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MECHANICAL TESTING METHODS CONCERNING THE STRESS ANALYSIS FOR A VEHICLE WHEEL RIM

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ABSTRACT

Wheels have vital importance for the safety of the vehicle and a special care is needed in order to ensure their durability. The development of the vehicle industry has strongly influenced the design, the material selection and the manufacturing processes of the wheels. The wheels loading manner is a complex one; further improvement and efficient wheel design will be possible only if their loading will be better understood.

In this paper, the car rim is analyzed with the finite element method, using the 40^0 loading test. The static stresses are studied in order to find the zones with higher stress concentration and to suggest the better design solution. The results have been compared to those obtained using an experimental stand. Finally, the Wöhler curve for the car rim is obtained.

Keywords: FEM, experimental, stress analysis, durability, Wöhler curve

1. INTRODUCTION

Wheels have a vital importance for the safety of the vehicle and special care is needed in order to ensure their durability. The development of the vehicle industry has strongly influenced the design, the material selection and the manufacturing processes of the wheels. They are loaded in a complex manner and further improvement and efficient wheel design will be possible only if their loading will be better understood.

In order to achieve an optimum design of the wheel, two requirements are needed: the precise knowledge of the loading and the mechanical properties and allowable stresses of the material, which depend on the vehicle characteristics, service conditions and manufacturing processes. Today, most manufacturers develop the wheel design based on results of the traditional dynamic radial fatigue test, also called the rim roll test, and on the dynamic cornering fatigue test, also called the rotating bending test [1, 2].

Another possibility is to use the finite element method in order to establish the stresses in the car rim and to compare the different design solutions. However, the modeling of the real loading of the rim cannot be accurate enough and that is why the results should be used with precaution. Anyhow, this method is very useful for comparing different design solutions and, therefore, foe slecting the car rim that should be tested further [3].

In this paper, the car rim is analyzed with the finite element method, using the 40^0 loading test. The static stresses are studied in order to find the zones with higher stress concentration and to suggest the better design solution. The results have been compared to those obtained using an experimental stand.

2. THEORETICAL MODEL

The car rim is analyzed with finite elements using the ANSYS program [4]. The geometrical conditions imposed to the rim wheel and the finite element model, with 5322 nodes and 5421 elements, are presented in Fig. 1. The finite element modeling was done using four node plate elements with 5 degrees of freedom per node and a constant thickness. In the areas of maximum stress concentration, the dimensions of the smallest elements are about 2.5 mm x 2.5 mm.

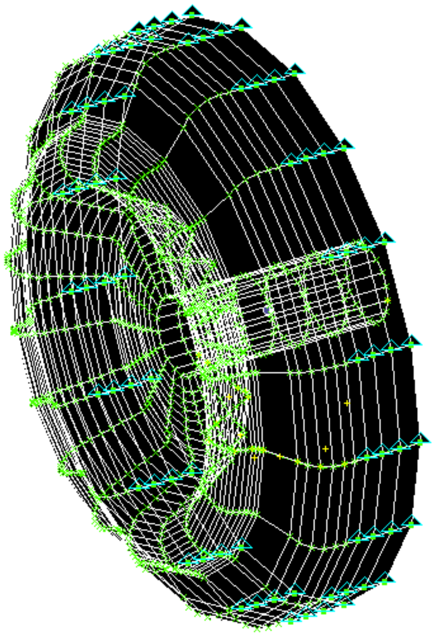

Fig. 1. The finite element model of the rim

The material considered for the rim was the steel sheet grade FePO3Bm, with 4 mm thickness. According to SR EN 10130/95 standard, the mechanical properties of this material are: the conventional yield limit, $\sigma_{0.2}$ = 330 MPa; the ultimate strength, σ_r = 440 MPa; the longitudinal elasticity modulus, $E = 2.1 \cdot 10^5$ MPa; the transversal elasticity modulus, $G = 8.1 \cdot 10^4$ MPa; Poisson's ratio $\nu = 0.33$.

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The loading applied to the rim (Fig. 2), corresponding to the 40^0 loading method, consists of an axial force (1500...3000 N), a radial force (2200...4300 N) and a bending moment (600...1300 N·m). This kind of load is equivalent for the turning simulation test [5].

