

# Numerical Simulation and Experimental Verification of Notched Axle Shaft for All-Terrain Vehicles



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## ABSTRACT

Notched axle shaft is used in most of the all-terrain vehicles (ATV). The purpose of this shaft is to transmit torque from differential to the wheel. So, due to the purpose and the position of the shaft, it is subjected to combined loadings at the same time i.e. bending load due to suspension and the torsion at the wheel side. Now, as these shafts are having notch on it due to some design specifications, the failures mostly occurs at the notches. There were various material used for these shaft till the date. Previously, these shafts were manufactured in Mild Steel and were straight tapering shapes. Later, they were modified into notched shapes for high power ATVs. Even after making lot of changes in its design, there was still a problem of its weight. Thus, the main objective of this study is to reduce the weight of the existing steel notched axle shaft by changing its material. The material selected for the shaft is Aluminium 7075-T6. For testing the shaft, structural analysis on the shaft is performed and also the fatigue analysis of the shaft is done. Analysis is performed mathematically, simulation on software and experimentally. In mathematical and software analysis, combined bending and torsion loading are considered. For experimental testing, only bending load is considered.

**Keywords—** Aluminium 7075-T6, ATV, Bending and torsion loading, Notched axle shaft, Simulation.

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## I. INTRODUCTION

An all-terrain vehicle (ATV) is defined as vehicle that travels on low-pressure tires with a seat that is straddled by the operator along with handle bars for steering control. ATV is also known as quad bike or four-wheeler. As the name implies, it is designed to handle a wider variety of terrain than most other vehicles. In most countries ATVs are not allowed on normal roads. ATV is intended for use by a single operator, although some companies have developed ATVs intended for use by the operator and one passenger.

ATVs are generally used on off roads where it has to bear tremendous amount of load while in its operation. Thus, it has a great effect of components weight on it. So, weight reduction becomes a major criterion while designing an ATV. Weight reduction implies reducing the overall weight of the vehicle by doing some modification in the various parts of the vehicle such as changing the design of the parts

or by changing the material. Rapid technological advances in engineering design field result in finding the alternate solution for the conventional materials. The design engineers brought to a point to finding the materials which are more reliable than conventional materials. Researchers and designers are constantly looking for the solutions to provide stronger and durable materials which will answer the needs of fellow engineers.

An ATV is an assembly of various components like tires, wheels, axles shafts, suspensions systems, transmission shafts, engine unit, chassis, seat, handle bar, etc.. Weight reduction is possible in various parts out of which, in this study, it is focused on axle shaft. There are varieties of axle shaft used in ATVs. Shafts can be single or double. Generally for low power engines single shafts are used and

for high power engines two shafts on either side of the differential are used.

The shape of the shaft has a great influence on handling of these vehicles. Previously, the shape of the shaft was tapering that is larger diameter on differential side which gets reduced towards the wheel hub and vice versa. Later these shafts were modified to notched shaft which gave more tensile strength along with reduction in weight of the vehicle. But it is found that this specification is not enough. So, changing the material of the shaft seems to be another option.

Now, these shafts are subjected to combined bending and torsional loads during operations in service. These complex loadings are defined as multi-axial loadings. Also, the shaft has geometric discontinuities which cause significant stress concentrations. Multi-axial loading paths produce complex stress and strain states near notches and can cause failure even without any evident large scale plastic deformation. Therefore, it is absolutely necessary to conduct durability analysis for notched components during design process. Due to the fact that notch regions are under the effect of multi-axial stress state, the strength and durability estimations of notched components subjected to multi-axial loading paths requires detail knowledge of stresses and strains in such regions.

In this study, a 350 cc ATV is considered. It is a four valve, four strokes, petrol engine ATV. In modern ATVs the axle shafts are having notches. The notches have considerable reduction in weight along with better tensile strength. But as this vehicle is designed for all terrain use, it has to bear tremendous amount of the load which leads to generation of stresses on the notched portion. Thus due to generation of stresses, fatigue failure of the shaft at the notches is a common sight.

Previously, the axle shafts were manufactured in Mild Steel and were straight tapering shapes. Later, they were modified into notched shapes for high power ATVs. Even after making lot of changes in its design, there was still a problem of its weight. Thus, the main objective of this study is to reduce the weight of the existing steel notched axle shaft by changing its material. The material selected for the shaft is aluminium 7075 series alloy. Specifically, aluminium 7075-T6 material is selected.

In this study, analysis on the shaft of both materials is done in three ways as mathematically, simulation on software and experimentally. In mathematical calculations, for modelling, exact dimensions of the shaft are measured from a 350 cc ATV having capacity of two persons. Inputs are considered as combined bending and torsion loading. Shear force diagram and bending moment diagram are used to find the critical area on the shaft. Also, mean stresses and alternating stresses along with the mean torque and alternating torque is calculated. Further endurance limit for both the material is found. Using *Modified Goodman Diagram*, the factor of safety is found out and the life of shaft for both the material is estimated.

