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Fatigue based design and analysis of wheel hub for Student formula car by Simulation Approach

V Gowtham[1], A S Ranganathan[2], S Satish[3], S John Alexis[4], S Siva kumar[5]

[1]UG student, Department of Automobile Engineering, Kumaraguru College of Technology, Coimbatore, Tamil Nadu, India, Corresponding author, Email id: gowthamvis94@gmail.com
[2]UG student, Department of Automobile Engineering, Kumaraguru College of Technology, Coimbatore, Tamil Nadu, India, Email id: vivekprabhu1994@gmail.com
[3]Assistant Professor, Department of Automobile Engineering, Kumaraguru College of Technology, Coimbatore, Tamil Nadu, India, Email id: satish.s.auto@kct.ac.in
[4]Professor, Department of Automobile Engineering, Kumaraguru College of Technology, Coimbatore, Tamil Nadu, India, Email id: johnalexis.s.auto@kct.ac.in
[5]Associate Professor, Department of Automobile Engineering, Kumaraguru College of Technology, Coimbatore, Tamil Nadu, India, Email id: sivakumar.s.auto@kct.ac.in

Abstract. In the existing design of Wheel hub used for Student formula cars, the brake discs cannot be removed easily since the disc is mounted in between the knuckle and hub. In case of bend or any other damage to the disc, the replacement of the disc becomes difficult. Further using OEM hub and knuckle that are used for commercial vehicles will result in increase of un-sprung mass, which should be avoided in Student formula cars for improving the performance. In this design the above mentioned difficulties have been overcome by redesigning the hub in such a way that the brake disc could be removed easily by just removing the wheel and the caliper and also it will have reduced weight when compared to existing OEM hub. A CAD Model was developed based on the required fatigue life cycles. The forces acting on the hub were calculated and linear static structural analysis was performed on the wheel hub for three different materials using ANSYS Finite Element code V 16.2. The theoretical fatigue strength was compared with the stress obtained from the structural analysis for each material.

1. Introduction

The existing models of wheel hub that are being used in Student formula car (SF car) which are participating in events like FSAE, SUPRA require dismantling the entire wheel assembly for removing the brake disc during damage or crack of the disc, which consumes huge time. In case of OEM hub, the mass is huge which will affect the performance of the SF car in terms of weight. The objective of this project is to design a hub for SF car in which the disc rotor can be easily removed in case of damage with reduced weight when compared to OEM hub. Since the disc in the SF car is subjected to various tests like endurance, skid pad, acceleration, autocross and brake test, a need for simple and quick removal of brake disc for replacement is necessary. The proposed design eliminates the above said difficulty by improving interchangeability and also reduced weight. The fatigue life requirement of the wheel hubs that are used for SF car is also less when compared to the fatigue life of the OEM hubs. Hence the modified wheel hub is designed for the required fatigue life.

2. Nomenclature

Table 1. Nomenclature of the terms used

S. No	Terms	Notations
1.	Maximum Acceleration	F
2.	Wheel base	B
3.	Distance of rear axle from C.G	L
4.	Height of C.G	H
	Wheel track	J

6.	Coefficient of adhesion	μ
7.	Acceleration due to gravity	G
8.	Dynamic reaction on rear wheels due to the effect of acceleration	w_r
9.	Maximum dynamic reaction on the rear inner wheels due to banked road	W_i
10.	Maximum bank angle	A
11.	Tractive force	T_e
12.	Driving torque	T_d
13.	Change in dynamic reaction force on wheel due to effect of centrifugal force	P_{ir}
14.	Cornering force	P_c
15.	Radius of curvature of shortest turn	C
16.	Maximum possible velocity of the vehicle while taking the shortest turn	V
17.	Allowable alternating stress	S_f
18.	No. of cycles to failure	N_f
19.	Fatigue strength coefficient	A
20.	Fatigue strength exponent	B
21.	Mean stress	σ_m
22.	Alternating stress	σ_a
23.	Surface roughness factor	K_a
24.	Size factor	K_b
25.	Loading factor	K_c
26.	Temperature factor	K_d
27.	Reliability factor	K_e
28.	Fatigue stress concentration factor	K_f
29.	Stress concentration factor	K_t
30.	Endurance limit	S'_e
31.	Diameter of bar before machining	D
32.	Notch diameter	D
33.	Notch radius	R
34.	Decrease in diameter at the stress concentrated area	H
35.	Neuber constant	$\sqrt{e}$
36.	Fatigue strength fraction	F

