

Design and Development of the Front Wheel Hub for All-Terrain Vehicle (ATV)

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ABSTRACT

An all-terrain vehicle (ATV) is a single seat, open cockpit, and open wheel off-road vehicle in which the engine is located behind the driver. The present paper discusses the important aspects of designing and development of the front wheel hub of ATV. This study discusses the design of the front wheel hub while considering that it should be of light weight and high strength. This paper discusses the material selection for the hub from the two different types of material. This study includes the improvement in the design of the hub with the help of various analyses of the hub. The hub is analyzed in the various loading conditions to obtain the appropriate factor of safety with the help of a static structural module of ANSYS software.

Keywords: *All-Terrain Vehicle; Hub; Failure Analysis; Development; Static Structural Analysis*

Introduction

A front wheel hub is a component whose main purpose is to connect the wheel to other suspension components via stub axle and to keep the free spinning of the wheel on the bearing while keeping it attached to the vehicle. It is located between the disc and the stub axle of the ATV as shown in Figure 1. In older vehicles, front wheel bearings have been built to be serviced with repair kits because individual parts can be disassembled, washed and re-packed with grease. Generally, new vehicles are designed with front wheel assemblies comprising axle, bearing assembly, installation,

and rim flange incorporated into one unit known as a wheel bearing and hub assembly.

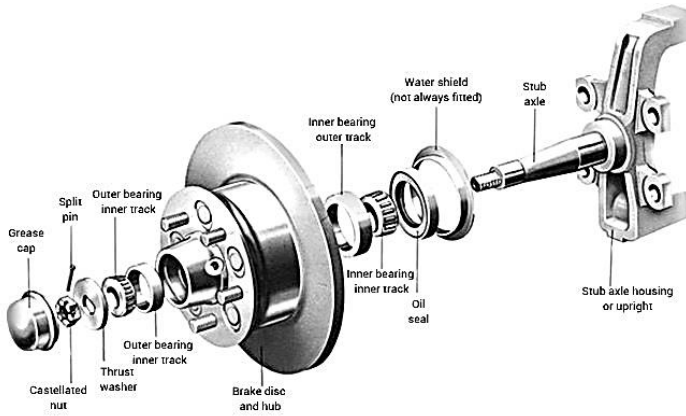


Figure 1: Wheel hub assembly [1]

BÖNAA [1] integrated the brake rotor and wheel hub in a single unit for use on automobiles and other vehicles. Conventional ATV uses different hubs and rotors are normally made as two separate parts bolted together to allow replacement of worn-out brake rotors. The disk's life span is relatively short compared to the hub which usually requires no replacement. This model results in more machining and thereby some possibilities of unbalance and misalignment during construction. Today's approach for the manufacture of hub and rotor as two separate parts requires additional material for bolt flanges, thereby increasing the overall size. Shrivastava [2] used the crash pulse scenario standard which is used for impact time to calculate the radial forces using Newton laws of motion on the hub in the worst-case scenario. Kumar et al. [3] discussed the use of aluminum composites in automobile and aerospace industries for various high performing components that are being used for varieties of applications owing to their lower weight excellent thermal conductivity. Among several series of aluminum alloys, aluminum possesses very high strength, higher toughness and is preferred for in the aerospace and automobile sector.

As the use of ATV is limited to only BAJA SAE 2018 events only thus, the hub design, material selection and structural development of the hub must choose accordingly. Baja SAE is made up of contests that represent projects in real-world engineering design and associated obstacles. Engineering students are responsible for designing and building an off-road vehicle that will survive the severe punishment of rough terrain. The goal of each team is to design and build a sporting single-seat, all-terrain vehicle with the driver's structure. The vehicle is to be a model for a robust,

functional, ergonomic and inexpensive production vehicle that meets a niche for recreational users. The semi-trailing or trailing arm suspensions are usually used in the rear of a vehicle while the MacPherson strut and double wishbone models can be used both in the front and rear [4]. Often recognized as a single control arm suspension is the MacPherson strut, comprising of a strut or shock unit, wheel hub and one control arm [5]. The strut as well as the control arm is connecting directly to the vehicle's chassis. Then the control arm attaches to the bottom of the wheel hub, while the strut connects to the top.

The reduction in the rotational mass of the hub and reduction in the overall weight of ATV lead to an increase in the acceleration. Furthermore, hub experiences the continuous stresses and impact stress due to the motion of ATV such as during braking, cornering and six feel fall. Therefore, the hub should be designed so that it should be of minimum weight and higher strength.

