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Design and Optimization of Wheel Hub for an ATV Quad Bike

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***Abstract*: The Concept of Design of Machine Elements is used in order to Design a wheel hub for an All-Terrain Vehicle (ATV). Finite Element Analysis is used as a method to optimize the design of wheel hub. Commercial vehicle industry is focusing on bringing quality products at competitive costs. Therefore, the design should meet the following criteria, Lightweight to maintain good performance to weight ratio of the ATV. Optimum stiffness to ensure low system compliance and maintaining designed geometries. Ease of maintenance for enhancing serviceability and setup repeatability. The design and analysis process is divided into three parts, first is to derive all the forces acting on the wheel hub in static as well as dynamic conditions. Second part is to prepare CAD models according to the constraints derive from steering and braking calculations. And third part is to analyze the model with the help of finite element analysis and optimize the design with main objective of reducing weight.**

***Keywords*: ATV, ansys, catia, design, hypermesh, optimization.**

1. Introduction

Hub is a main component in wheel assembly. The wheel hub spins along with the wheels bolted to it and provide the power to the wheels in order to rotate. This wheel hub is designed according to Pitch circle diameter of the selected rim. On the other side of hub, brake disk is mounted which generates required torque to stop the rotating hub as a result stopping the rotation of wheels. Due to failure of hub there will be chances of fetal accidents hence hub is a very crucial component of the vehicle hence it is necessary to use best material and must be design to avoid failure. The weight of hub is considered under unsprung mass of vehicle unsprung mass affects the performance of vehicle to a large extent. Therefore, the ultimate aim is to minimize the weight of hub without affecting its strength. To fulfill the purpose of project, the weight and dimensions of hub must be small as possible which will result in reduction of rotational mass. In this project, design and analysis of front hub is done. The purpose of this project is to reduce the weight and increase the strength with designing and analysis software.

2. Methodology

Firstly, calculations are done on steering, braking, and suspension subsystems. For the design of steering and suspension system this report uses special type of ICR (instantaneous center of rotation) diagram and for calculating various parameters related to the front wheel hub an extensive survey is done. For obtaining the vehicle behavior in various conditions LOTUS software is used. The formulae derived are validated using design books and research papers.

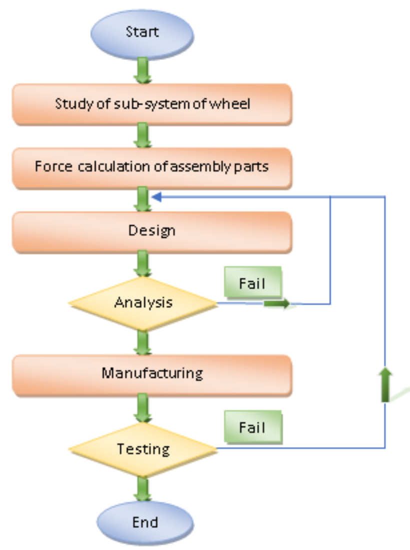

Fig. 1. Flowchart

3. Design consideration

According to the pitch circle diameter and pattern of bolts of rim selected, the same is to be selected on the hub face.

- According to the pitch circle diameter and pattern of bolts of disc selected, the same is to be selected on the other hub face.
- According to the designed kingpin offset from steering calculations, the overall length of the hub is decided.
- According to the OD of the stub axle, wheal hub bearing is decided. And according to the OD of the bearing, the ID of the hub is decided.
- For calculating the forces over hub, moment generated by caliper over brake disc is considered.
- According to the temperature conditions and force calculation, a suitable material is chosen.

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4. Calculations

The following forces are acting on the hub.

A. Force due to torque on brake disc petal

A torque of 276.95 N-m is generated by the caliper on the rotor. [as per braking calculations]

= Force acting on each hole

$= \frac{Moment \div Radius\ (PCD\ of\ brake\ disc)}{No.of\ holes}$

$= \frac{276950 \div 40}{3}$

= 2307.916 N

B. Force due to torque on wheel petal

The total torque generated by braking system should be compensated by the wheels that is, the wheel should apply equal and opposite torque to compensate the braking torque.

Force acting on each hole $= \frac{moment \div Radius\ (PCD\ of\ rim)}{No.of\ holes}$

$= \frac{276950 \div 70}{4}$

= 989.10 N

C. Force due to side impact

Side impact is essential to be considered because the hub petal can bend due to other vehicles collision or may be any kind of accidents.