In software, again the inputs are considered as combined bending and torsion loading. Using the same dimensions as in mathematical, model of the shaft is prepared. Modelling, meshing and structural analysis is done in software simulation. The von mises stresses, strain, deflection, factor

of safety and life of the shaft for both materials are found out.

For experimental testing, the machining of the shaft of new selected material and the design and fabrication of fixture is done. Only bending load is applied on the shaft for experimental results. Critical area of failure of the shaft with new material is found out and also the number of cycles after which shaft fails is estimated. For validation, software results and experimental results of the new material are compared.

## II. DESIGN AND ANALYSIS OF NOTCHED AXLE SHAFT

In this study, a 350 cc ATV is considered. It is a four valve, four strokes, petrol engine ATV. It produces maximum power of 14 KW @ 6500 rpm.



Fig.1 Actual 350 CC ATV

Special feature of this ATV is that it has individual suspensions on both front and rear side. Macpherson Strut type of suspension system is used. Due to its independent rear suspension, two individual axle shafts are used. It is having Continuously Variable Transmission (CVT) with three gears; high, low and reverse. The vehicle has a top speed of 60 kilo meters per hour. Capacity of this vehicle is one driver and one passenger.

The dimensions of the shaft required for analysis are measured from a 350 cc ATV. The axle is a notched axle with 450 mm length and 30 mm highest diameter. The position of this axle is on the right side of the differential when viewed from behind. The left portion of the shaft is fixed to the differential and right portion at the wheel hub. All dimensions are taken in millimetre.

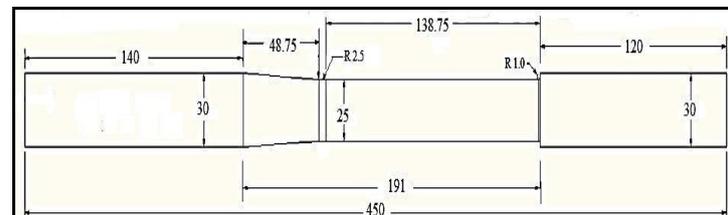


Fig. 2 Front View of Complete Geometry with All Dimensions in (mm)

TABLE I  
Material Properties

	C4140 Steel	Aluminium 7075-T6
Density ( $\rho$ )	7850 Kg/m <sup>3</sup>	2810 Kg/m <sup>3</sup>
Young's Modulus (E)	210 GPa	71.7 GPa
Poisson's Ratio	0.3	0.33
Fatigue Strength	146 MPa	159 MPa

I.

A. Numerical Approach

Weight of the Vehicle = 450 Kg  
 Rider Weight = 70 Kg  
 Passenger Weight = 70 Kg  
 Total Weight = 450 + 70 + 70  
 = 590 Kg

Therefore,

Max. weight on one axle = 590 / 2  
 = 295 Kg

i.e. Bending Load on one axle = 295 × 9.81  
 = 2893.95 N

Thus,

Maximum Bending Load = 2893.95 N

Now, length of the shaft where load is applied is 120 mm.

Thus, bending load on one axle per unit area  
 = 2893.95 / 120  
 = 24.116 N/mm

At minimum loading condition that is when vehicle is running on smooth road, all the loads are absorbed by the suspension system itself. Therefore, minimum bending load on one axle is considered as zero. Thus minimum Bending Load = 0 N

The vehicle is having three gears as high, low and reverse. The top speed of the vehicle at low gear is 60 Km/hour and the maximum torque produced by the vehicle at low gear is 320 N-m.