The theoretical hypothesis are:

- the car rim will be mashed in finite elements only in the area of the central disk;
- the boundary conditions are imposed by the existence of the soldering cordons between the disk and the rim (all six degrees of freedom of the adjacent nodes are blocked).

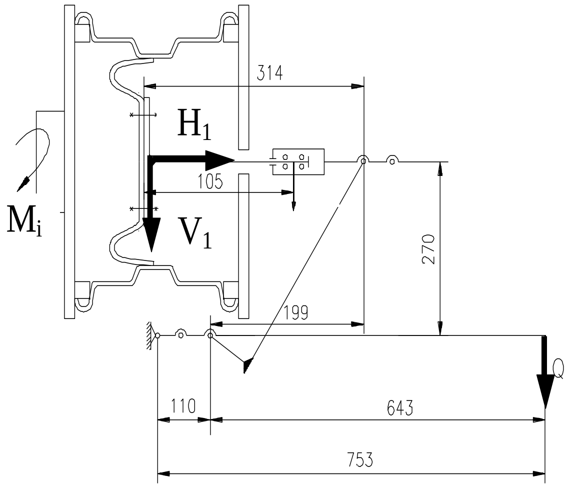

Fig. 2. The loading applied to the car rim

3. FINITE ELEMENTS RESULTS

The calculus model permits the determination of the maximum stress that produces the fatigue break of the car rim.

The finite element analysis was conducted in two stages. The first step is the analysis of the state of stresses in the central area of the rim, for the particular conditions of the loading, characteristics for the initial version of the car rim. In this case, the charges that have produced the crack (Fig. 3) were V_1 = 4134 N, H_1 = 2986 N and M_i = 1298 N·m, which correspond to an axial-radial loading mass m = 60 kg. Figure 4 shows the loading

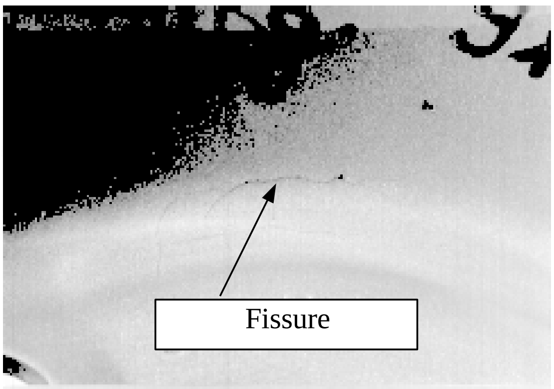

Fig. 3. Fissure in the area of the central disk

scheme corresponding to this charge.

The stress distribution in the rim (Fig. 5) is characterized by the existence of two asymmetrical zones of maximum stresses $\sigma_{max\ 2} < \sigma_{max\ 1}$ and two symmetrical zones of equal minimum stresses σ_{min}. The solicitation cycle in the central aria (Fig. 6) is a pulse cycle, characterized by $\sigma_{max\ 1}$ = 140 MPa, $\sigma_{max\ 2}$ = 77 MPa and σ_{min} = 29 MPa.