The side impact is taken to be 2G generally,

Therefore, Impact force = m×2× g

= 220×2×9.81

= 4316.4 N

Impact force on each bolt $= \frac{Impact\ force}{No.of\ bolts}$

Rim has 4 bolts and brake disc has 3.

Therefore, considering no. of bolts as 3 for better factor of safety.

Impact force $= \frac{4316.4}{3}$ = 1438.8 N

D. Selection of wheel bolt

The wheel bolt selected was grade 5 due to their best properties over other grades. The availability is also good.

According to data of grade 5,

S_{yt} =81,000 psi

= 560 N/mm^2

1) Shear stress on bolts, $\tau = \frac{S_{yt} \times 0.5}{FOS} = \frac{560 \times 0.5}{2}$

$\tau = 140$ N/mm^2

Now, $\tau = \frac{Force\ acting\ on\ each\ hole}{Area}$

$= \frac{990}{\frac{\pi}{4} dc^2}$

$140 = \frac{990}{\frac{\pi}{4} dc^2}$

d_c = 3mm

d = d_c ÷ 0.8

= 3 ÷ 0.8

d = 3.75mm

We can select any bolt of size M04 and higher.

Therefore, M12 bolt is selected due to better factor of safety.

E. Selection of brake disc bolt

Tremendous amount of force which is generated by the caliper on brake disc, therefore there are higher chances for the bolts to fail. Therefore, the bolts of grade 8 are selected for the brake disc due to their maximum strength.

According to the data S_{yt} for grade 8 = 130000 PSI

Therefore S_{yt}= 896 N/mm^2

$\tau = \frac{S_{yt} \times 0..5}{2}$

$= \frac{896 \times 0.5}{2}$ = 224 N/mm^2

Now, $\tau = \frac{Force\ acting\ on\ each\ hole}{Area}$

$= \frac{2307.916}{\frac{\pi}{4} \times {d_c}^2}$

d_c= 3.621 mm

For determining the bolt size,

$d = \frac{d_c}{0.8}$

$= \frac{3.621}{0.8}$

d = 4.5274 mm

We can select any size of bolt M5 and above. Therefore, bolt M6 is selected for Better factor of safety.

F. Design of wheel petal

The main cause of failure of Wheel Petal is due to shear stress and bending moment. The Wheel petal is designed in order to overcome these stresses.

EN24 is Selected due to its best strength properties.

According to the data s_{yt} for EN24 is 950 N/mm^2

Therefore s_{yt} =950 N/mm^2

Shear failure

$\tau = \frac{950*0.5}{2}$

$= 237.5$ N/mm^2

Now, $\tau = \frac{\text{force acting on each hole}}{\text{area}}$

$237.5 = \frac{989.1}{2*t*b}$

$t*b = 2.0840$ mm^2

therefore, this is the required area

where, t = thickness of the brake petal

b = distance between the hole and the petal

for actual area,

while designing t is taken as,

8mm and b=8mm

Therefore, area = t*b

=8*8

=64mm2

The actual area is greater than the required area. Therefore, design is safe in shear failure.

G. Bending of wheel petal

Bending will occur at the holes on the hub which are at the distance of 70mm from the centre.

Therefore effective radius = $\frac{P.C.D}{2} - \frac{d}{2}$

Where, PCD = Pitch Circle of the rim = 120mm

D = diameter of the bolt = 12mm

$= \frac{120}{2} - \frac{12}{2}$

=54mm

Mb = Force * effective radius

= 989.1*54

= 53411.4 N/mm

By using the equation

$\frac{M_b}{I} = \frac{\sigma_b}{y}$

Where, σ_b = bending stress

$= \frac{S_{yt}}{Fos} = \frac{950}{1.5}$

= 633.33 N/mm2

$I = \frac{1}{12} * t * b^3$

And $y = \frac{b}{2}$

t = thickness of the wheel petal

b = distance between the hole and the end of the petal

$\frac{M_b}{I} = \frac{\sigma_b}{y}$

$\frac{53411.4}{\frac{1}{12}*t*b^3} =$

t * b2 = 506.6005mm3

Therefore, this is the actual volume of material needed.

