Nexus : Journal of Innovative Studies of Engineering Science (JISES)
Volume: 02 Issue: 05 | 2023 ISSN: 2751-7578
http://innosci.org/

Assessment of the Strength Properties of Solid-Rolled Wheels of Type 2 Freight Wagon, Taking into Account the Residual Process Stress

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Annotation: The solid-rolled wagon wheel is one of the most important bearing parts of the running gear of the railway rolling stock, on which the safety of train traffic depends. The paper presents the coefficient of strength, taking into account residual stresses in the transition zone of the disk part into the rim of solid-rolled wheels of freight and passenger cars by the calculation method.

Key words: residual technological stresses, strength, wheelset, safety factor, fatigue crack.

Introduction

Currently, the main directions of rolling stock modernization are: - the use of new materials and structures in the repair and manufacture; – increase in axle load up to 27 tf for new generation locomotives and freight cars; – reduction of tare weight of freight cars by 25%. Widely used domestic wheel designs for freight and passenger cars, locomotives and metro cars, designed and tested for lower axle loads and design speeds, have more than half a century of development history. The used standard wheel designs are characterized by a high weight compared to the closest Western counterparts, or do not meet the requirements of strength and reliability when operating under rolling stock with increased payload [1].

The wheel bogie of railway passenger and freight cars is an important bearing part, which must withstand both vertical and transverse dynamic forces created by a moving train.

The reasons for the appearance of fatigue cracks in the disk part of the wheels are uneven rolling or a slider on the tread surface, which create increased dynamic loads on the disk part of the wheel, metal fatigue and internal residual technological stresses formed as a result of manufacturing.

It should be noted that in the standards [2, 3], the calculation of the strength of the wheels of freight and passenger cars is carried out taking into account different values of the thickness of the rims (as a result of turning after a certain run), as well as the probable appearance of sliders on the tread surface or uneven rolling.

Table 2 Mises stresses in the design zone of the wheel arising during the service life

j	Wagon condition	Active loads	Proportion of wheel movement, λ	Average von Mises stress in the calculated zone of the wheel at n turning, MPa:				
				n = 0	n = 1	n = 2	n=3	n=4
1	laden	P1=142,83 kN	0,2514	45	50	55	57	60
2		P2=196,88 kN	0,045	55	60	65	72	77
3		P3=273,8 kN	0,0036	75	85	90	100	110
4	empty	P1=37,26 kN	0,1676	10	11	12	15	16
5		P2=51,36 kN	0,03	15	18	19	21	22
6		P3=182,7 kN	0,0024	50	55	60	70	75

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7	laden	P1 =142,83 kN Q=21,7 kN	0,2514	48	53	60	64	59
8		P2 =196,88 kN Q=21,7 kN	0,045	60	70	78	82	92
9		P3=273,8 kN Q=21,7 kN	0,0036	80	90	100	110	120
10	empty	P1 =37,26 kN Q=5,67 kN	0,1676	10	12	15	17	18
11		P2=51,36 kN Q=5,67 kN	0,03	15	19	21	22	24
12		P3=182,7 kN Q=5,67 kN	0,0024	55	60	67	73	76
Note: P_1- for a wheel without a defect on the tread surface; P_2- for a wheel with a slider; P_3- for a wheel with uneven rolling								

This table shows the average von Mises stresses calculated using the SolidWorks/Simulation software.

In [2], the vertical P and transverse loads Q acting on the wheel are calculated, which act in operation, taking into account the state of the wheel (without defects, with a slider, with uneven rolling), the movement of the wheel in any state during the service life, the state of the car (see Fig. table 1).

The consequences of the destruction of the wheels in motion can be the most severe [4]. Therefore, the accuracy of determining the safety factor of the car bogie wheel at the design stage affects the safety of train traffic and is an urgent task. The paper sets the task of calculating the safety factor of the wheel of a freight car bogie with a conical disk in accordance with the current standards [2].

The solution of the task was carried out on the basis of the norms for the calculation and design of cars with an axle load of 23.5 tf.

However, the refinement of the values [5] of the average endurance limit by the amplitude of the disk in the design zone was made under the steady state mode of variable loading on the basis of tests of 10^8 cycles, the blocks of design loads were corrected.

The algorithm for calculating the wheel safety factor is implemented in the MathCad program.

The calculation scheme of the wheel is shown in Fig. 1.

$$n=\frac{\sigma_к^а}{\sigma_{аэ}}; \quad (1)$$

$\sigma_к^а$ — the average endurance limit in amplitude of the full-scale wheel disk in the design zone under steady state variable loading on the basis of 10^8 cycles,

$\sigma_{аэ}$ — is the calculated value of the amplitude of conditional stationary loading of the wheel, reduced to the base of 10^8 cycles, the effect of which in terms of damage accumulation is equivalent to the actual non-stationary operating mode for the estimated life of the wheel [6].

