Stiffness Analysis and Design Optimization of Hot-Stamped Lightweight Wheels

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Abstract. In this article, the design of the 22.5 × 9.00 hot-stamped lightweight wheel was optimized based on finite element analysis of stiffness. To improve the bending and radial stiffness of the wheel, under the condition to meet the requirements of bending and radial loading stress, the finite element stiffness analysis was carried out on the following three optimizations: increasing the spoke's thickness, modifying the shape and size of the windows, and setting up shuttle-shaped stiffeners between the windows. The numerical simulation results showed that the maximum bending stiffness around the bolt holes of the spoke increased by 1.4×10^6 N/m, and the maximum radial stiffness at the outer edge of the rim increased by $(2.25–4.18) \times 10^7$ N/m, which met the product requirements for stiffness and stress and achieved the goal of wheel lightweighting.

Keywords: Lightweight wheel · Hot-stamped · Stiffness analysis

1 Introduction

Lightweighting is the most effective means of energy-saving and emission reduction for automobiles. Research shows that 75% fuel oil consuming related to mass of automobile; usually, when the automobile mass decreases 10%, the fuel oil consuming will decrease 6%–8% [1]. As wheels are unsprung mass rotating parts, the outcome of energy-saving is more significant than that of sprung mass components after weight reduction [2]. For instance, the weight of a 22.5″ × 9″ wheel made of 380 CL and 420 CL steel is about 45 kg. And the weight of the wheel made of 650 CL high-strength steel is about 36 kg. The number of wheels on a commercial vehicle trailer is 6–7, and the number of wheels on a stake truck is 12–13, which means the outcome of weight reduction will be more considerable in terms of energy-saving.

Wheels, rotating components that bear the load of the whole vehicle, transmit traction, braking forces, driving moment, and braking torque, and absorb shock from the road [3]. Therefore, when new technologies such as high-strength materials, hot forming, and laser welding are involved in reducing wheels' weight, higher requirements are put forward for the design of the wheel shape and structure. Not only the stress and fatigue life of the wheel, but also its stiffness needs to be analyzed through finite element

Y. Zhang and M. Ma (Eds.): ICHSU 2022, AHMST 2, pp. 346–354, 2023.
https://doi.org/10.2991/978-94-6463-114-2_46

analysis software to ensure its safety and other performances are not compromised after weight reduction.

In this paper, with target weight of 28 kg and bolt tightening torque of 500 N·m, design optimization and stiffness analysis of the 22.5″ × 9″ wheel that bears the load of 3750 kg were carried out to ensure the wheel's stiffness and reliability were not impacted due to lightweighting, which provided a reference for designing hot-stamped lightweight wheels. As the calculation method of wheel stiffness has not had a unified standard in the industry, its calculation in this paper will be conducted by using the ratio loads/the deformation of the wheel under the loads, according to the definition of the stiffness - the stiffness of a wheel reflects the ability of the wheel to resist deformation under loads.

2 Design Optimization for Improving Bending Stiffness

2.1 Loads and Boundary Conditions of the Numerical Simulation Under Bending Loads

The simulation was based on standard [4]. As shown in Fig. 1, the edge of the rim was clamped, so all its translations and rotations were fixed in the simulation [5]. A concentrated force was applied on the drive shaft vertically (Fig. 2).

Bending moment formula:

$$M = (\mu R + D)F_V S$$

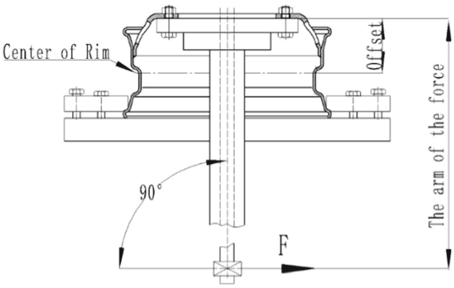


Fig. 1. Schematic diagram of the bending fatigue test.