3. Parameters Considered

3.1. Vehicle and Track specification

Table 2. Considerations for calculation

S. No.	Particulars	Units
1	Car Weight	220 kg
2	Driver Weight	80 kg
3	Wheel base [2]	1600 mm
4	C.G Height [2]	340 mm
5	C.G Distance from rear wheel [2]	688 mm
6	Engine Model	CBR 600 F4i
7	Wheel track [2]	1260 mm
8	Final drive Ratio	4
9	Maximum Engine Torque	66 Nm
10	Co-efficient of adhesion	0.6
11	Radius of shortest turn	7m
12	Wheel radius	530 mm
13	Maximum bank angle	12^0
14	First gear ratio	2.833
15	Final drive ratio	4

3.2. Material Properties

Table 3. Physical Properties

S. No	Properties	Notations	EN24 Steel [11]	EN8 Steel [12]	Al 7075-T6 [8]
1.	Density	P	7.83 kg /m^3	7.83 kg /m^3	2.77 kg /m^3
2.	Young's Modulus	E	210 GPa	210 GPa	71.7 GPa
3.	Poisson's ratio	N	0.3	0.3	0.33
4.	Yield strength	S_y	654 MPa	465 MPa	503 MPa
5.	Ultimate strength	S_u	850 MPa	700 MPa	572 MPa

4. Theories Adopted

4.1. Loads Considered

Drive torque acting on the wheel hub, traction force and maximum cornering force acting while taking the shortest turn are the dynamic forces acting on the wheel hub. Dynamic reaction force acting on the wheel will be the sum of vehicle weight, effect of load transfer due to centrifugal force and banking of roads. All these force will be acting on the rear wheel hub, so it is considered for designing. Since there will not be any bumps in formula track, bump loads are neglected. Since traction force will create a moment on wheel hub, it can be determined using the below equation.

$$T_e = \mu * W_i \quad (1)$$

Torque from the final drive shaft is transmitted to the wheels by means of wheel hub. Hence, it is necessary to include the effect of drive torque which can be calculated from

$$T_d = \text{Engine torque} * \text{First gear ratio} * \text{Final drive ratio} \quad (2)$$

From the above equation the torque supplied to each rear wheel can be calculated by multiplying it by half. While taking a turn, the wheels will be acted upon by cornering force, which can be determined by the following equation,

$$P_c = \frac{m_r * V^2}{C} \quad (3)$$

In order to account for dynamic load transfer to the wheels, it is necessary to know the effect of acceleration and deceleration. Since the load transfer to the rear wheels will be maximum during acceleration, maximum possible acceleration should be determined [5].

$$\frac{F}{g} = \frac{(B-L)}{\left(\frac{B}{\mu}-H\right)} \quad (4)$$

Knowing acceleration from the above equation, the dynamic rear axle load due to acceleration can be determined from the following equation [5]

$$w_r = \frac{(B-L)}{B} * w + \frac{H}{B} * \frac{W}{g} * F \quad (5)$$

Increase in dynamic reaction on rear inner wheel while moving on banked roads due to lateral load transfer can be determined form the following equation [5].

$$W_i = \frac{w_r}{2} + \frac{H}{j} * w_r * \alpha \quad (6)$$

Centrifugal force acting on the vehicle while taking a turn will create load transfer, it is necessary to include the effect of centrifugal force [5]

$$P_{ir} = \frac{w}{g}\ (1 - \frac{l}{b}) * \frac{v^2}{g*c} * \frac{H}{j} \quad (7)$$

