times l^3 / (3 \times E \times I)$
$\delta max = (945.281 \times 23^3) / (3 \times 88.9 \times 10^3 \times 12658.12)$
$\delta max = 0.0034\ mm.$

6. FINITE ELEMENT ANALYSIS

Finite element analysis of rocker arm is carried out as per following steps,

Modeling of Rocker Arm

The detailed dimension of diesel engine rocker arm which is used in marine application are given below,

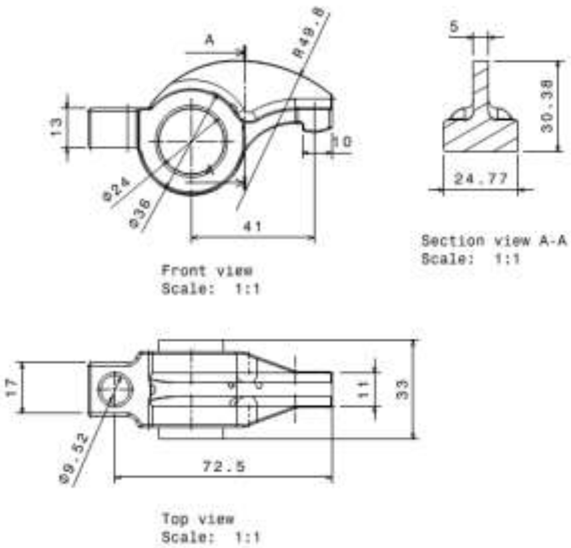

Fig-2: Dimensions of Rocker Arm

Three dimension (3D) CAD model is created using solid works 16. Software. The model is then exported in STEP (Standard for exchange of product data) format and it is then imported in ANSYS for further processing.

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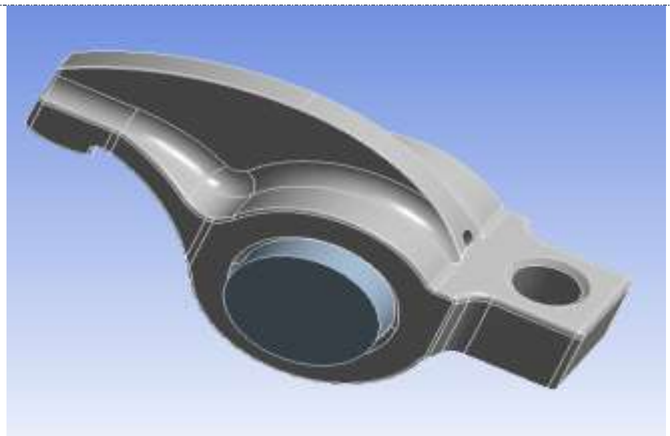

Fig-3: CAD model of Rocker Arm

Processing

In this step, load and fixed support are applied to the model. As theoretical analysis, the design of rocker arm should carry a force of 945.28 N in –Y direction. But to analyze modified rocker arm in comparison with the existing design, the boundary condition used must be same and hence 1376.40 N of force has been applied which is calculated for existing design.. The Rocker arm is fixed at the valve rod end and valve force is applied at the tip. The shaft is fixed at both ends and frictionless contact is defined between rocker arm and shaft is indicated in the figure 4,

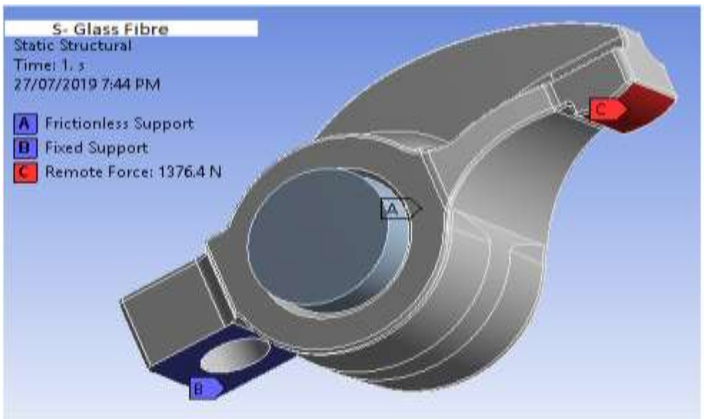

Fig-4: Applied Force and fixed support in Rocker Arm

Post processing

This is the last step in a finite element analysis. With the load of existing material of 1376.4 N by applying it on the proposed material, we got the result of deformation, Shear Stress and Von-Mises Stress**,** which is acceptable. And the deformation, Shear Stress and Von-Mises Stress of rocker arm is indicated in the figures.