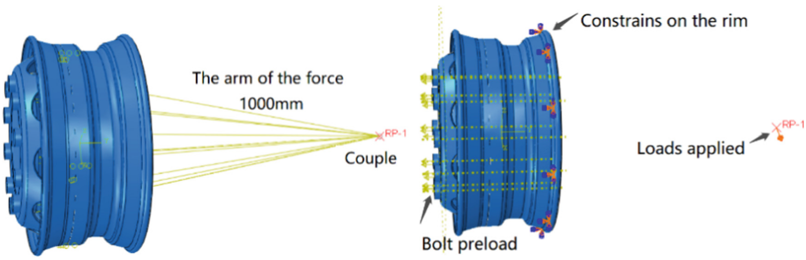
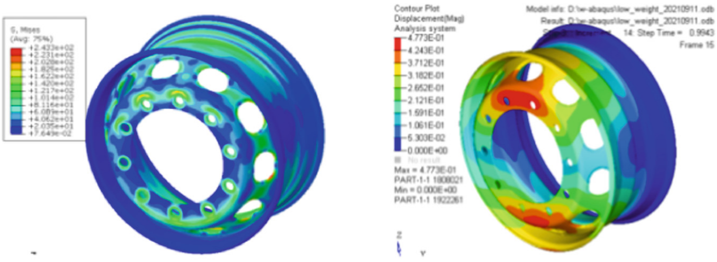


Fig. 2. Loads and boundary conditions of the finite element model.



(a) (b)

Fig. 3. Von-mises bending stress plot (a) and Deformation plot (b) under bending loads.

where,

$\mu = 0.7$, coefficient of friction between tires and roads.

$R = 504$ mm, the radius of the biggest tire can be installed on the wheel under static loads.

$d = 175$ mm, wheel offset.

$F_V = 37500$ N, rated load of the wheel (Gravitational acceleration is made to be 10 m/s^2 to simplify calculation.)

$S = 1.1$, experimental enhancement coefficient.

M = Bending moment.

Calculation results:

$M = 21771$ N·m.

As $M = F \times L$,

Where,

F = Torsional force.

$L = 1000$ mm, the arm of the force.

Thus, $F = M/L = 21771$ N.

2.2 Analysis of Bending Stiffness

The numerical simulation results under bending loads are shown in Fig. 3. The maximum stress is 243.3 MPa, located at the windows. The maximum deformation is 0.4773 mm, around the bolt holes. The stiffness is calculated as 4.56×10^7 N/m.

2.3 Design Optimization and Stiffness Variations

Increasing the thickness of the disc can improve the bending stiffness of the wheel, but with the increase of the thickness of the spoke, the weight of the spoke increases correspondingly. When the thickness of the spoke increases from 8 mm to 8.5 mm, its weight increases by 0.36 kg. The weight of the models' wheel for 8 mm and 8.5 mm spoke is respectively 27.9723 kg and 28.3286 kg, which fails to meet the target weight of 28 kg, so instead of adjusting the thickness of the spoke, the windows of the spoke were optimized.

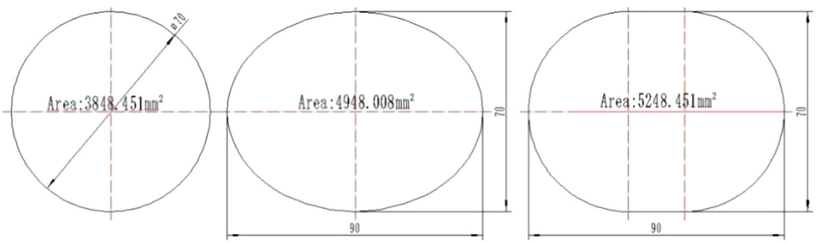
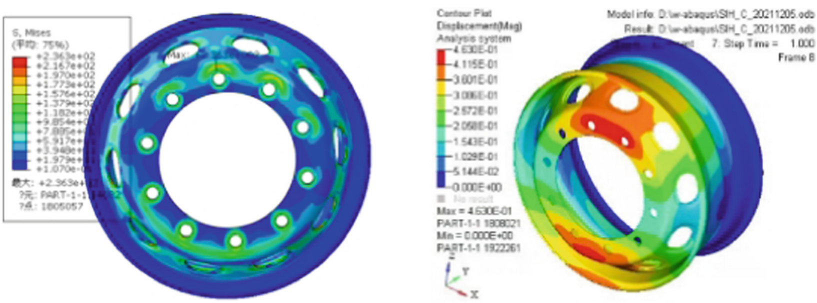


Fig. 4. Area comparison among circular, elliptic, and waist drum shape window.



(a) (b)

Fig. 5. Von-mises stress plot after optimization (a) and deformation plot after optimization(b).