(It is added to the reaction on inner wheel and reduced from the reaction on the outer wheel), v can be determined by following equation

$$V = \sqrt{\frac{c*g(\sin\alpha + \mu * \cos\alpha)}{(\cos\alpha - \mu * \sin\alpha)}} \quad (8)$$

The overall dynamic reaction force acting on the rear wheel can be determined from the following equation,

$$= W_i + P_{ir} \text{ (for inner wheel)} \quad (9)$$

4.2. Von Mises yield criterion
The Von Mises yield criterion is used to predict yielding of materials under any loading condition from results of simple uniaxial tensile tests. The Von Mises stress satisfies the property that two stress states with equal distortion energy have equal Von Mises stress. The condition for failure is

$$\left[\frac{(\sigma_1-\sigma_2)^2+(\sigma_2-\sigma_3)^2+(\sigma_3-\sigma_1)^2}{2}\right]^{\frac{1}{2}} \geq \sigma_y \qquad (10)$$

This expression for failure is obtained from the distortion energy failure theory. Since it is found to be more accurate, this theory is followed for the analysis of wheel hub [3].

4.3. Stress life approach
In order to calculate the allowable stress amplitude to achieve the desired number of cycles, Stress life approach is followed. For high cycle fatigue (i.e.) life > 10^3 cycles, Stress life approach will be appropriate [6].It is based on the S-N curve which depicts the relationship between stress and no. of cycles. The type loading pattern in the wheel hub is considered as fully reversed with zero mean stress[4]. For S-N diagram involving high cycle fatigue with finite life, the Basquin Equation is used to find the allowable alternating stress.

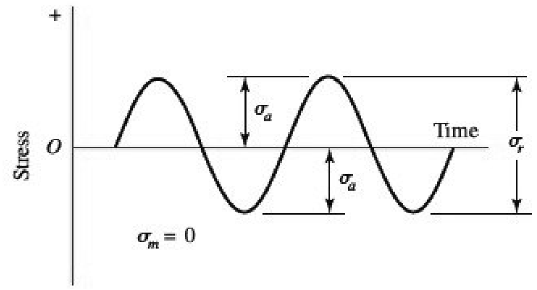

Figure 1. Type of loading pattern

5. Fatigue Strength
The distance covered by a student formula car is very less compared to commercial vehicles. Hence, the wheel hub of SF car can be designed for lower fatigue life. In order to determine the required fatigue strength, the number of fatigue cycles covered by the wheel hub should be determined, which can be calculated as follows

5.1. Required minimum fatigue life
There are total numbers of three events in which the vehicle should be driven, which are endurance, autocross, and skid pad [2]. The distance covered in each event and in testing phase was taken into account to determine the required minimum fatigue life.

Table 4. Minimum distance covered for one event [2]

Distance for each event	Value (km)
Distance covered in endurance	22
Distance covered in Autocross	4
Distance covered in Skid pad	2
Distance covered in tests at event	3
Distance covered in on track test	400
Total distance covered	431

The minimum number cycles that a hub should undergo is calculated as follows
Perimeter of the wheel along with the tire = 1.638 m
Required minimum number of cycles = 431000/1.638
= 263125 = 300000 cycles (Approx.)

5.2. Theoretical fatigue strength calculation:

5.2.1. Surface roughness factor

Fatigue failure initiates at surface, in order to account for the effect of surface roughness on fatigue strength surface roughness factor should be determined.

$$K_a = a_1 * (S_u)^{b1} \tag{11}$$

Table 5. Surface factor [7]

S. No	Surface Finish	Factor a_1	Exponent b1
1.	Machined	4.51	-0.265

Since the surface of the wheel hub is machined, the corresponding correction factor determined from the above equation was 0.76 for EN 24, 0.795 EN 8 steel and 0.838 for Al 7075-T6.