Thus,

Actual Torque = 320 × 10<sup>3</sup> N-mm

B. Shear Force and Bending Moment Diagram

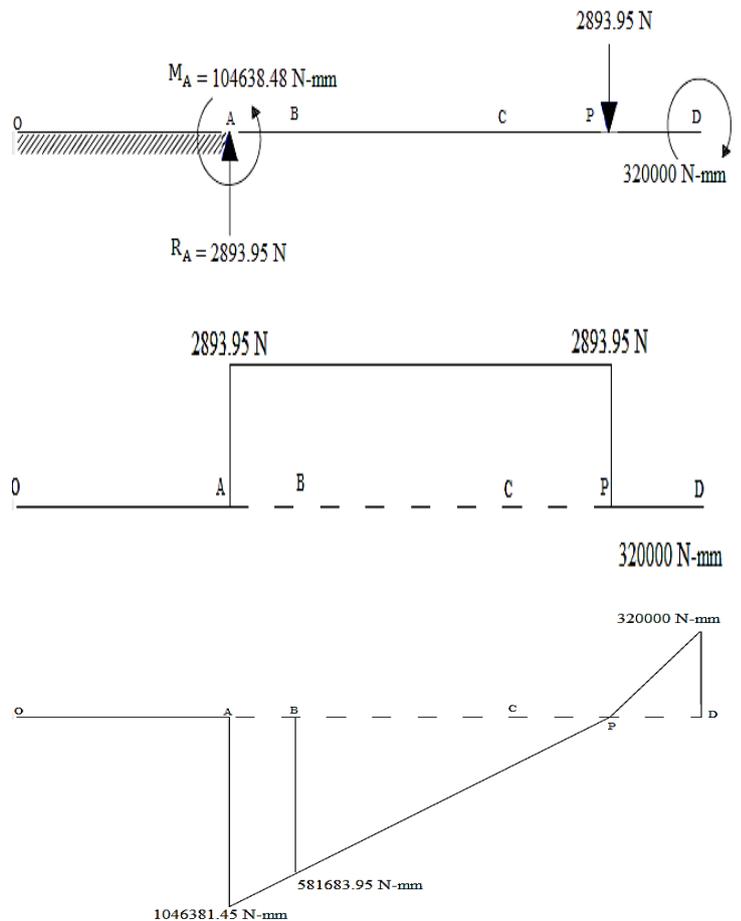
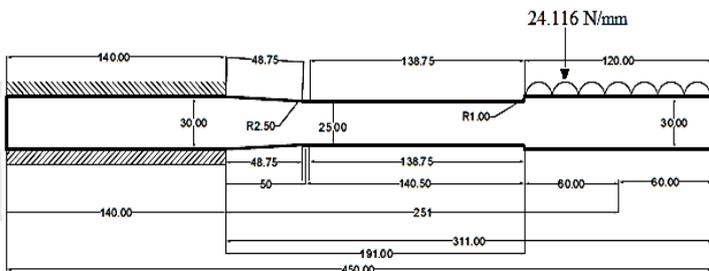


Fig. 3 SFD and BMD

From the shear force and bending moment diagram, region on the shaft where maximum stresses are occurring is observed. It is found that maximum stresses are occurring at point 'A'. Therefore failure will occur at point 'A'. But on the right hand side of point 'A' there is sudden drop in cross section area, to failure actually occurs at point 'B'. Thus maximum bending moment at point 'B' is -581683.95 N-mm and minimum bending moment at point 'B' is 0 N-mm.

C. Mean Stresses ( $\sigma_m$ ) and Alternating or Amplitude Stresses ( $\sigma_a$ )

Considering stresses to be completely reversed, mean stresses on the shaft can be calculated by equation,

$$\sigma_m = \frac{\sigma_{max} + \sigma_n}{2} \dots\dots\dots [1]$$

$$= 189.6 \text{ N/mm}^2$$

where,  $\sigma_m$  is mean stress,  $\sigma_{max}$  is maximum stress at critical point and  $\sigma_{min}$  is minimum stress at critical point.

$$\sigma_a = \frac{\sigma_{max} - \sigma_n}{2} \dots\dots\dots [2]$$

$$= 189.6 \text{ N/mm}^2$$

where,  $\sigma_a$  are alternating stresses.

D. Mean Torque ( $\tau_m$ ) and Alternating Torque ( $\tau_a$ )

$$\tau_m = \frac{\tau_{max} + \tau_{min}}{2} = 52.152 \text{ N/mm}^2 \quad [3]$$

where,  $\tau_m$  is mean torque,  $\tau_{max}$  is maximum torque and  $\tau_{min}$  is minimum torque.

$$\tau_a = \frac{\tau_{max} - \tau_{min}}{2} = 52.152 \text{ N/mm}^2 \quad [4]$$

where,  $\tau_a$  is alternating torque.

The resultant mean stresses and resultant alternating stresses can be expressed as

$$\sigma_{Rm} = \sqrt{(\sigma_B)_m^2 + 3(\tau_m)^2} = 210.02 \text{ N/mm}^2 \quad [5]$$

where,  $\sigma_{Rm}$  is resultant mean stress.

$$\sigma_{Ra} = \sqrt{(\sigma_B)_a^2 + 3(\tau_a)^2} = 210.02 \text{ N/mm}^2 \quad [6]$$

where,  $\sigma_{Ra}$  is resultant alternating stress.

1) *Endurance Limit:* The endurance limit of the material can be expressed by using following equation

$$S_e = K_a \times K_b \times K_c \times K_d \times K_e \times K_f \times K_g \times S'_e = 174.34 \text{ N/mm}^2 \quad [7]$$

Where,  $S_e$  is endurance limit of material,  $K_a$  is surface finish factor,  $K_b$  is size factor,  $K_c$  is load factor,  $K_d$  is temperature factor,  $K_e$  is modifying factor for load concentration,  $K_f$  is fatigue stress concentration factor,  $K_g$  is reliability factor and  $S'_e$  is endurance limit of standard material.