$\sigma_к^а$ is found, taking into account the reduction factor of 0.75, from the ratio:

$$\sigma_к^а = 0.75(\sigma_к^{max} - \sigma_к); \quad (2)$$

Where $\sigma_{к}^{max} = 0.5\sigma_{в} = 0.5 \cdot 910 = 455$ MPa - is the maximum stress of the cycle when determining the endurance limit of wheel steel grade T according to GOST 10791 - 2011, [7] depending on the tensile strength;

$\sigma_{к} = \frac{\sigma_{к}^{max} + \sigma_{к}^{min}}{2} = 250\ MPa$ - the average stress of the cycle when determining the fatigue limit of grade T steel with asymmetry coefficient r = 0.1 [8].

From formula (2) $\sigma_{к}^{а} = 154\ MPa$

The calculated value of the amplitude in the zone of interface between the disc and the rim is calculated by the formula:

$$\sigma_{аэ} = \sqrt[m]{\frac{N_c}{N_0} \cdot \frac{0,5}{1+k_n} \cdot \frac{1}{n} \sum_{i=1}^{n} \sum_{j=1}^{12} \frac{\lambda_j}{\sqrt{2\pi} S_{\sigma_{ij}}} \int_{\sigma_{min}}^{\sigma_{max}} \sigma^m e^{-\frac{(\sigma - \bar{\sigma}_{ij})^2}{2S_{\sigma_{ij}}^2}} d\sigma}, \quad (3)$$

where m=9 is the degree of the wheel fatigue curve;

$N_0 = 10^8$ – basic number of cycles;

$N_c = 30 \cdot 10^8$ - the total number of vertical load cycles for the estimated service life

wheels;

σ_{min}, σ_{max} - minimum and maximum values of stress amplitudes in the design zone;

j - is the number of the load block;

i - is the serial number of the calculated rim thickness option;

$K_п$ - is the ratio of the empty run of the car;

n=5 - number of variants of wheel rim thickness, taking into account turning during repair;

λ_j - is the proportion of wheel movement without defects, with a slider and uneven rolling in the total run wheels, [9] respectively λ_1=0.838; λ_2=0.150; λ_3=0.012;

$\bar{\sigma}_{ij} = \sigma \bar{P}_j \left(1 + 0,683 \frac{\bar{Q}_j}{\bar{P}_j}\right)$ – average value of stress in the block;

$S_{\sigma_{ij}} = \sigma S_{P_j} \left(1 + 0,683 \frac{S_{Q_j}}{S_{P_j}}\right)$ - standard deviation in the block.

The standard deviations calculated according to [10] were: S_{P_1}=26,5 kN - for a wheel without defects; S_{P_2}=21,2 kN - for a wheel with a slider; S_{P_3}=62,7 kN; S_Q=14,2 kN.

The damaging amplitudes are $\sigma_{min} \geq 0,5\sigma_к$. If $\sigma_{max} \leq 0,5\sigma_к$, then the value of the integral in formula (3) was taken equal to zero [11].

In expression (3) it was accepted: $\sigma_{min} = \sigma_j - 2,5\ S_{\sigma j}$; $\sigma_{max} = \sigma_j + 2,5\ S_{\sigma j}$.

The loads given in table 1 were applied to the wheel model developed in the SolidWorks program, according to Fig.1. The wheel material was selected from the SolidWorks/Simulation finite element analysis database, the properties of which corresponded to grade T in accordance with GOST 10791 - 2011. The wheel fastening was assigned based on the design scheme of the wheel [1]. After that, a finite element mesh was automatically generated from tetrahedral 10-node finite elements with 3 degrees of freedom at each node. The total number of finite elements in model 40586. As a result of the acting loads given in Table 1, the von Mises stresses in the design zone of the wheel were determined and summarized in the same table.

- a solid-rolled wheel with a flat-conical disk, with a diameter of 957 mm along the tread circle, manufactured in accordance with GOST 10791-2011, grade T steel;