Methodology of Designing

Designing is one of the most important and thoughtful processes. At different design stages, the various problems occurred that need to be sought out using the following five steps which are given in Figure 2 [6]:

1. Define the problem
2. Gather pertinent information
3. Generate multiple solutions
4. Analyze and select a solution
5. Test and implement the solution

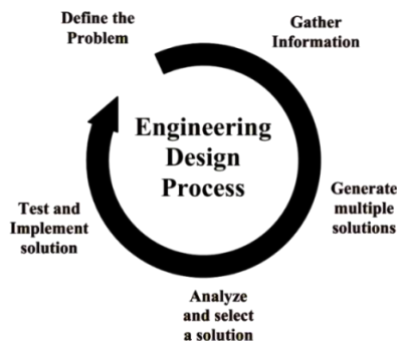


Figure 2: Design stages [6]

Selection of Material

In the present study, the material is selected on the basis of the following properties:

1. The material should have a low density or cheap or a combination of both.
2. The material should have enough strength values to carry the load with a sufficient factory of safety (FOS).
3. It should be easily machinable.

The two materials considered for the wheel hub are Aluminium 7075-T6 and EN8 Mild Steel. Table 1 illustrates the different properties of these materials [7, 8]. The material is selected on the basis of strength to weight ratio.

Table 1: Material property chart

Property	Aluminum 7075-T6	EN-8 Mild Steel
Density	2.81 g/cm ³	7.85 g/cm ³
Brinell Hardness Number	150 BHN	201 BHN
Ultimate Tensile Strength	572 MPa	650 MPa
Tensile Yield Strength	503 MPa	415 MPa
Poisson's Ratio	0.33	0.33
Fatigue Strength	159 MPa	--
Strength to Weight Ratio	203.56 MPa·cm³/g	83.22 MPa·cm³/g

Designing of Hub

The main objective of designing the hub is to be reliable, durable and light weight to overcome the failure of last years. In order to satisfy these requirements, AL-7075 T6 material is selected, from the two material options i.e. Al-7075-T6 and EN8, based on material strength to ratio. Prior to designing a thoroughly study is done on the previous year's design, experiences, and failures. Considering the facts and following design considerations, parameters, and last year's failure data, the preliminary design is made on CATIA software as shown in Figures 3 and Figure 4.

Design considerations

Several design considerations are proposed before the CAD modeling of hub:

1. The pattern of bolt connected to the wheel and brake rotor is determined by the type of rim and disc respectively.
2. The size and pitch circle diameter of the rim should be considered.
3. Pitch circle diameter of the brake disc.
4. Hub length is decided by the caliper dimension constraint.
5. The material should opt accordingly as strong enough to take the weight of the car and variable stresses.

6. Wheel bearing in the hub depends on internal and external diameters of stub axle coming out of the hub.
7. Bolt size should be considered.

Design parameters of the hub

Following design parameters of the hub are considered in the present study:

1. Loading condition.
2. Manufacturing process.
3. Material behaviour on the application of load.

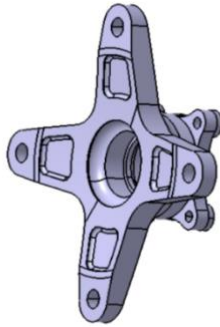


Figure 3: Final CAD model

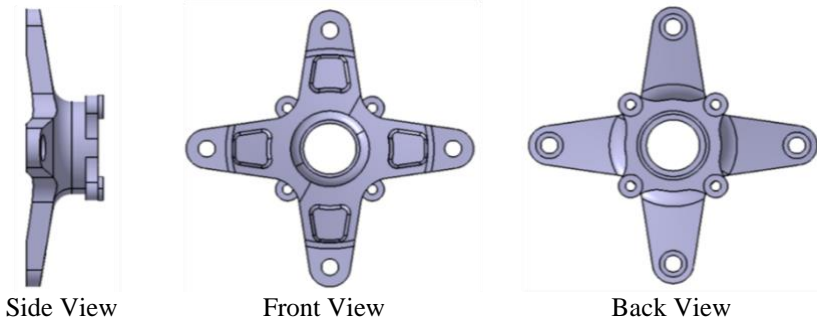


Figure 4: Different views of the hub

After several iterations as shown in Figure 5, the tapered cross-section with the fillet model is finalized for the analysis to evaluate the stressed area, deflections and life cycle of the designed component.