2) *Modified Goodman Diagram*

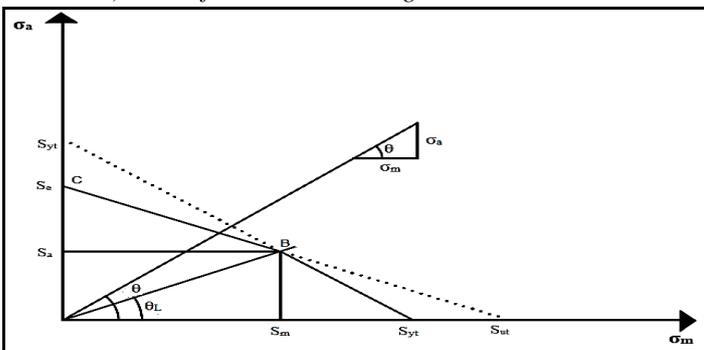


Fig. 4 Modified Goodman Diagram

Equation of line AB

$$\frac{S_m}{S_{yt}} + \frac{S_a}{S_{yt}} = 1 \quad [8]$$

Used if  $\theta < \theta_L$

Equation of line BC

$$\frac{S_m}{S_{ut}} + \frac{S_a}{S_e} = 1 \quad [9]$$

Used if  $\theta > \theta_L$

As point 'B' is intersection of two lines, At point 'B'

$$\frac{S_m}{S_{yt}} + \frac{S_a}{S_{yt}} = \frac{S_m}{S_{ut}} + \frac{S_a}{S_e} \quad [10]$$

Therefore, putting values in above equation, we obtain value of the ratio  $\frac{S_a}{S_m} = 0.2654$

Now,

$$\theta_L = \tan^{-1} \left( \frac{S_a}{S_m} \right) \quad [11]$$

$$\theta = \tan^{-1} \left( \frac{\sigma_a}{\sigma_m} \right) \quad [12]$$

From above results, it is observed that  $\theta > \theta_L$

Therefore,

Equation of the line BC be used

$$\text{i.e. } \frac{S_m}{S_{ut}} + \frac{S_a}{S_e} = 1$$

thus,

$$\frac{\sigma_m}{S_{ut}} + \frac{\sigma_a}{S_e} = \frac{1}{N_f} \quad [13]$$

Therefore,

$$\text{Factor of Safety } (N_f) = 0.732$$

Hence, the material will have finite life cycles.

3) *Life of Shaft*

On the curve, values of  $\log_{10} (0.9 \times S_{ut})$ ,  $\log_{10} (S_f)$  and  $\log_{10} (S_e)$  is calculated where  $S_{ut}$  is ultimate tensile strength,  $S_f$  is mean stress and  $S_e$  is endurance limit.

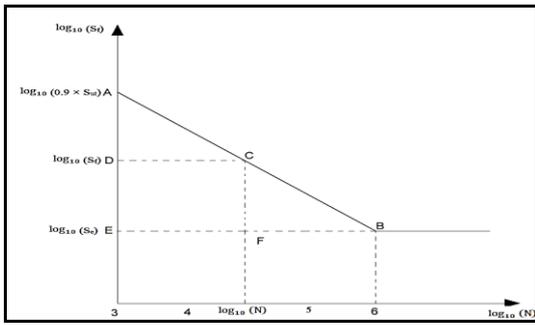


Fig. 5 S-N Curve

Putting all the values in following equation, we obtained the life of the shaft

$$\frac{\log_{10}(S_f) - \log_{10}(S_e)}{6 - \log_{10}(N)} = \frac{\log_{10}(0.9 S_{ult}) - \log_{10}(S_e)}{6 - 3} \dots\dots\dots [14]$$

= 92,172 Cycles.

Similar process is applied for the other material selected in this study. Analytical results obtained on the shaft are shown in following table.

TABLE II  
Analytical Results

	C4140 Steel	Aluminium 7075-T6
<b>Resultant Mean Stresses or Von Mises Stresses</b>	<b>210.02 MPa</b>	<b>210.02 MPa</b>
<b>Critical Area</b>	<b>188.75 mm (From Left End)</b>	<b>188.75 mm (From Left End)</b>
<b>Endurance Limit</b>	<b>174.34 MPa</b>	<b>152.25 MPa</b>
<b>Factor of Safety</b>	<b>0.732</b>	<b>0.634</b>
<b>Life of Shaft</b>	<b>92,172 Cycles</b>	<b>81,283 Cycles</b>

*E. Finite Element Analysis*

The Finite Element Method (FEM) is a technique in which a given domain is represented as a collection of simple domains called finite elements. FEM is a very powerful technique for determining the stresses and deflections in structures which are too complex to analyze by analytical methods. FEM is a very effective technique for solving complex problems with less time consumption and more cost saving. Because of FEM engineers do not have to go for experimental results directly. Also, while designing any new object, lot of iterations are sometimes necessary. This leads to lot of time wastage and cost. So, FEM helps in reducing wastage of time and money. Finite element procedures at present are widely used in engineering analysis. The procedure can be used in almost every field like solids, structures, heat transfer, fluids, etc..