Where, t = thickness of the brake petal

b = distance between the hole and the petal

For actual area,

While, designing t is taken as 8mm and b=8mm.

Area = t*b

=8*8

=64 mm^2

The actual area is greater than the required area. Therefore, design is safe in bending failure.

H. Design of brake disk petal

The main cause of failure of brake disk petal is due to the shear stress, therefore the petal is designed in order to overcome this stresses.

I. Shear failure of petal

The allowable stress in shear which the material EN24 can withstand is given by

$\tau = \frac{S_{yt}*0.5}{Fos}$

$= \frac{950*0.5}{2}$

= 237.5 N/mm2

Now, $\tau = \frac{\text{Force on each hole}}{\text{Area}}$

$237.5 = \frac{2307.916}{2*t*b}$

T*b = 4.858 mm2

Where, t = thickness of the brake disc petal

b = distance between the hole and the end of the petal this is the required area.

For actual area

While designing t is taken as 6mm and b is taken as 8mm.

T*b = 6*8

$= 48\ mm^2$

Hence design is safe.

5. Calculation result

Table 1
Forces on Hub

S. No	Parameters	Results
1	Force due to brake disc petal	2307.916 N
2	Force due to Torque on wheel petal	989.10 N
3	Force due to side impact	4316.4 N

Table 2
Selection of bolts

S. No	Parameters	Results
1	Wheel Bolt	M12
2	Brake Disc Bolt	M06

Table 3
Hub design

S. No	Parameters	Results
1	Material	EN24
2	Thickness of the Wheel petal	8 mm
3	Distance between hole and Wheel petal	8 mm
4	Thickness of the brake petal	6 mm
5	Distance between hole and Brake petal	8 mm

6. Design of hub cad model

The preparation of CAD model of Hub is done with the help of CATIA V5 (Academic software). CATIA is widely used software for component designing in various Industrial fields. Dimensions of hub model is based on the results obtained as per as the calculations performed. The CAD model is mainly influenced by the rim design and blot pattern on it.

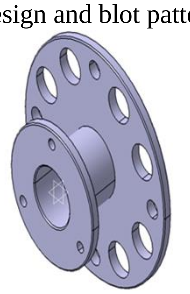

Fig. 2. CAD model of hub

A. Optimized part

The major change adapted in the optimization process is the change of material from EN24 to Aluminium 6061. The mass density of EN24 is 7750 Kg/m^3 and mass density of Aluminium 6061 is 2700 Kg/m^3. According to given properties Aluminium 6061 is 65% lighter than EN24. For the support of brake disc mounts on hub additional 1mm extrusion is provided over it. Ball bearings are best suited for axial loads but tapper bearing take up radial loads in addition of axial load. Therefore, tapper section is provided integrally. Instead of using circular geometry for the mounting of rims, a star shaped geometry is used which tends to reduce the weight of the hub.

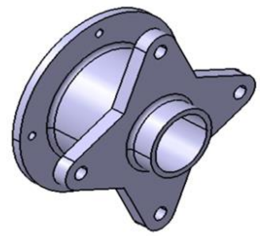

Fig. 3. CAD model of optimized hub

7. Finite element analysis

Meshing is the process of dividing a particular component into small number of elements. FEA is the finite element analysis method which make calculation only a limited number of points. Take any object it will have a continuous surface and infinite degree of freedom, so it is not possible to solve using FEA approach. FEA method reduces degree of freedom from infinite to finite with help of discretization that is meshing.

If the load is applied on the structure or a body and the body is considered to be meshed, then the load is distributed uniformly on the entire structure. After Meshing, the entire structure is divided into number of elements and each element having its own stiffness while loading. Adding all those elements stiffnesses, we can get the Global Stiffness Matrix with which the stress, strain, etc. developed in the structure is calculated. If the Vonmises stress is less than yield stress of the material, then the product analyzed is safe, else it is of failure type. If a load is applied on the body which is not meshed, then the load distribution is not uniform and irregular or faulty results are obtained. Hub is meshed by tetra mesh 3D technique using a software known as hypermesh. In real time, the designers also manually mesh the components using a software and then imported for analysis.