2.3.1 Modified Design of the Window

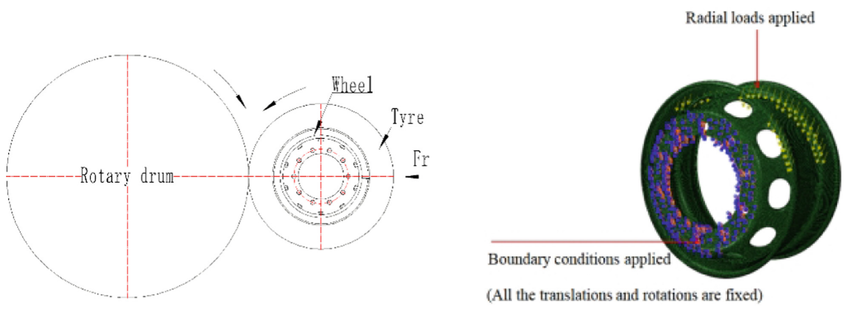
The size and the number of windows directly affect wheel weight, stress distribution, stiffness, and the cooling of the components in use. Hence, the design of the window is crucial for hot-formed lightweight wheels. As shown in Fig. 4 with the same number of windows on wheels, when the minor axis is 70 mm and the major axis is 90 mm, the waist drum shape window has the largest area compared to the circular and elliptic windows. Therefore, the waist drum shape window achieves the best performance in terms of weight reduction.

The weight of the models' wheel corresponding for three diffrence shape window is respectively 28.6108 kg, 28.0372 kg and 27.9745 kg (from left to right in Fig. 4).

To satisfy the same loading condition and further improve bending stiffness, the dimensions of the waist drum shape window were modified and optimized from 90 × 70 to 85 × 70, the weight of the wheel model is 28.13 kg.

2.3.2 Simulation Results After Optimization

Simulation results under bending loading conditions after optimization are shown in Fig. 5. The maximum bending stress at the edge of the windows is 236.3 MPa, which is lower than the maximum stress before optimization., and the maximum deformation is 0.463 mm. The stiffness is calculated as 4.70×10^7 N/m, which is 0.14×10^7 N/m higher than that before epimutation increased by 3.07%.



(a) (b)

Fig. 6. Schematic diagram of radial loading fatigue test (a) and radial loading in simulation (b).

3 Design Optimization for Improving Radial Stiffness

3.1 Loads and Boundary Conditions of the Numerical Simulation Under Radial Loads

Schematic diagram of radial loading fatigue test of the numerical simulation is shown in Fig. 6.

Loads: Forces are applied on the tire.

Boundary conditions: All translations and rotations of the mounting pad are fixed.

Radial load formula:

$$F_r = F_V K$$

where,

F_r = radial load.

F_v = 37500 N, rated load of the wheel (Gravitational acceleration is made to be. 10 m/s^2 to simplify calculation.)

K = 1.6, experimental enhancement coefficient.

Calculation results:

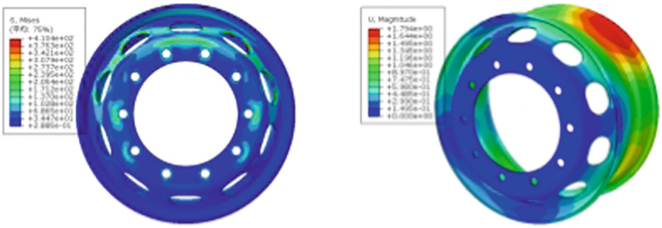
F_r = 60000 N.

3.2 Analysis of Radial Stiffness

The simulation results of von-mises stress plot and deformation plot under radial loads are shown in Fig. 7. The maximum deformation is 1.794 mm, at the outer edge of the rim, and the stiffness is calculated as 3.34×10^7 N/m.

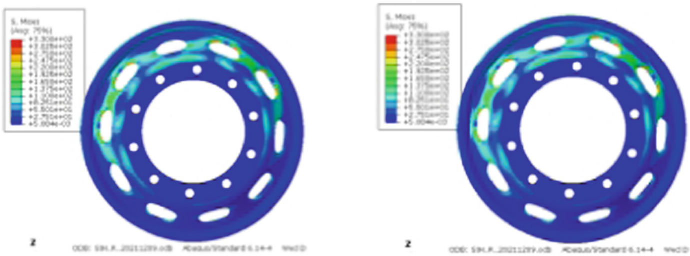
3.3 Design Optimization for Improving the Radial Stiffness of the Wheel

Usually, increasing the thickness of the rim can improve the wheel's radial stiffness. However, it will also put weight on the wheel. According to the CAD model, every 0.1 mm increase in the rim's thickness will result in a 0.41 kg increase in the wheel's weight, which does not meet the original goal. Therefore, structural optimization is adopted here to improve the radial stiffness.



(a) (b)

Fig. 7. Von-mises stress plot (a) and deformation plot (b) under radial loads.



(a) (b)

Fig. 8. Von-mises stress plot (a) and deformation plot (b) under radial loads after optimization.