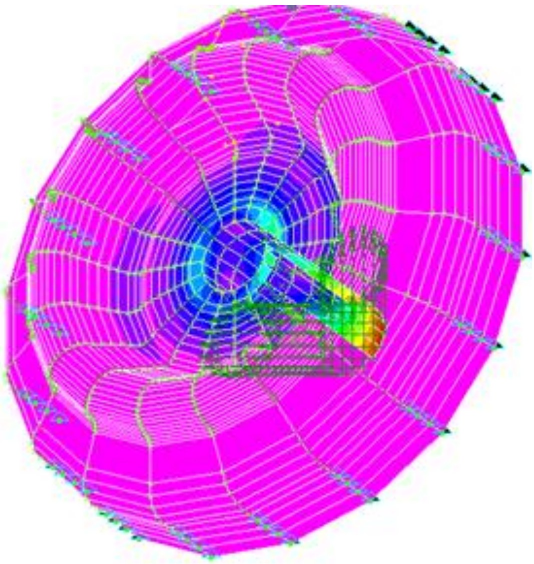

Fig. 4. Loading scheme on the rim

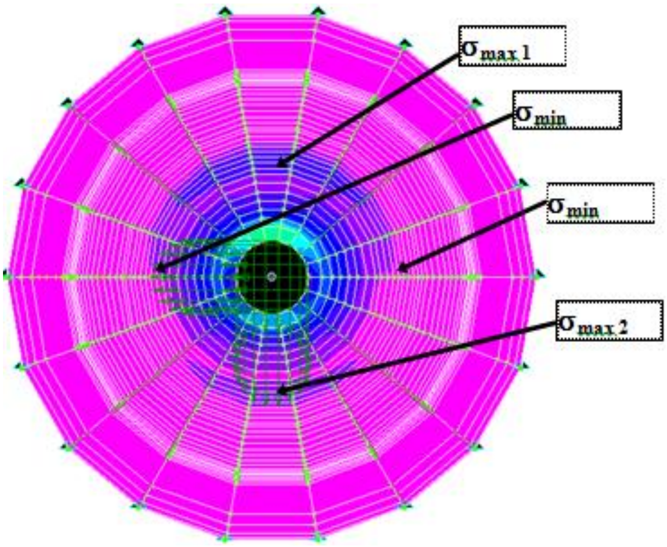

Fig. 5. Stress distribution in the area of the central disk (initial version)

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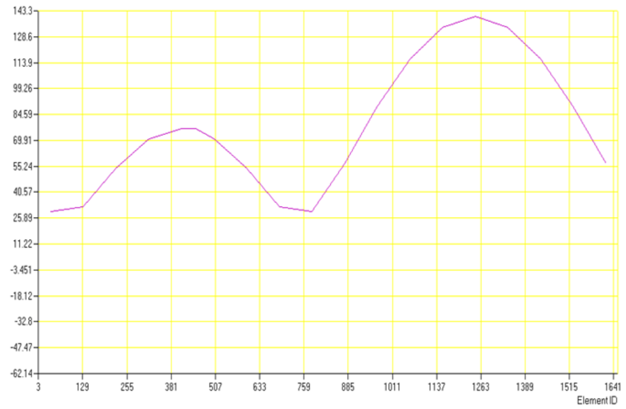

Fig. 6. The solicitation cycle in the central aria (initial version)

The second step consists in analyzing the state of stresses in the central area of the rim (Fig. 7), for the same loading, in the case of the optimized version of the rim: a greater radius of curvature ($R_{initial}$ = 30 mm versus $R_{modified}$ = 37 mm) in the central area of the disk.

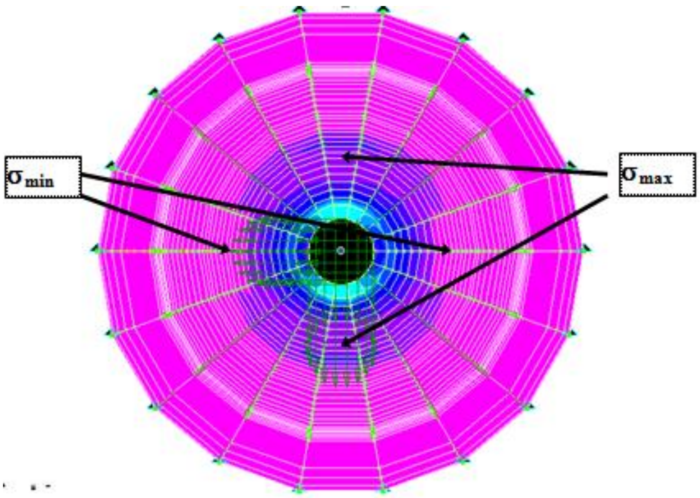

Fig. 7. Stress distribution in the area of the central disk (optimized version)

Thus, the stress distribution in the central area is roughly symmetrical, with two regions characterized by maximum stresses of σ_{max} = 108 MPa and two regions with minimum stresses of σ_{min} = 33 MPa (Fig. 8).