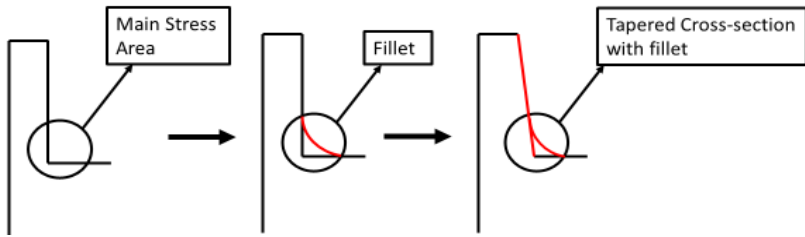


Figure 5: Improvement in the design of the hub

Results and Discussion

For the analysis of the hub, a 3-D model is generated in CATIA and imported into ANSYS. Material specifications of Al7075-T6 shown in Table 1 are assigned in engineering data in ANSYS. To observe maximum stress produced in the hub model is subjected to extreme conditions and static analysis is carried out in ANSYS. Mesh model of the hub is shown in Figure 6, having 75420 nodes and 44464 total elements.

Loading Conditions

Breaking torque

As the brakes are used frequently in the ATV so the hub comes under the frequent stresses, therefore, it requires to be analyzed properly in braking conditions. The hub experiences shear stress due to braking torque. Constraints are applied on the two end faces of the hub which are fixed on the disc side along with the application of torque 243 N-m (calculated in Appendix A) on the rim side face as shown in Figure 7.

As shown in Figures 8 and 9, the maximum stress comes near the disc mountings. Therefore, the fillet is provided at this place, however, the value of stress is very less hence the chances of failure will insignificant here.

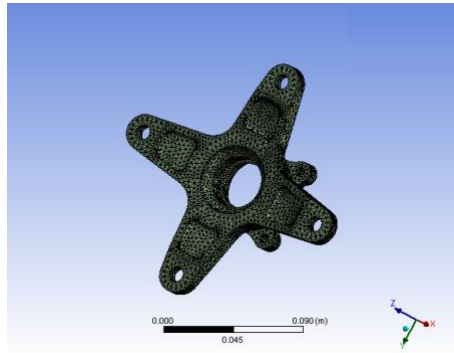


Figure 6: Mesh model of the hub

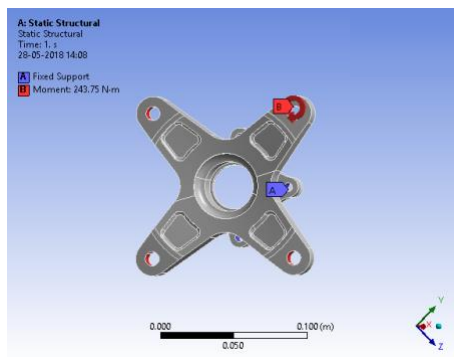


Figure 7: Loading condition in case of breaking torque

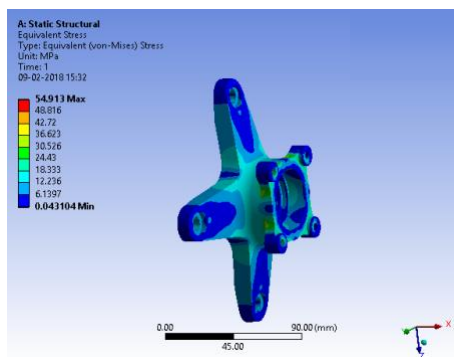


Figure 8: Von-Mises stress distribution

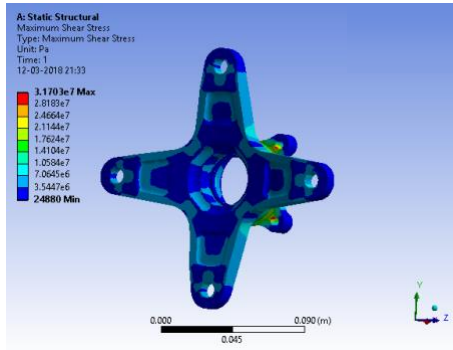


Figure 9: Maximum shear stress distribution

Six feet fall

This is the harsh condition that can occur in the ATV especially during the suspension and traction event. During six feet fall, the hub comes under the bending condition. The normal and shear stresses both occur during the bending thus either of these stresses cannot particularly decide the FOS of the hub, therefore, the Von-Mises Stress criteria is used to decide the FOS of hub under this loading condition. Constraints are applied on both the bearing surfaces as cylindrical supports and load of 6130N (calculated in Appendix B) as remote force is applied on the rim side end face as shown in Figure 10.