- *Quality Checks* - Quality check parameters are the measures of how far an object or model deviates from the ideal shapes. Quality check parameters are as follows;
- *Warpage:* It is a measure of how to close a QUAD element is to being planer. A perfect planer element will have the warpage of zero. Warpage of up to five degrees is generally acceptable
- *Skew:* The angles between the lines join opposite midsides. It Measure the angle created as square is turned into parallelogram or rhombus. Typical required values are to have less than 450 or 600.
- *Tria angle:* The Angle between two sides of a tri element should be 600 as much as possible. Typical required values are to have all tria angles between 200 to 1200. Sometime smaller angles are require to model geometry with small angle.
- *Aspect ratio:* It is the ratio of max length side of element to min length side of an element. This should be reduced as much as possible.

- Quad angle: The angle between two sides of a quad element should be 900 as much as possible. Typical required values are to have all angles between 450 and 1350.
- Jacobian: It is measure of close the shape of element is to ideal shape. Jacobian is really the best measure of finite element mesh quality. Typically, less than 5% of element in a mesh should have Jacobian less than 0.7 within the minimum value of 0.5.

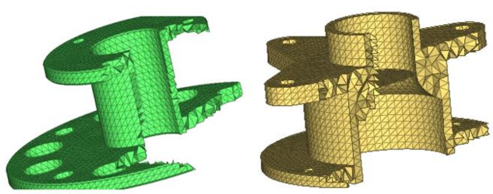

Fig. 4. Tetramesh (cross section)

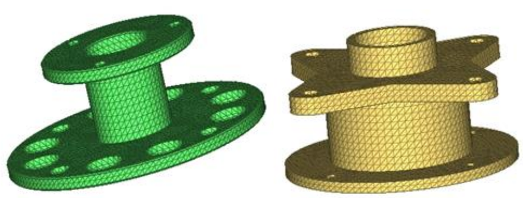

Fig. 5. Tertamesh

For the analysis various software's are available in the market but the analysis is carried out on ANSYS (academic software) due to its ease in operation. ANSYS is one of the widely used analysis software.

Loading conditions- The forces and moments calculated from above calculations are applied on the model which is tabulated below.

Table 4
Loading Conditions

Loading conditions	Fixed support	Moment	Force due to Torque on brake disc petal:	Force due to Torque on wheel petal:
Values	On the neck	276.95 N-mm	2308 N	990N

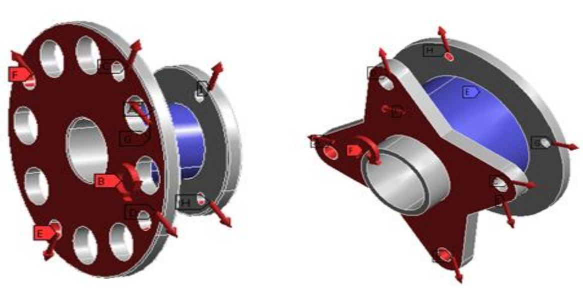

Fig. 6. Boundary conditions

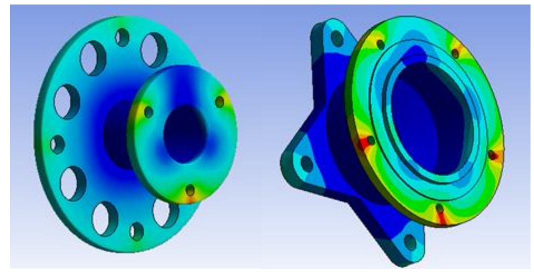

Fig. 7. Total deformation

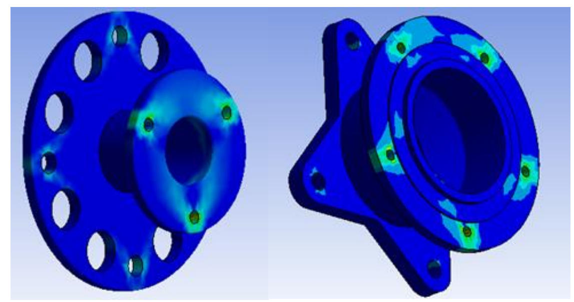

Fig. 8. Stress (von misses)