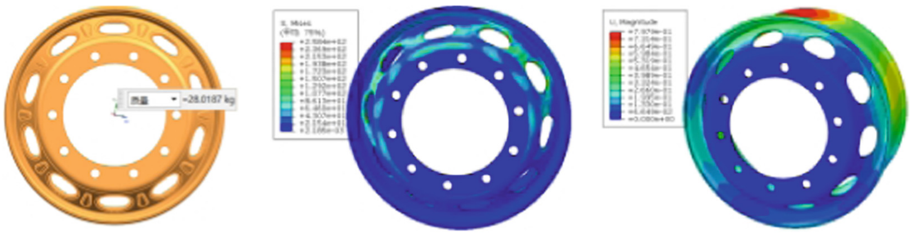
3.3.1 Optimization of Windows and the Simulation Results

The simulation results after optimization are shown in Fig. 8. The maximum stress under radial loads is 330 MPa. The deformation is 1.074 mm. The stiffness is calculated as 5.59 $\times$ 10^7 N/m, which is 2.25 $\times$ 10^7 N/m higher than that before optimization, increasing 67.36%.

3.3.2 Optimization of Windows and the Simulation Results

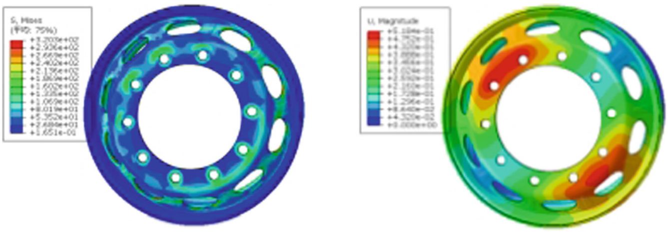
To obtain the optimal design, the size of the windows was unchanged at 90 $\times$ 70. Only 3 mm-height shuttle-shaped stiffeners were set up between the windows, as shown in Fig. 9(a).

The simulation results are shown in Fig. 9(b) and 9(c). After the optimization, the maximum stress under radial loads is 258.4 MPa. The maximum deformation is 0.7979 mm. The stiffness is calculated as 7.52 $\times$ 10^7 N/m, which is 4.18 $\times$ 10^7 N/m higher than that before optimization, increasing 125.15%.



(a) (b) (c)

Fig. 9. The weight of wheel optimized design (a), von-mises stress plot under radial loads (b) and deformation plot under radial loads (c).



(a) (b)

Fig. 10. Von-mises stress plot (a) and deformation plot (b) under bending and radial loads after optimization.

4 Analysis of the Optimizations

According to the above simulation results, the optimization in terms of improving wheel bending and radial stiffness is effective. However, as a critical part and safety component of the vehicle, the wheel needs to both meet the stiffness and stress requirements. Therefore, it is necessary to conduct stress analysis on the optimized wheel under bending and radial loads. The simulation results are shown in Fig. 10. The maximum stress is 320.3 MPa under bending loads, which is 77 MPa higher than that before optimization, increasing 31.65%. The maximum deformation under bending loads is 0.5184 mm. The stiffness is calculated as 4.20×10^7 N/m, which is 0.36×10^7 N/m lower than before optimization, decreasing 7.89%.

Comparison among the three optimization methods is listed in Table 1. Considering that setting up stiffeners will need more molds and equipment, resulting in an increase in manufacturing cost. Thus, the optimization of the size of the windows is fostering.

Table 1. Comparison among the three optimizations.

Optimization	Simulation results under bending loads			Simulation results under radial loads			Weight kg
	Stress MPa	Deformation mm	Stiffness N/m	Stress MPa	Deformation mm	Stiffness N/m	
Original design	243.3	0.4773	4.56×10^7	410.4	1.794	3.34×10^7	27.97
Windows 85 × 70	236.3	0.4630	4.70×10^7	330	1.074	5.59×10^7	28.13
3mm-height shuttle-shaped stiffeners	320.3	0.5184	4.20×10^7	258.4	0.7979	7.52×10^7	28.02

5 Conclusion

By optimizing the structure of the spoke and the size of the windows and the results of computer simulation, the bending and the radial stiffness of the wheel can be improved. The mentioned results are of important reference value for optimization design of hot-formed lightweight wheels. When doing product design in practice, it is possible to select the optimal solution based on the above-mentioned analysis and optimization to align with the product requirements - materials, loading conditions, appearance, costs, etc.

Acknowledgements. The work is supported by grant 2-4570.5 of the Swiss National Science Foundation.

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