While solving any problem by finite element method, few steps are to be followed. Firstly, the model of the object or assembly is to be created and the material properties are to

be assigned to it. Secondly, meshing has to be done on the object. After meshing is done, boundary conditions are applied on the object. Finally, the results are calculated. Even though the FEM do not provide results with exact accuracy, it gives results with minimum error which can be used in further designing of the object or work piece.

In this study, ANSYS Workbench software is used in analysis on the notched axle shaft. Creation of model, meshing and obtaining results is all done in same software. Now, in this study, the analysis is to be done on two different shafts having different material. So, for both materials, modeling, meshing and all required processes are done separately.

Before going for modeling of both the shafts, material properties are applied on the shafts. Material properties are different so separate analysis on both the material is done. All the material properties are assigned in the same software and the results are obtained.

- 1) *Modeling*: 3D model of the shaft is created in the ANSYS. The dimensions are taken from the 2D model created during numerical solution. The model is same for shaft with both materials.

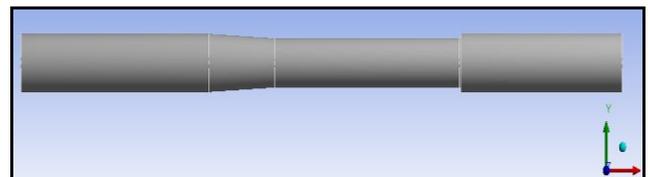


Fig. 6 Modeled shaft

- 2) *Meshing*: Mesh generation is the practice of generating a polygonal or polyhedral mesh that approximates a geometric domain. The term "grid generation" is often used interchangeably. Typical uses are for physical simulation such as finite element analysis. Three-dimensional meshes created for finite element analysis need to consist of tetrahedral, pyramids, prisms or hexahedra. If the accuracy is of the highest concern then hexahedral mesh is the most preferable one. The density of the mesh is required to be sufficiently high in order to capture all the flow features but on the same note, it should not be so high that it captures unnecessary details of the flow, thus burdening the CPU and wasting more time. Whenever a wall is present, the mesh adjacent to the wall is fine enough to resolve the boundary layer flow and generally quad, hex and prism cells are preferred over triangles, tetrahedrons and pyramids. Quad and Hex cells can be stretched where the flow is fully developed and one-dimensional. Based on the smoothness and aspect ratio, the suitability of the mesh can be decided.

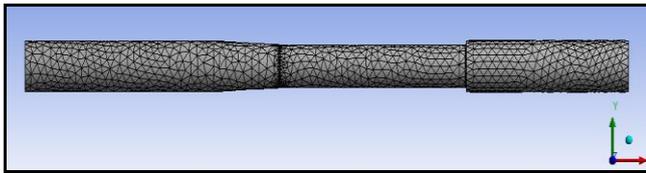


Fig. 7 Meshed shaft

In this study, meshing is done in ANSYS Workbench. Triangular mesh is selected. As the geometry is symmetric, triangular mesh is suitable for the analysis. While meshing, fine meshing is selected. This gives better energy flow around the notches.

- 3) *Boundary Conditions:* On the shaft, three boundary conditions are to be applied. First the shaft is fixed at left end. The length of the fixed support is 140 mm from the left end where all degrees of freedom are set zero. Second condition is combined bending and torsion loading at right end. Bending is applied on the right side for the length of 120 mm from the right end of the shaft. Bending force applied is 2893.95 N in downward direction. On the right face of the shaft, torque/moment in anti-clockwise direction is applied. Torque applied is 320 N-m.

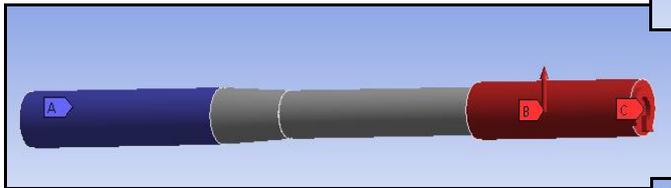


Fig. 8 Applied boundary conditions

- 4) *Results for C4140 Steel:*

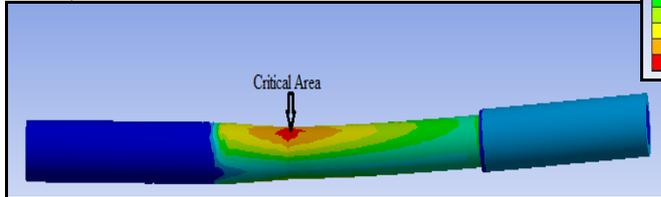


Fig. 9 Critical area

It is seen from the figure that most critical area on the shaft is at 188.75 mm from the left end. This is because of sudden change in cross section at that point

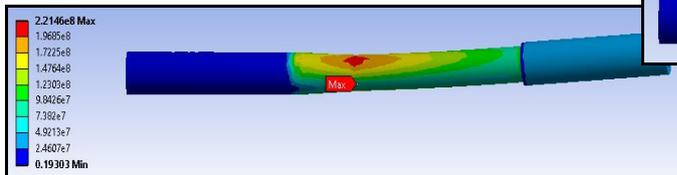


Fig. 10 Stresses

On the shaft, the combined loading is applied. The stresses obtained due to this loading are  $2.2146 \times 10^8 \text{ Pa} = 221.46 \text{ MPa}$ . These are the maximum stresses which are occurring on most critical area.