5.2.2. Size factor

With the increase in size the number of internal defects will also increase. Since the diameter of the standard specimen is very much lower than the diameter of the bar from which the wheel hub is machined, size effect on fatigue strength is considered here.

$$K_b = 1.51 * (d)^{-0.157}, \text{ for } 51 < d < 254 \text{ mm } ^{[7]} \tag{12}$$

Size factor of 0.71 was calculated by considering the diameter of the bar used for machining wheel hub as150 mm.

5.2.3. Reliability factor

The experimental results taken are always mean values. Hence, in order to account for the deviation reliability factor is used. Based on the required reliability the factor can be calculated from the following equation.

$$K_e = 1 - 0.08(Z_a) \tag{13}$$

Table 6. Reliability factor [7]

S.No	Reliability	Transformation Variate, Z_a	Reliability Factor, K_e
1.	95	1.645	0.868

Assuming Reliability as 95%, the corresponding reliability factor calculated is 0.868.

5.2.4. Loading factor

The endurance limit is taken from the specimen tested under bending load. Since bending load will be predominant in wheel hub it is considered to have a load factor of 1[9].

5.2.5. Temperature factor

The operating temperature of wheel hub is ambient temperature itself. Hence the effect of temperature on the fatigue strength of wheel hub will be negligible and so the temperature factor is taken as 1.

5.2.6. Fatigue stress concentration factor

The design of hub includes variation in cross section diameter which may result in rise of stress in that area. In order to include the effect of stress concentration the following method is followed.

Fatigue stress concentration factor [7], $K_f = 1 + \frac{(K_t - 1)}{1 + \frac{\sqrt{e}}{\sqrt{r}}}$ (14)

$$K_t = c1 + c2 * \frac{2*h}{D} + c3 * \left(\frac{2*h}{D}\right)^2 + c4 * \left(\frac{2*h}{D}\right)^3 \tag{15}$$

$$K_t = 1.49 \text{ for } r = 2 \text{ and } h = 5 \text{ mm} \tag{16}$$

$$\sqrt{e} = 0.245799 - 3.07E\text{-}03\,(S_u) + 1.5E\text{-}5\,(S_u)^2 - 2.6E\text{-}08\,(S_u)^3 \tag{17}$$

From equation 14, the fatigue stress concentration factor for the wheel hub design is 1.454

5.2.7. Endurance limit

The stress allowed to achieve infinite life (i.e.) 10 lakhs is called as endurance limit and the endurance limit of steel materials can be calculated from equation 18.

$$S'_e = 0.5*S_u \text{ for steel} \quad (18)$$

$$S'_e = 160 \text{ for Al 7075-T6 [10]}$$

By accounting the effect of surface roughness, size, reliability, stress concentration, load and temperature, the modified endurance strength for the wheel hub can be calculated as follows

$$\text{Modified endurance limit [7]}, S_e = K_a* K_b* K_c* K_d* K_e* S'_e / K_f \quad (19)$$

Based on the above values and equation the modified endurance limit for each material is mentioned in the Table 7.

Table 7. Comparison of endurance limit of materials

S. No	Material	Endurance limit of the material MPa	Modified endurance limit of the material MPa
1.	EN 24	425	136.9
2.	EN 8	350	117.93
3.	Al 7075	160	56.83

5.2.8. Fatigue strength at 300000 cycles

In order to determine the required fatigue strength, the fatigue strength exponent and the fatigue strength coefficient should be calculated. The fatigue strength fraction, f is taken from the graph shown in Figure 2, which is based on ultimate strength of the material.

