Shape Optimization of Passenger Vehicle Wheel on Fatigue Failure

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Abstract. Based on the basic passenger vehicle steel wheel shape design, revised design was proposed to solve the common fatigue failure problems. Both basic and optimized design were modeled using Abaqus 3D software to calculate the stress and strain contours in the case of bending load and radial load. Consequently, fatigue life contours of models for each load case were calculated with the help of Brown-Miller biaxial fatigue theory. It was shown that, at the condition of unchanged wheel material and number of disc vents, optimized rim profile can extend fatigue life index of disc by 73.8% and 58.5% under bending load and radial load respectively. Meanwhile, the fatigue life index of rim could be extended by 17.5% and 134.4% under bending load and radial load respectively.

1. Introduction
Passenger vehicle wheel is a critical component that bears the auto-body weight and load weight. Structural design is a vital part of the whole automobile industry to ensure that wheels work in the normal condition without earlier fatigue failure [1-4]. The safety and stability of an operating vehicle are largely attributed to superior stiffness, elastic property and fatigue performance. Several complex operating conditions are inevitably introduced while driving, which as a consequence possibly leads to the structural failure [1]. The failure modes of wheels are mainly divided into strength failure and fatigue failure, and above 80% are induced related to the latter mode [5-6]. Safety is always the primary consideration of current international advanced wheel design, highlighting the significance of the strength and fatigue life of the wheel [7-8].

Fatigue fracture is a kind of typical high-cycle fatigue [1]. In consideration of the importance of fatigue life, previous work has taken special research in the fatigue testing machine [9]. With the rapid development of computer technology, CAD/CAE techniques are widely used in industrial design and manufacturing process [10, 11]. Applying the simplified model without considering the effect of circumferential position variation of loads, stress-strain distribution and fatigue life of the wheel can be easily simulated with a computer, which considerably improves the efficiency of product design and reduces the development cost [12-15]. A precondition for using simulation technique to calculate the fatigue and optimize the structure is that a precise enough stress-strain distribution is required. Thus, in order to further improve the accuracy of the simulating calculation, this work has made improvements in the finite element modeling analysis process based on the previous research: 1. Assembly of a certain type of wheel and bolts has been modeled, with considering the effect of pre-tightening force, thus the calculation precision of regions near to the bolt holes has been enhanced; 2. Under the radial load condition, integral modeling of the wheel rim has been proposed, achieving the
actual stress state of the wheel in operating conditions; 3. In consideration of the discrete distribution of bolt and vent holes along the circumference, multiposition calculation (every 45° set a calculation in this paper) has been applied for the solving process of bending and radial loads; 4. After analysis of the stress-strain distribution under both bending and radial loads, Brown-Miller biaxial fatigue model has been applied and fatigue life contour has been figured out. Furthermore, via comparing the fatigue life of wheels with various structures, shape optimization of this type of wheel has been achieved, improving its fatigue life.

2. Fatigue failure of wheels
A certain type of wheel without tube is shown in Figure 1(a), assembled by a drop-slot rim and a metal spinned disc. The center part of the wheel has 8 bolt holes and a center hole. The disc is connected to the hub by bolts. The center hole is used for centering location of the wheel, making sure that the center lines of the wheel and hub are coaxial. The edge of the disc is connected with the rim by welding, and several vents are set at the lateral of the disc. For this type of wheel, fatigue fractures normally appear at vicinity of bolt and vent holes, or at the joint areas (shown in Figure 1(b), (c) and (d), respectively).

(a) schematic diagram of wheel (b) area around the bolt holes (c) area around the vents (d) area around the welding

Figure 1. Common fatigue failure of a certain type of wheel

3. Structural optimization and FEM calculation

3.1. Basic shape of wheel
A wheel is generally composed of a disc and a rim. The rim provides a stage for installation of vehicle tyre, and the disc is jointed with the hub via bolts. As the thickness of the disc gradually tends to thin from its bottom to the edge, a reasonable distribution of materials can be obtained after metal spinning, resulting that the whole structure is of a constant strength but various cross-sections. This metal spinned disc has the advantages of high dimensional precise, superior surface quality and light weight[9]. The cross-section outline and 3D structure of a certain wheel are illustrated in Figure 2(a) and 2(b), respectively.