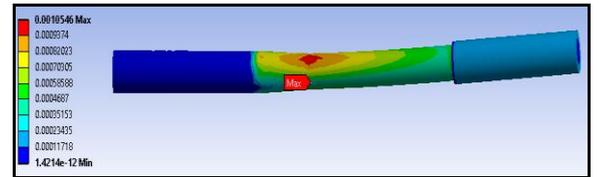


Fig. 11 Strain

Maximum strain on the shaft is 0.0010546. Also the maximum strain obtained is at the most critical area.

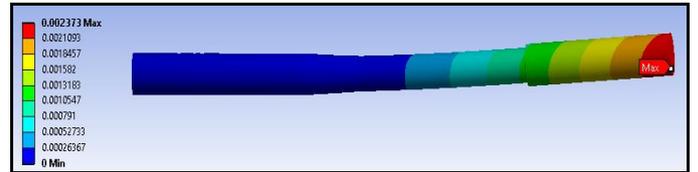


Fig. 12 Deflection

The maximum deflection of the shaft is 0.002373 m.

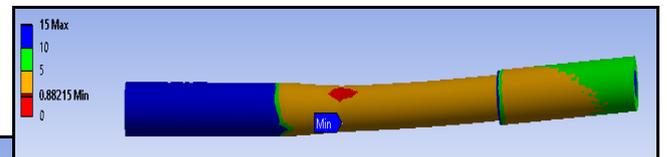


Fig. 13 Factor of safety

The factor of safety of the material is found as 0.88215

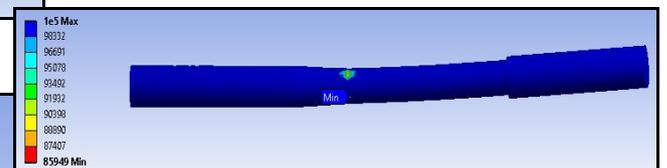


Fig. 14 life of shaft

The maximum number of cycles after which shaft failed are found to be 85,949 cycles.

- 5) *Results for Aluminium 7075-T6:*

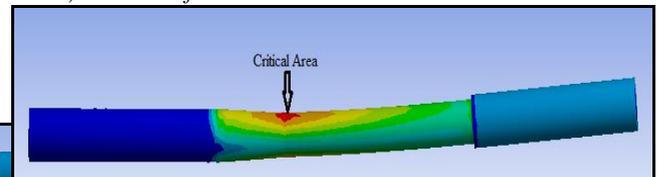


Fig. 15 Critical area

It is seen from the figure that most critical area on the shaft is at 188.75 mm from the left end. This is because of sudden change in cross section at that point.

### III. EXPERIMENTAL ANALYSIS OF NOTCHED AXLE SHAFT FOR BENDING LOAD

Experimental results are carried out on new material that is Aluminium 7075-T6. Only bending load is considered while calculating experimental results. For that first new material that is Aluminium 7075-T6 is analyzed again in the software. In the ANSYS, again model is created, meshing is done on it. While providing input, bending load of 2893.95 N is applied. Further for the same load, testing is done on the shaft. From software and experimental results, critical area on the shaft and the life of shaft are estimated.

#### F. Software Analysis

Model of the shaft is created in ANSYS Workbench software. Material properties are applied for Aluminium 7075-T6. Further meshing is done and bending load is applied on the shaft. Load is applied on right end of the shaft. The left side of the shaft is fixed. All degrees of freedom are set zero on the left side of the shaft. The shaft is fixed up to 140 mm from the left end.

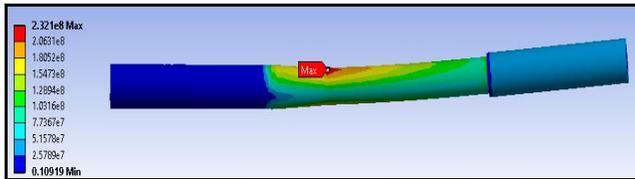


Fig. 16 Stresses

The stresses obtained due to combined loading are  $2.321 \times 10^8 \text{ Pa} = 232.10 \text{ MPa}$ . These are the maximum stresses which are occurring on most critical area.

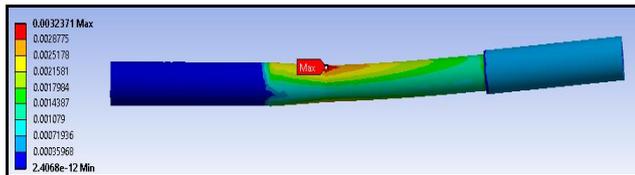


Fig. 17 Strain

Maximum strain on the shaft is 0.0032371. Also the maximum strain obtained is at the most critical area.