Fatigue strength exponent [7], $$b = -\frac{1}{3} * \log\left(\frac{(f*S_u)}{S_e}\right) \quad (20)$$

Fatigue strength coefficient [7], $$a = \frac{(f*S_u)^2}{S_e} \quad (21)$$

Knowing the above parameters, the allowable alternating stress can be calculated from the Basquin equation [7], $$S_f = a\,(N_f)^{b} \quad (22)$$

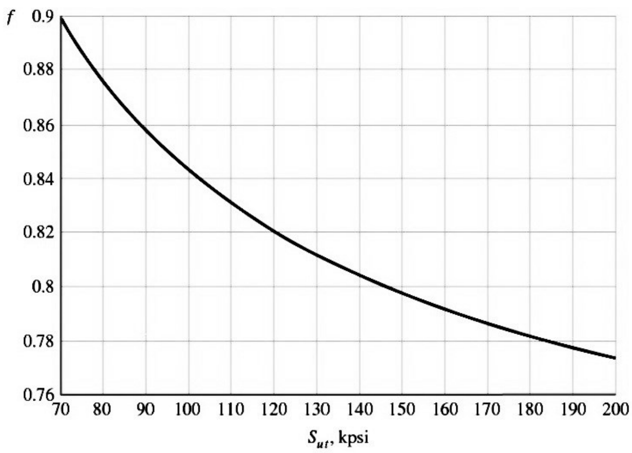

Figure 2. Fatigue strength fraction vs. Ultimate strength of the material [7]

Using the above equations and graph, the fatigue strength fraction, fatigue strength coefficient, fatigue strength exponent and required fatigue strength were calculated, which are shown in Table 8.

Table 8. Allowable alternating stress for each material

S. No	Material	Fatigue strength fraction, f	Fatigue strength exponent, b	Fatigue strength coefficient, a MPa	Allowable alternating stress, S_f MPa
1.	EN 24	0.82	-0.2356	3448.64	176.7
2.	EN 8	0.845	-0.2246	2626.07	154.57
3.	Al 7075	0.87	-0.3141	4357.656	82.96

6. Modeling and Analysis

The wheel hub used here is designed by considering the dimensions of the assembling components, which are wheel bearing, knuckle, wheel rim, brake disc and half shaft. Doing subsystem analysis will provide more accurate stress results. Hence, the entire wheel assembly was taken for analysis to obtain stress acting on wheel hub. The wheel assembly was meshed with tetrahedral elements and the meshing was refined near the stress concentrated areas. The loads were applied and the resulting Von Mises stress acting on the wheel hub was obtained.

Table 9. Dimensions of components with the wheel hub

S. No	Parameters	Values
1.	Inner diameter of wheel bearing	35 mm
2.	Width of wheel bearing	17 mm
3.	PCD of the wheel lobe	90 mm
4.	PCD of the disc lobe	100 mm