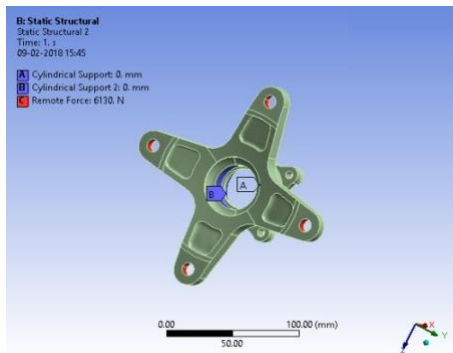


Figure 10: Loading condition in case of six feet fall

Figure 11 shows that the maximum stress occurs near the end of flanges towards the stub-axle side due to the bending of flanges, hence, failure can take place there. To avoid failure, the tapered cross-section with a fillet is kept here.

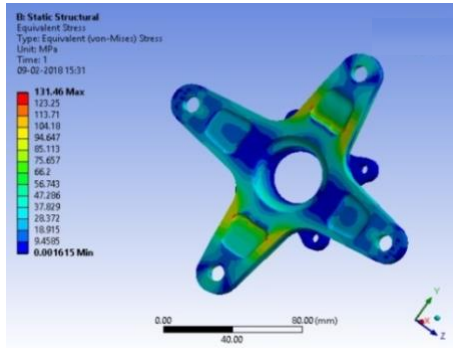


Figure 1: Von-Mises stress distribution

Cornering

Particularly in the maneuverability event, the ATV should respond to the quick turning, therefore due to sudden turns the hub comes under the cornering forces which cause the bending of the flanges of the hub. Therefore, the Von-misses stress criteria are used for deciding the FOS of the hub in this condition. Boundary conditions are applied on both the bearing surfaces as cylindrical supports and load of 1500 N as remote force is applied on the rim side end face as shown in Figure 12.

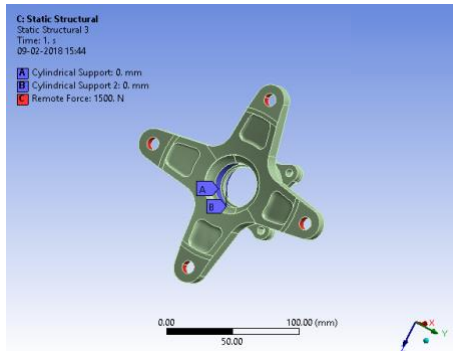


Figure 2: Loading condition in case of cornering

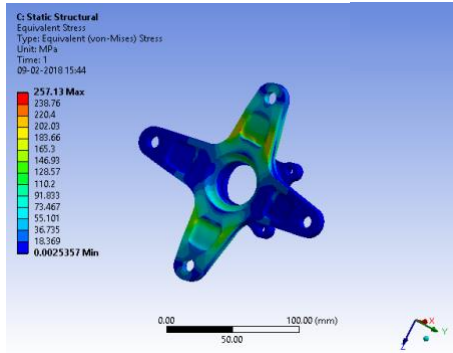


Figure 3: Von-Mises stress distribution

In this case of cornering, the chances of failure are maximum because the maximum stress with the value of 257.13 MPa occurs near the end of flanges towards the stub-axle side due to the bending of cantilever flanges as shown in Figure 13. Tapered cross-section with a fillet is provided to avoid the failure in this case also as described in the case of six feet fall.

After the analysis and improvements in various loading conditions, it is noted that the maximum stress in the most severe condition is 257.13 MPa under cornering condition on the flanges as shown in Figure 13, which is very lower as compared to the yield strength of material i.e. 503 MPa. Table 2 illustrates the maximum induced Von-Mises stress and FOS for different boundary conditions.

Table 2: Von-Mises stress and FOS for braking, six feet fall and cornering condition

Parameters	Unit	Boundary Conditions		
		Braking	Six Feet Fall	Cornering
Von-Mises stress	MPa	54.91	131.46	257.13
FOS	-	9.16	3.82	1.96

Conclusions

The weight of the manufactured hub is found to be approximately equal to the CAD model of the hub i.e. 314 g. The least weight, higher reliability, and durability of the hub are achieved through the various structural development of hub as shown in Figure 4 and with the help of the proper material selection, i.e. components Al 7075-T6 for the hub. In the various loading conditions i.e. six feet fall, cornering and braking the stresses come out to be

131.46 MPa, 257.13 MPa and 54.91 MPa with the appropriate factor of safety 3.82, 1.96, and 9.16 respectively. The front wheel hub is rigorously tested on the track and good performance is achieved.

