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

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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 unsprung 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

| 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         |
| 5     | Wheel track         | J         |

Table 1. Nomenclature of the terms used

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6. Coefficient of adhesion \( \mu \)
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. Ttractive force \( T_e \)
12. Driving torque \( T_d \)
13. Change in dynamic reaction force on wheel due to effect of centrifugal force \( P_a \)
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 \( \sigma_m \)
22. Alternating stress \( \sigma_a \)
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 \( S' \)
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

| 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\(^{o}\) |
| 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/m³      | 7.83 kg/m³     | 2.77 kg/m³     |
| 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.

\[ T_e = \mu \times W_i \]  

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

\[ T_d = \text{Engine torque} \times \text{First gear ratio} \times \text{Final drive ratio} \]  

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,

\[ P_c = \frac{m \times v^2}{c} \]  

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].

\[ \frac{P}{g} = \frac{(B-L)}{\mu-H} \]  

Knowing acceleration from the above equation, the dynamic rear axle load due to acceleration can be determined from the following equation [3]

\[ w_r = \frac{(B-L)}{B} \times w + \frac{H}{B} \times \frac{w}{g} + F \]  

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].

\[ w_i = \frac{w_r}{2} + \frac{H}{l} \times w_r \times \alpha \]  

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]

\[ P_{ir} = \frac{w}{g} \left(1 - \frac{l}{b}\right) \times \frac{v^2}{g} \times \frac{H}{l} \]  

(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

\[ V = \sqrt{\frac{c \times g \left(\sin \alpha + \mu \times \cos \alpha \right)}{(\cos \alpha - \mu \times \sin \alpha)}} \]  

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

\[ = W_i + P_{ir} \text{ (for inner wheel)} \]
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

\[
\frac{(\sigma_1-\sigma_2)^2+(\sigma_2-\sigma_3)^2+(\sigma_3-\sigma_1)^2}{2} \geq \sigma_y
\]  

(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.

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. 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.

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. The distance covered in each event and in testing phase was taken into account to determine the required minimum fatigue life.

| Event                        | Distance (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 = \(\frac{431000}{1.638} = 263125 = 300000\) 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.

\[ K_a = a_1 \cdot (S_u)^{b_1} \]  

Table 5. Surface factor \[^7\]

| S. No | Surface Finish | Factor \(a_1\) | Exponent \(b_1\) |
|-------|----------------|----------------|-----------------|
| 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.

\[ K_b = 1.51 \cdot (d^{-0.157}) \] \[ \text{for } 51 < d < 254 \text{ mm} \] \[^7\]  

Size factor of 0.71 was calculated by considering the diameter of the bar used for machining wheel hub as 150 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.

\[ K_e = 1 - 0.08(Z_a) \]  

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 \[^7\], \(K_f = 1 + \left( \frac{(K_t-1)}{1+\frac{c}{r}} \right)^{1+\frac{c}{r}} \)  

\[ K_t = c_1 + c_2 \cdot \left( \frac{2h}{D} \right) + c_3 \cdot \left( \frac{2h}{D} \right)^2 + c_4 \cdot \left( \frac{2h}{D} \right)^3 \]  

\[ K_t = 1.49 \text{ for } r=2 \text{ and } h = 5 \text{ mm} \]  

\[ \sqrt{e} = 0.245799 - 3.07E-03 (S_u) + 1.5E-5 (S_u)^2 - 2.6E-08 (S_u)^3 \]  

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.

\[
S_e' = 0.5S_u \text{ for steel} \\
S_e' = 160 \text{ for Al 7075-T6} \quad [10]
\]

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

Modified endurance limit \[7\], \[S_e = K_a * K_b * K_c * S_e' / K_f \]

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

| 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 \[7\],

\[
b = -\frac{1}{3} * \log \left( \frac{(fS_u)}{S_e} \right)
\]

Fatigue strength coefficient \[7\],

\[
a = \frac{(fS_u)^2}{S_e}
\]

Knowing the above parameters, the allowable alternating stress can be calculated from the Basquin equation \[7\],

\[
S_f = a (N_f)^b
\]

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.

| 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

![2D of the Wheel hub](image)

![Isometric view of the Wheel hub](image)

**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                         | -       | -             |
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.
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 up to the required number of fatigue life cycles and also the weight is reduced considerably when compared to OEM hubs.

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