Software used for designing: CATIA V5R20

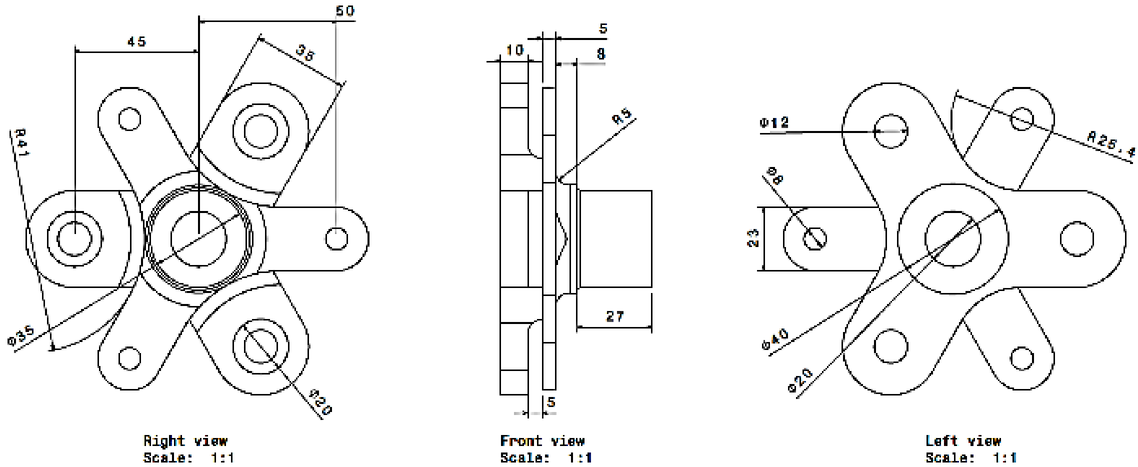

Figure 3. 2D of the Wheel hub

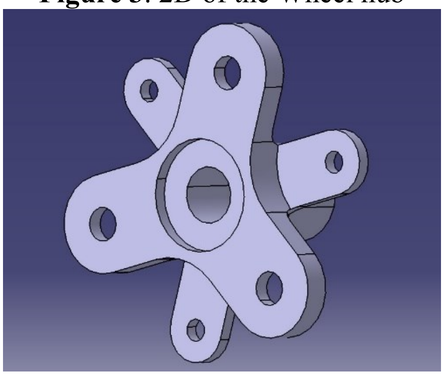

Figure 4. Isometric view of the Wheel hub

Table 10. Applied loads and constraints

S. No	Loads and constraints	Value	From equation
1.	Total dynamic reaction force, C	1042 N	9
2.	Driving torque, A	347 Nm	2
3.	Traction force, D	647 N	1
4.	Cornering force, E	611 N	3
5.	Fixed to frame, B	-	-

Analysis software used: ANSYS WORKBENCH 16.2

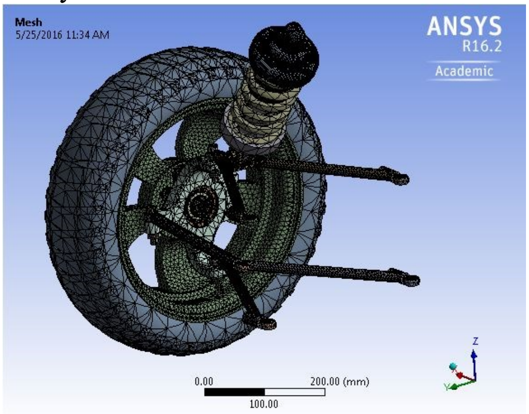

Figure 5. Mesh model

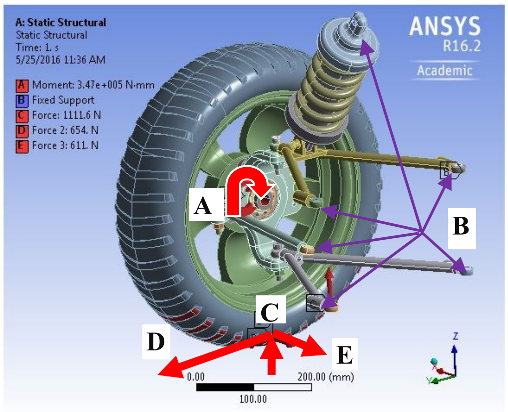

Figure 6. Loads and constraints

7. Results and discussion

In this work a wheel hub has been designed for lower weight by considering the required fatigue strength. The required fatigue life cycle was determined from the total distance covered by a vehicle for an event. Three different materials EN24, EN8 and Al 7075-T6 were selected for the wheel hub by considering yield stress, availability, machinability and material cost. Based on the material properties, the allowable stress to achieve the required number of fatigue life cycles was calculated. The Von Mises stress obtained from the Static structural analysis of wheel assembly was compared to the allowable stress for each material. Analysis was done for three materials and the results are shown in Table 11.