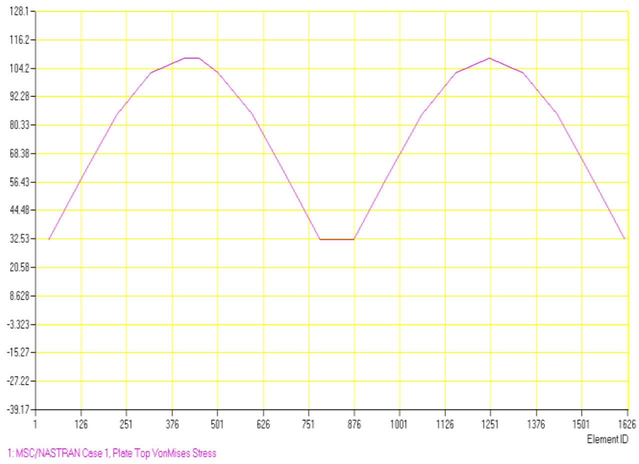

Fig. 8. The solicitation cycle in the central aria (optimized version)

Taking account of these preliminary tests, the theoretical study has been extended for a large domain of the loading mass, between 30 kg to 60 kg. The theoretical results are presented in Table 1.

Table 1. Equivalent stresses in the central area of the optimized car rim

Loading mass m, [kg]	**Equivalent load Q, [N]**	**Effectiv load in the car rim**			**Maximum stress σ_{max} [MPa]**	**Minimum stress σ_{min} [MPa]**
		Radial load V_1, [N]	**Axial load H_1, [N]**	**Bending moment M_i, [Nm]**		
30	294	2257	1483	658	37	14
35	343	2593	1730	763	42	17
40	392	2928	1978	868	63	19
45	441	3264	2225	974	78	22
50	490	3599	2472	1079	89	24
55	540	3941	2724	1186	101	27
60	592	4297	2987	1298	108	33

4. EXPERIMENTAL RESULTS

The purpose of the experimental tests, realized on the 40^0 loading stand (Fig. 2), was to validate the proposed theoretical model. For each loading mass, three car rims were tested, until the first crack appears. The results are presented in Table 2.

Table 2. Experimental results for the car rim durability

Loading mass, kg	Von Mises equivalent stresses		Mean life time, h	Dispersion of the life time, h
	σ_{max}, MPa	σ_{min}, MPa		
47	80	23	15h 37’	52’
50	89	24	11h 15’	1h 44’
55	101	27	8h 43’	45’
60	140	33	5h 47’	41’

The experimental results are numerically treated using the regression analysis method [6] and the Wöhler fatigue curve is presented in Figure 9.

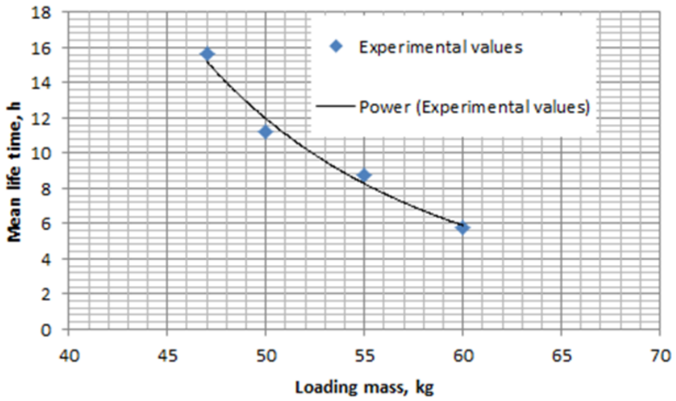

Fig. 9. Experimental results

The regression equation between lifetime (T) and mass (m) attached to this curve is: $T = 5 \cdot 10^7 \cdot m^{-3.881}$, with the correlation coefficient equal to 0.9855.

5. CONCLUSIONS

The theoretical model, realized with the finite element method, demonstrates the existence of two zones with high stresses, disposed in the central area of the disk. These stresses are responsible for the fatigue breaks of the rim.

The experimental results confirm the existence of these zones, where the fissures appear.

Following optimization of the car rim, the authors managed to reduce supplementary costs, eliminating the hub of stresses and increasing the reliability of the rim.

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