(a) cross-section of wheel (b) 3D model of wheel

Figure 2. The cross-section view and 3D model of wheel
3.2. **Shape optimization of wheel**
Under the condition that the wheel disc remains unchanged, the optimized solution has been put forward as follow. The rim shape has been optimized by adjustment of the cross-sectional shapes in this work to reduce the stress concentration.

**Table 1.** Comparison of basic and optimized scheme

| Basic model | Optimized model |
|-------------|----------------|
| ![Basic model image](image1.png) | ![Optimized model image](image2.png) |

3.3. **Computing standards and loading conditions**
According to “Performance requirements and test methods of commercial vehicles wheels” (GB/T 5909-2009), the fatigue life has been assessed by the verification of dynamic bending fatigue and radial load (shown in Figure 3).

![Schematic diagrams of fatigue failure test of vehicle wheel](image3.png)

(a)schematic diagram of bending load   (b)schematic diagram of radial load

**Figure 3.** Schematic diagrams of fatigue failure test of vehicle wheel

Referring to the requirements of GB/T 5909-2009, the finite element model has been established. Elastic modulus of the material used is 215000MPa and the Poisson ratio is 0.3. The stress-strain curve of the material is shown in Figure 4.

![Stress-strain curve of wheel material](image4.png)

**Figure 4.** Stress-strain curve of wheel material
Based on the national standard “Hexagon socket head cup screws” (GB/T 70.1-2000), M14 bolts with the washer diameter of 21.3mm have been chosen in this work. The torque of bolts is 610N.m and the pre-tightening force coefficient chosen is 0.3\[9\].

**Figure 5.** FEM model of bending and radial load of wheel

Finite element modelling of wheel has been implemented by using the FEM software Abaqus. The basic mode has been meshed into 65206 elements; Optimized mode has 63172 elements.

### 3.4. Analysis of bending torque and radial load

Under 6.885KN.m bending torque, the stress contour and the maximum stresses on critical areas of each mode are shown in Table 2. Under 8800lbs radial load, the stress contour and the maximum stresses on critical areas of each mode are shown in Table 2.

| Item          | Bending          | Radial          |
|---------------|------------------|-----------------|
| Model         | Basic model      | Optimized model |
| Stress contour| Bolt hole:307MPa | Bolt hole:257MPa| Bolt hole:200.4MPa | Bolt hole:182MPa |
|               | Vent:319 MPa     | Vent:303 MPa    | Vent:134 MPa      | Vent:129 MPa     |

### 4. Fatigue life analysis based on stress condition

In general operating condition, the wheel is simultaneously subject to bending and radial loads, thus the stress state belongs to multi-axial stress. Multi-axial fatigue is also termed as biaxial fatigue, which is different from uniaxial fatigue that cracks normally first appear on the component surface under multi-axial stresses [16]. Brown-Miller fatigue law, a critical plane approach, states that generation of fatigue cracks has certain directionality, showing that fatigue cracks first appear at the plane subject to the maximum shearing strain [17]. Due to the actual physical significance of this law, it is widely applied as a prediction method for multi-axial fatigue life [18].

#### 4.1. Brown-Miller fatigue law

Based on the results of elastic FEM calculation, the stress history of any node can be obtained. With the principal stress, the relevant principal strain can be figured out according to generalized Hooke law, thus the time history of the shearing strain and normal strain can be further calculated.

\[
\frac{\Delta \gamma}{2} + \frac{\Delta \varepsilon_n}{2} = 1.65 \frac{\sigma_f}{E} \left(2N_f\right)^b + 1.75\varepsilon_f \left(2N_f\right)^c
\]

After correction of the mean stress:
\[
\frac{\Delta \gamma}{2} + \frac{\Delta \varepsilon_n}{2} = 1.65 \left( \frac{\sigma'_j - \sigma_{mn}}{E} \right) (2N_j)^b + 1.75 \dot{\varepsilon}_j (2N_j)^c
\]

(2)

Where \( \Delta \gamma \) is the shearing strain; \( \Delta \varepsilon_n \) is the value of shearing strain in the normal direction.