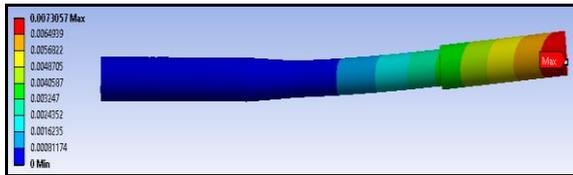


Fig. 18 Deflection

The maximum deflection of the shaft is 0.0073057m.

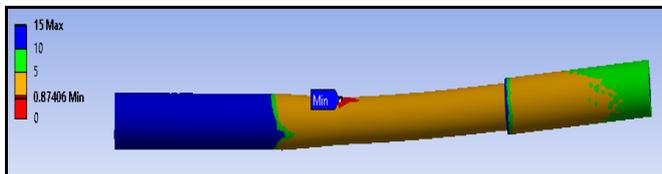


Fig. 19 Factor of safety

The factor of safety of the material is found as 0.87406.

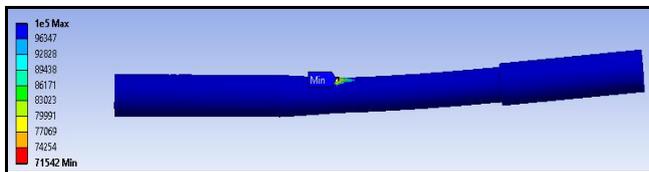


Fig. 20 Life of shaft

The maximum number of cycles after which shaft failed are found to be 71,542 cycles.

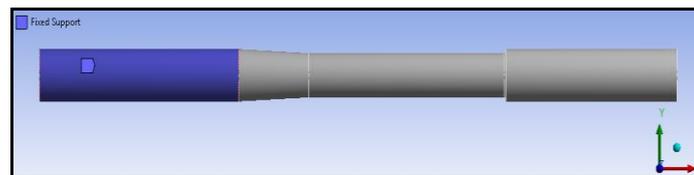


Fig. 21 Fixed Support

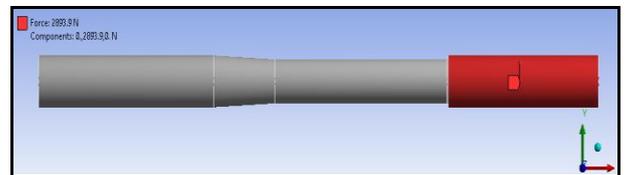


Fig. 22 Bending load

For same bending load (2893.95 N) and frequency of 1 Hz, critical area on the shaft is observed. The critical area is found at a distance of 188.75 mm from the left end of the shaft.

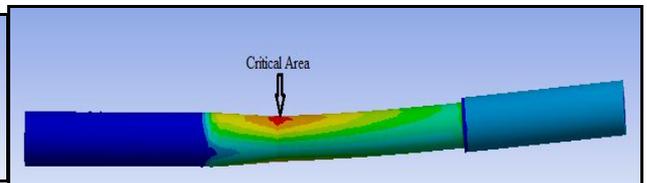


Fig. 23 Critical area

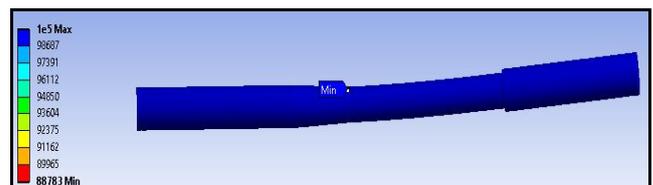


Fig. 24 Life of shaft

Also the life of the shaft is estimated. The shaft failed after 88,783 cycles.

### G. Shaft Machining

For the testing purpose, the two shafts were purchased from a local vendor. The material of the shaft purchased was Aluminium 7075-T6. The shaft purchased was of 480 mm length and its diameter was 32 mm. The shaft was purchased in bright metal so reducing the time required for finishing.

Further for the testing purpose, the shafts were machined as per required dimensions. The length of the shafts is machined to 450 mm. Diameter of the shaft is 30 mm. There is a tapering portion on the shaft at 140 mm from left hand side. After taper, there is a fillet of 2.5 mm radius and 138.75 mm long notch of 25 mm diameter. After notch the shaft is again having 30 mm diameter for 120 mm length. Thus maximum diameter on the shaft is 30 mm and minimum diameter is 25 mm.