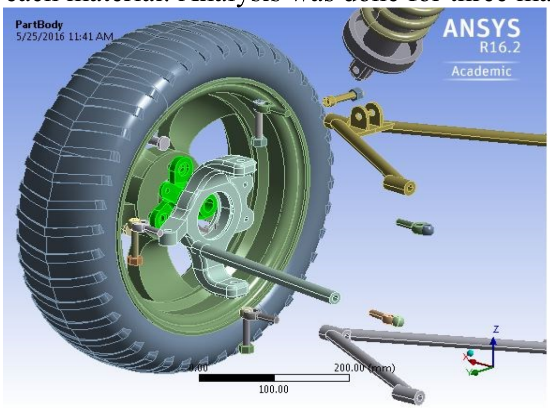

Figure 7. Exploded view of assembly

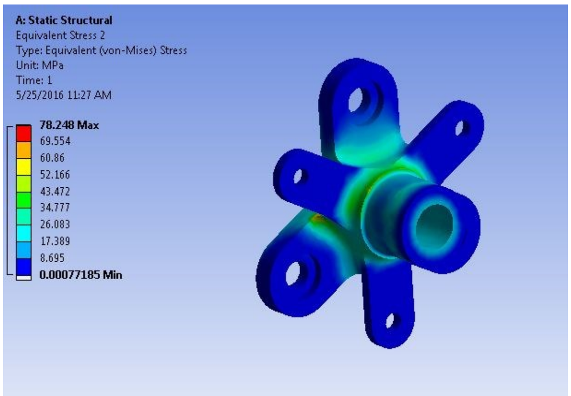

Figure 8. Von Mises stress for EN 24

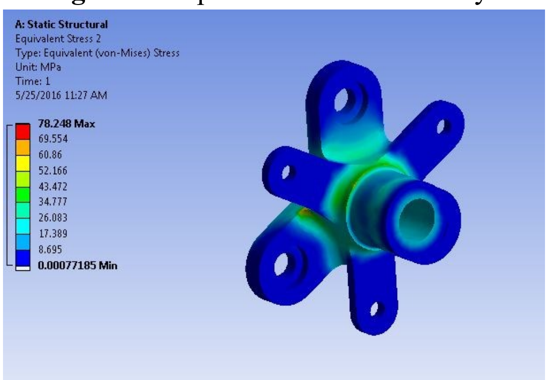

Figure 9. Von Mises stress for EN 8

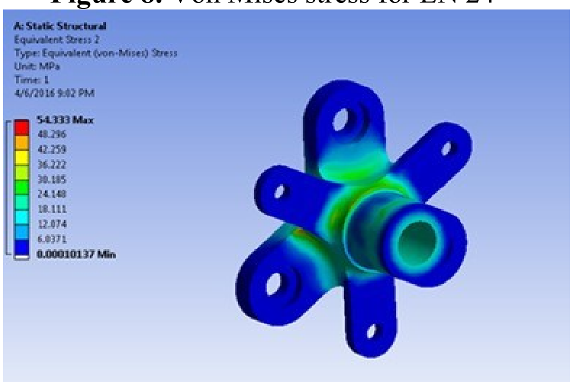

Figure 10. Von Mises stress for Al 7075-T6

Table 11. Overall resulting values

S. No	Materials	Allowable stress MPa	Alternating stress MPa	Factor of Safety
1.	EN 24	176.7	78.25	2.26
2.	EN 8	154.57	78.25	1.98
3.	Al 7075-T6	82.96	54.34	1.53

Fatigue factor of safety should be greater than 1 [1]. From Table 11, it is understood that all materials have FOS greater than 1, yet EN 24 will be the suitable material for wheel hub as it has higher allowable stress when compared to the other two materials. Also the cost of Al 7075 is very high when compared to EN 24. Further the weight of the wheel hub is 0.850 kg which is very low when compared to the OEM wheel hubs.

8. Conclusions

In order to overcome the difficulty of replacing the brake disc from the wheel hub of SF car and to reduce the weight of wheel hub, a new model of wheel hub is designed for SF cars. The design was done based on the requirements; loads acting on the wheel hub, the required minimum fatigue life of the wheel hub and the allowable stress amplitude. Simulation analysis was carried out using ANSYS Workbench 16.2. From the results of the analysis, it is found that the design is safe upto the required number of fatigue life cycles and also the weight is reduced considerably when compared to OEM hubs.

9. References

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