4.2. Calculation of fatigue life under bending and radial load
Based on Brown-Miller fatigue law and the stress distributions obtained by FEM calculation, the multi-axial fatigue life contour of each mode has been calculated, shown in Table 3.

| Item                  | Bending | Radial |
|-----------------------|---------|--------|
| Model                 | Basic model | Optimized model | Basic model | Optimized model |
| Fatigue Contour       | ![Fatigue Contour](image1) | ![Fatigue Contour](image2) | ![Fatigue Contour](image3) | ![Fatigue Contour](image4) |
| Fatigue Life          | Fatigue life index of disc was increased from 3.66 to 3.9; fatigue life index of rim was increased from 6.59 to 6.66. | Fatigue life index of disc was increased from 3.6 to 3.8; fatigue life index of rim was increased from 4.73 to 5.1. |

By analyzing the fatigue life contour of each mode, it is concluded as follow:

1) Under the radial load, the critical parts of the rim have been identified as the slot zone. In addition, some other potential failure regions have been observed periodically appearing in the disc/rim welded zones facing the vents. Under the radial load, the potential failure regions are also concentrated in both the slot and disc/rim welded regions.

2) Under the bending load, the bending area adjacent to the bolt holes of the disc becomes the potential region of fatigue failure, which has the trend extending to the vents. Meanwhile, similar result has been obtained under the radial load.

3) Under both bending and radial loads, fatigue fracture greatly tends to appear at regions adjacent to bolt holes and vent holes.

4.3. Analysis and Comparison of fatigue life
According to the calculated fatigue life of bending and radial loads, the life index of each mode has been plotted against the location of cross-section. Comparisons of the life index of optimized mode with the basic shape have been shown in Figure 6, with shadow areas indicating the life index distribution related to optimized modes and solid lines representing the distribution of the basic mode.
From the life index distribution, it is observed that the fatigue life index of disc under bending is 3.66, which of the optimized model is 3.9; for radial load, the life index of basic model is 3.6, which of the optimized model is 3.8; the fatigue life index of rim under bending is 6.59, which of the optimized model is 6.66; for radial load, the life index of basic model is 4.73, which of the optimized model is 5.1.

5. Conclusions
Aiming to alleviate the fatigue fracture issue of a certain type of vehicle wheel, this paper presents optimized proposal by modifying geometric shape of rim cross-section. In this paper, 3D modeling and finite element method were used to calculate bending/radial load stress of both basic and optimized model. Then Brown-Miller multi-axial fatigue model was applied to calculate bending/radial load fatigue life of models. After analyzing and comparing fatigue life distribution of wheel, the following conclusions have been proposed:

1. Under the condition that the wheel material and number of disc vents remain unchanged, fatigue life index of disc under bending load was increased from 3.66 to 3.9 by 73.8% (from $10^{3.66}$ to $10^{3.9}$).
2. Under the condition that the wheel material and number of disc vents remain unchanged, fatigue life index of disc under radial load was increased from 3.6 to 3.8 by 58.5% (from $10^{3.6}$ to $10^{3.8}$).
3. Under the condition that the wheel material and number of disc vents remain unchanged, fatigue life index of rim under bending load was increased from 6.59 to 6.66 by 17.5% (from $10^{6.59}$ to $10^{6.66}$).
4. Under the condition that the wheel material and number of disc vents remain unchanged, fatigue life index of rim under radial load was increased from 4.73 to 5.1 by 134.4% (from $10^{4.73}$ to $10^{5.1}$).
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