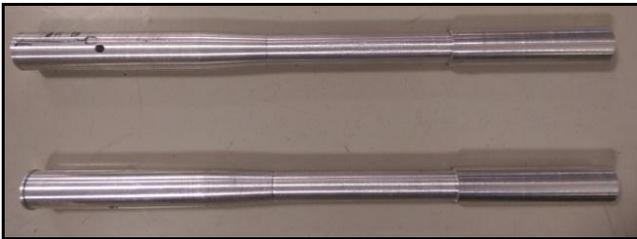


Fig. 25 Machined shaft

For testing the shaft, it was necessary to hold it as a cantilever so that force can be applied on the other end. For that a fixture stand was fabricated. The stand has a square plate on its base which is welded to long bar along with four triangular plates for support. The height of the stand is 900 mm from the ground. A hole of diameter 30 mm was drilled on it to pass the shaft from it. When the shaft is passed through the hole on the stand, a bolt is fixed to restrain its slipping and rolling movement while testing. A holding jaw is fabricated to apply load on the shaft. The holding jaw can be fixed on the shaft rigidly and a M40 bolt can be attached on it which is then fixed to the loading lever. The dimensions of the jaw were taken from a sample jaw provided by the lab. The weight of the jaw is 3.91 Kg. This weight was subtracted from the total weight applied by the machine.

The name of the machine used is *InstronHydropuls-Actuator*. The capacity of the machine is 10 tones that is it can apply load up to 10 tones. There were two loading lever, first has a loading capacity of 0 to 2.5 tones and the second has a loading capacity of 0 to 10 tones.

The whole fixture and shaft assembly was held on the machine bed. The holding jaw was attached to the loading lever with the help of a M40 bolt. The M40 bolt was fixed on the lever rigidly.



Fig. 26 Test setup

The load of 292 Kg was applied on the shaft in upward and downward direction. The frequency of 1 Hz was set that is one cycle was completed in one second.

The test was continued for 21 hours. The shaft failed after 79000 cycles. The failure occurred at a distance of 190 mm from the left end of the shaft.

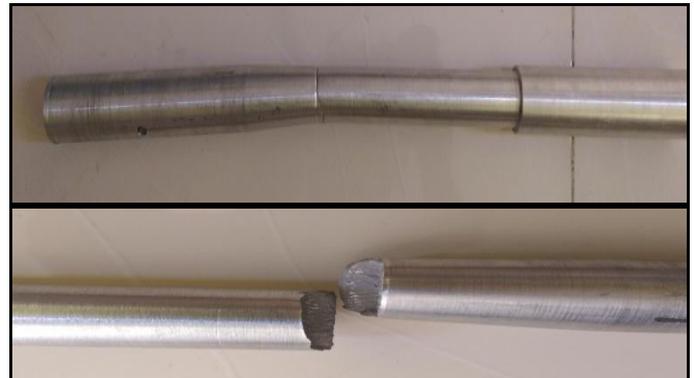


Fig. 27 Cracked shaft

TABLE III  
Results for Combined Loading

	C4140 STEEL		ALUMINIUM 7075-T6	
	Mathematical	Software	Mathematical	Software
<b>Critical Area</b>	Same	Same	Same	Same
<b>Resultant Mean or Von Mises Stresses</b>	210.02 N/mm <sup>2</sup>	221.46 N/mm <sup>2</sup>	210.02 N/mm <sup>2</sup>	232.10 N/mm <sup>2</sup>
<b>Factor of Safety</b>	0.732	0.88215	0.634	0.87406
<b>Life of Shaft</b>	92,172 Cycles	85,949 Cycles	81,283 Cycles	71,542 Cycles
<b>Strain</b>	—	0.001054	—	0.003237
<b>Deflection</b>	—	0.002373mm	—	0.007305mm

TABLE IV  
Results for Only Bending Load

	ALUMINIUM 7075-T6	
	Software	Experimental
<b>Critical Area</b>	Same	Same
<b>Life of Shaft</b>	88,783 Cycles	79000 Cycles

#### IV. CONCLUSION

The proposed material Aluminium 7075-T6 for the notched axle shaft used in the ATV's has been analysed in this study. The new material reduced the weight of the shaft by three times than the previously used material in ATV's.

Two different procedures had been used in this study for validating the results. Firstly, both materials that are C4140 Steel and Aluminium 7075-T6 are applied with combined loading (Bending and torsion) and solved for calculating stresses, endurance limit, factor of safety and life of shaft by both mathematical as well as software solution.

Thus, it was observed from the results that for combined loading both materials came out with comparable results.

Secondly, experimentation was done on new material Aluminium 7075-T6 with loading as bending only. For this, new material was first analyzed in software for only bending load and the results were obtained. For the same bending load the material had been tested in laboratory where life of the shaft was calculated.

The cycles obtained after testing the shaft in the laboratory are 79000 cycles. But these cycles are for maximum loading. In actual practice, only 20 percent of the maximum loading will act on the axle. So the cycles will be more than a million.

Now, the weight of the shaft with old material (C4140 Steel) was around 8 Kg and that of new material (Aluminium 7075-T6) is 2.41 Kg. Thus the weight of the shaft is reduced by three times. Now, there are four axles used on the vehicle, so by changing the material of all other shafts there can be in total large amount of weight reduction in the vehicle.

Thus, the new material Aluminium 7075-T6 can be an alternative for the old/existing material.

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