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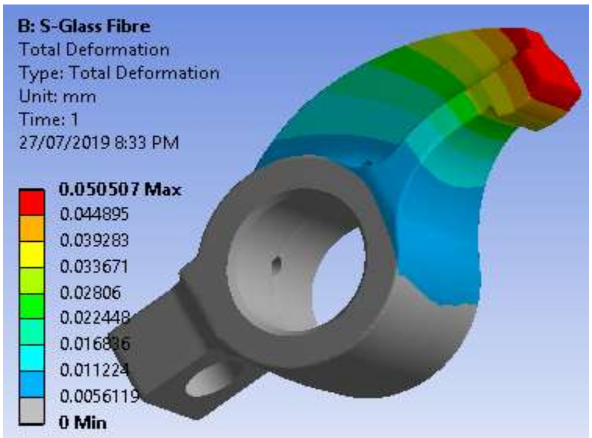

Fig-5: Total Deformation

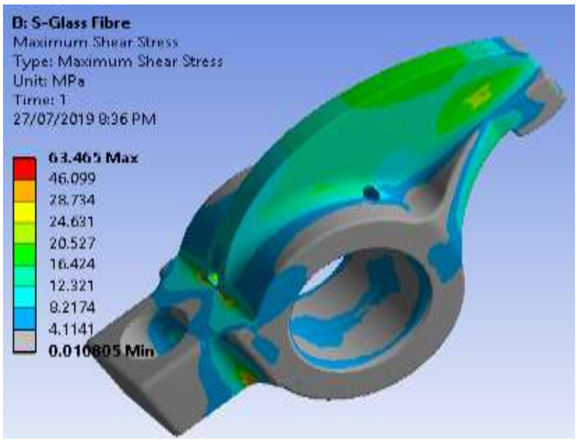

Fig-6: Shear Stress of Rocker Arm

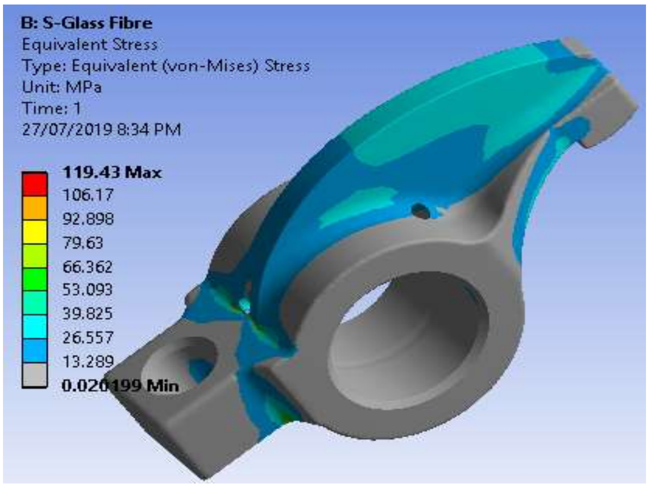

Fig-7: Von-Mises Stress

Summary of the Theoretical and Finite element analysis is given in table-3**,**

Table-3: Summary of results

Method	Maximum deformation (mm)	Max. Von-Mises stress (MPa)	Max. shear stress (MPa)
Theoretical calculation	0.0034	121.562	61.540
Finite element method	0.0505	119.430	63.465
Percentage of variation (%)	4.71	1.78	3.12

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Table-3: Comparison of results

Rocker arm material	Maximum deformation (mm)	Max. Von-Mises stress (MPa)	Max. shear stress (MPa)
Proposed material of S- glass fiber	0.0505	119.430	63.465
Rocker arm of alloy steel (Manufacturer data)	0.0843	164.153	94.55

7. CONCLUSION

The percentage of variation in maximum deformation, maximum Von-Mises stress and maximum shear stress is 4.71%, 1.78% and 3.12%, which is acceptable. Thus the results of analysis of rocker arm are validated. The values of maximum deformation, maximum Von-Mises stress and maximum shear stress for the S-glass fiber were comparatively smaller than the corresponding values for the existing rocker arm of alloy steel. Therefore the proposed material of glass fiber provides more strength to the rocker arm and hence this material is suggested for the rocker arm of diesel engines.

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