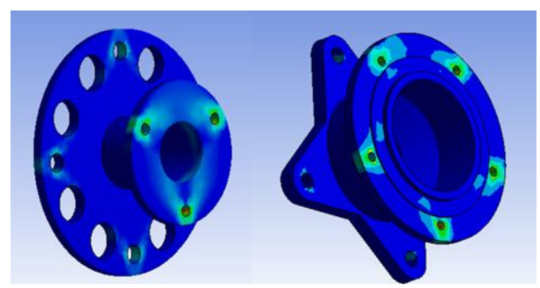

Fig. 9. Strain (von misses)

Table 5
Analysis Results

Parameters	Before optimization		After Optimization	
	Minimum	Maximum	Minimum	Maximum
Total Deformation	0	0.004394mm	0	0.0038867mm
Equivalent stress	0.01564 MPa	58.564 MPa	0.02469 MPa	69.85 MPa
Equivalent strain	4.906×10^{-7}	3.1192×10^{-4}	6.77×10^{-7}	3.66×10^{-4}

From the above table the total deformation of the optimized hub is found less than pre-optimized hub. Equivalent strain is nearly the same but only equivalent stress value after optimization has increased. But as seen from above considerations the tensile strength of aluminium 6061 is 276 MPa, it can easily take up stress of 69.85MPa without any failure.

8. Manufacturing

A. Raw Material- Aluminium 6061

Aluminium 6061 was initially known as Aluminium 61S and was founded in 1935. It is an alloy which is heat treated which results into improving its yield strength. Its maximum composition is of mainly aluminium, silicon and magnesium. It has excellent mechanical properties and good weldability therefore it is widely use amongst other alloys of aluminium.

B. Chemical composition

The chemical composition of Aluminium 6061 is tabulated below;

Table 6
Chemical Composition

Components	**Maximum (%)**	**Minimum (%)**
Aluminium	98.56	95.85
Silicon	0.8	0.4
Iron	0.7	-
Copper	0.4	0.15
Manganese	0.15	-
Mangnesium	1.2	0.8
Chromium	0.35	0.04
Zinc	0.25	-
Titanium	0.15	-
Other Element	0.05	-

C. Material properties

Some of the material properties of Aluminium 6061 are tabulated below;

Table 7
Chemical Composition

Material	ALUMINUM 6061
MASS DENSITY (Kg/M^3)	2700
YOUNGS MODULUS (Gpa)	69
Hardness (Bhn)	95
TENSILE STRENGTH (Mpa)	276
YEILD STRENGTH (Mpa)	310

D. Manufacturing process

The steps involved in manufacturing Hub are as follows;

- Turning
- Facing
- Milling
- Drilling
- Chamfering
- Spline cutting

Fig. 10. Manufactured Hub

9. Validation of manufactured hub model

- *Technical Inspection:* All the reports are validated by the judges (Quad Torc 2018). The report included are as follows; Analysis Report, Calculation Report, CAD Reports. The vehicle was examined according to the given reports and was declared technically O.K
- *Drop Test:* In this test the vehicle was thrown from a height of 7 feet from ground level. In this case the force encountered by the wheel hub is almost 3 times that of static conditions. The hub was found safe after the test.
- *Brake Test:* The purpose of this test is to check the functioning of braking system. Judges determine whether the vehicle can achieve a minimum stopping distance without any wheel lock within given period of time. During breaking the hub undergone opposing force from the brake disc side and inertia developed from wheel side. Maximum torque was encountered during this test and hub found safe during this test too.
- *Acceleration Test:* This test is conducted to check the performance and condition of the vehicle. In this test judges record the time taken by the vehicle to obtain the speed up to 40KM/Hr. while accelerating the RPM of the wheel gets increased suddenly which results in torque from the side where wheel is mounted. In this test also it found that hub is O.K.

Fig. 11. Validation Proof

10. Result

Table 8
Results

Parameters	**Before Optimization**	**After Optimization**
Material	EN 24	ALUMINIUM 6061
Weight	1.82KG	0.657KG
Total Deformation	0.004394mm	0.0038867mm
Equivalent Stress	58.564 MPa	69.85 MPa
Equivalent Strain	3.1192×10^{-4}	3.66×10^{-4}
Stopping Distance during braking	2.5m	1.7m
Time taken to reach 40 KM/Hr	**35s**	**24s**

11. Conclusion

Hence the purpose of hub optimization of ATV vehicle is successfully achieved with the weight reduction over 50%. The performance of the vehicle is increased successfully due to the reduction in the unsprung mass, without affecting its strength.

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