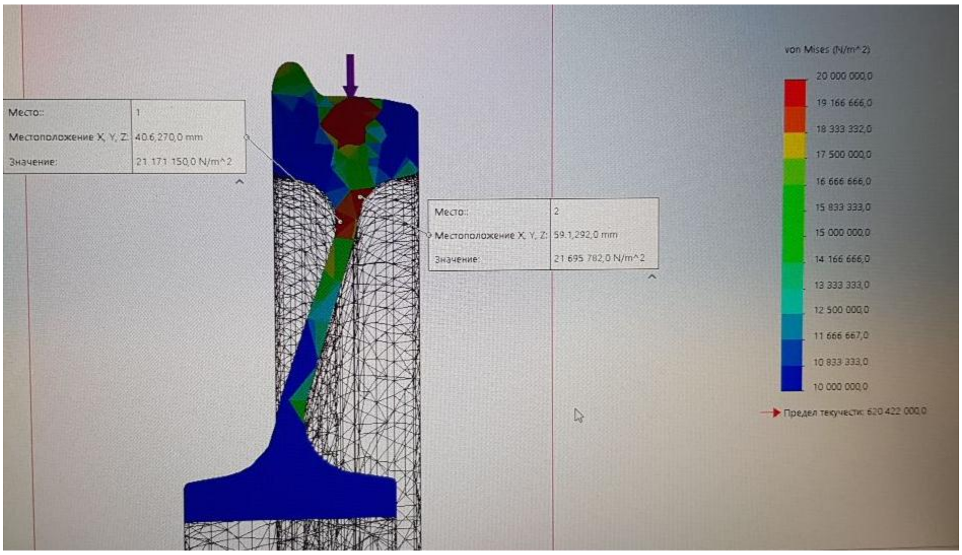

Figure 1. Plano disk under stress in Solidworks

The calculation of the safety factor of the wheel of the design zone for known average stresses that occur over the service life is implemented in the MathCad program. A fragment of the program code recording is shown in Fig. 2

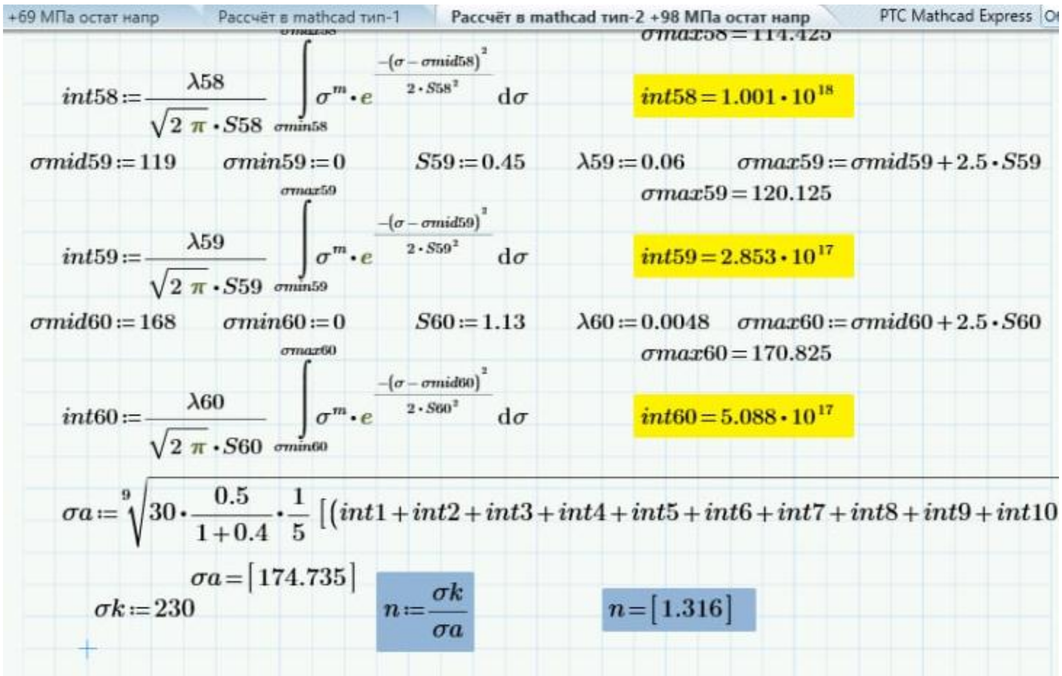

Figure 2. A fragment of the code entry in the Mathcad program for calculating the margin of safety of a wheel in the calculation zone according to formula 2

The result of the calculation, taking into account the axial residual technological stresses of the wheel, was n = 1.32, which can cause its breakage from loads not taken into account in the norms in operation.

The proposed method for calculating the safety factor is relevant only for the disc-to-rim transition zone.

Taking into account the fact that in operation there are failures of the wagon wheels associated with the appearance of fatigue cracks in the zone of transition of the disk into the rim before the end of the designated service life, residual technological stresses should be taken into account in the calculations of the wheels for strength from the action of dynamic loads.

From the studies carried out, the following conclusions can be drawn:

1. The compiled program code in MathCad, when exchanging data on stresses calculated in the SolidWorks/Simulation finite element package, made it possible to automate the calculation of the wheel strength;

2. The presented calculation algorithm, the obtained initial data, implemented in the MathCad program, can serve as a basis for assessing the strength of wheels with different disk configurations, as well as the level of residual technological and operational stresses.

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