1. Appendix A - Calculation of the braking torque

Braking force on disc exerted from pedal [9]

Pedal force	=	600 N
Pedal ratio	=	6.5
Master cylinder bore diameter	=	0.75 inch
Caliper bore diameter	=	1.12 inch
Ratio of bore areas	=	2.23 inch
Caliper pad and disc friction coefficient	=	0.35

$$\text{Braking force} = 600 \times 6.5 \times 2.23 \times 0.35 \times 2 = 6088.02 \text{ N}$$

Vehicle mass considered for calculation (m)	=	185 kg
Deceleration (a _d)	=	0.8 g
Traction coefficient (μ)	=	0.8
C.G. height	=	17 inch
Wheel base	=	54 inch
C.G. height/ wheelbase (γ)	=	0.315
Weight distribution	=	front - 45%; rear - 55%

Therefore,

$$\text{Static axle load distribution (v)} = \frac{\text{Static rear axle load}}{\text{Vehicle weight}} = 0.45$$

$$\begin{aligned} \text{Dynamic normal load on the front axle (F}_{zF,dyn}) &= (1 - v + \gamma a_d) m \times g \\ &= (1 - 0.45 + 0.314 \times 0.80) \times 185 \times 9.81 = 1454.05 \text{ N} \end{aligned}$$

$$\text{Dynamic normal load on each tyre} = \frac{F_{zF,dyn}}{2} = 727.02 \text{ N}$$

$$\text{Tractive force} = 727.02 \times 0.80 = 581.62 \text{ N}$$

$$\text{Braking torque} = 581.62 \times 11 \times 0.0254 = 162.50 \text{ Nm}$$

$$\text{Factor of safety} = 1.50$$

$$\text{Braking torque on front wheel} = 162.50 \times 1.50 = 243.75 \text{ Nm}$$

2. Appendix B - Calculation of six feet fall force

According to third equation of motion [10]

$$v^2 = u^2 + 2as$$

Here, $u = 0 \text{ m/s}$, because the velocity in vertical direction the initial velocity in the free fall is 0.

$$a = g(9.8 \text{ m/s}^2)$$
$$s = 6 \text{ feet or } 1.81 \text{ meter}$$

so, $v^2 = 0 + 2 \times 9.8 \times 1.81$

$$v = 5.97 \text{ m/s}$$

$$\text{Impact force} = \frac{\text{Change in momentum}}{\text{Impact time}} = \frac{\Delta mv}{\Delta t}$$

Initial velocity in vertical direction, $u = 0 \text{ m/s}$

Final velocity in vertical direction, $v = 5.97 \text{ m/s}$

Mass of the vehicle (m) = 185kg

$$\Delta mv = 185 \times 5.97 = 1104.45 \text{ kg} \cdot \text{m/s}$$

Impact time(Δt) = 0.18 s

$$\text{Impact force} = \frac{1104.45}{0.18} = 6130 \text{ N [2]}$$

Appendix C - Cornering force calculation for front hub [11]

G	=	gross weight of vehicle
$F_{\mu 1}, F_{\mu 2}$	=	Longitudinal and lateral frictional forces respectively
N_{s1}, N_{s2}	=	Dynamic normal reactions on the front & rear tires respectively
C.G height (r_h)	=	0.432 meter
r_{as}	=	Track width/2
μ	=	friction coefficient between tire and ground
k_a	=	cornering force
g	=	gravitational acceleration

Vertical mass considered for calculation (m) = 185 kg

$$G = m \times g = 185 \times 9.81 = 1814.85 \text{ N}$$

$$F_{\mu_1} = G \times \mu = 185 \times 9.81 \times 0.80 = 1451.88 \text{ N}$$

$$F_{\mu_2} = F_{\mu_1} \times \mu_1 = 185 \times 9.81 \times 0.80 \times 0.80 = 1161.50 \text{ N}$$

$$\begin{aligned} F_{\mu_1} \times r_h + F_{\mu_2} \times r_h + (G + F_{dec}) \times r_{as} - N_{s_1} \times (r_{as} + r_{as}) &= 0 \\ = (1451.88 + 1161.50) \times 0.432 - N_{s_1} \times 1.244 & \\ + (1814.8 + 1451.88) \times 0.622 &= 0 \end{aligned}$$

$$N_{s_1} = 2540.05 \text{ N}$$

$$K_a = \left(\frac{N_{s_1}}{2} \right) \times 0.80 = 1016.02 \text{ N}$$

$$\text{Factor of safety} = 1.48$$

$$K_a = 1016 \times 1.48 \text{ N} = 1500 \text{ N}$$

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