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Analytical, numerical and bench tests of axles in rail vehicles

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Abstract. On the basis of experimental, analytical and numerical tests, a strength analysis of a rail vehicle axle was presented, as well as an alternative approach to this type of issue. The axle of the wheelset was tested on the experimental stand. Then, analytical calculations of the tested axis were performed in accordance with the EN 13 103-1 standard in the sections where strain gauges were located during the stand test. A numerical model was also created in a program based on the Finite Element Method. The obtained results were compared and summarized. It turned out that the results from all studies coincided, which suggests that each of the methods used is correct. None of the obtained values exceeded the permissible fatigue stresses.

Introduction

Wheelsets (axle + wheels) are the basic construction unit of a rail vehicle. Among many assemblies, they are the most exposed to fatigue wear, which was also strongly emphasized in articles [1, 2]. Sobaś [3] presented technological measures increasing the service life of wheelset axles on the basis of applicable standards. Michnej and Krwala [4] characterized the surface-reinforcement treatments that increase the durability of the axles of railway wheelsets. Whereas, Antolik [5] described the sources of fatigue cracks in railway axles. Many studies on the durability of railway axles and attempts to strengthen the axles show the essence of the problem. This is particularly important in the currently designed vehicles, where the main assumptions are to design a vehicle with the lowest possible weight, moving at higher and higher cruising speeds.

The current applicable European standard EN 13 103-1 [6] containing the necessary rules for the construction and testing of wheelset axles clearly suggests that axle strength calculations should be performed using the analytical method. Thanks to the rules contained there, it is possible to correctly perform a mathematical model of the tested object and correctly determine the forces and boundary conditions for analytical calculations. Nevertheless, in this work, in order to compare the analytical method and experimental tests with FEM simulation, it is necessary to thoroughly understand the operation of the machine for testing the fatigue strength of axles of wheelsets, which was presented in detail in Stasiak's academic textbook [7].

Nowadays, analytical methods are often superseded or only supplement the finite element method, which is more accurate and has a wider range of applications. Similarly, in the case of axis calculations, it seems necessary to adapt the current approach to modern simulation tools. An alternative axis calculation method would be a bridge between analytical calculations resulting from the standard and simulation methods, which would significantly shorten the calculation time and enable multi-directional analyses.

Tested axle

The object of research of this work is the non-powered axle of the type A wheelset (PN-92/K-91048) [8]. The axle is made of EA1N steel.

The values of permissible stresses are presented in table 1. The values have been selected according to the European standard EN 13103-1 [6] and they result from the fatigue limit at rotational bending for the axis. It takes into account the safety factor S = 1.2 and the fact that the places where we will make the measurement are outside the embedment areas.

Table 1 Maximum permissible stresses for solid axles [6]

Steel	σ_{dop}	σ_{dop} S=1.2
	[MPa]	[MPa]
EA1N	200	166

Bench tests

Bench tests were carried out at the Łukasiewicz Research Network - Poznań Institute of Technology in the Rail Vehicles Testing Laboratory at the 18SB test stand intended for testing the fatigue strength of wheelsets axles. For this purpose, a sample was delivered to the plant in the form of a half-set, which consisted of an axle and one pressed-in wheel. The wheel was pressed onto the axle with the applicable dimensional tolerances and forces. The tests included the performance of a fatigue test in the range of 10 million cycles and strain gauge measurements to determine the stresses and control of these loads during the test. Strain gauge measurements belong to experimental methods of measuring deformations on the surface of the tested element. The centers of the electro-resistance strain gauges were respectively 300 and 250 [mm] from the center of the wheel (Fig. 1).

Fig. 1 View of the arrangement of resistance strain gauges

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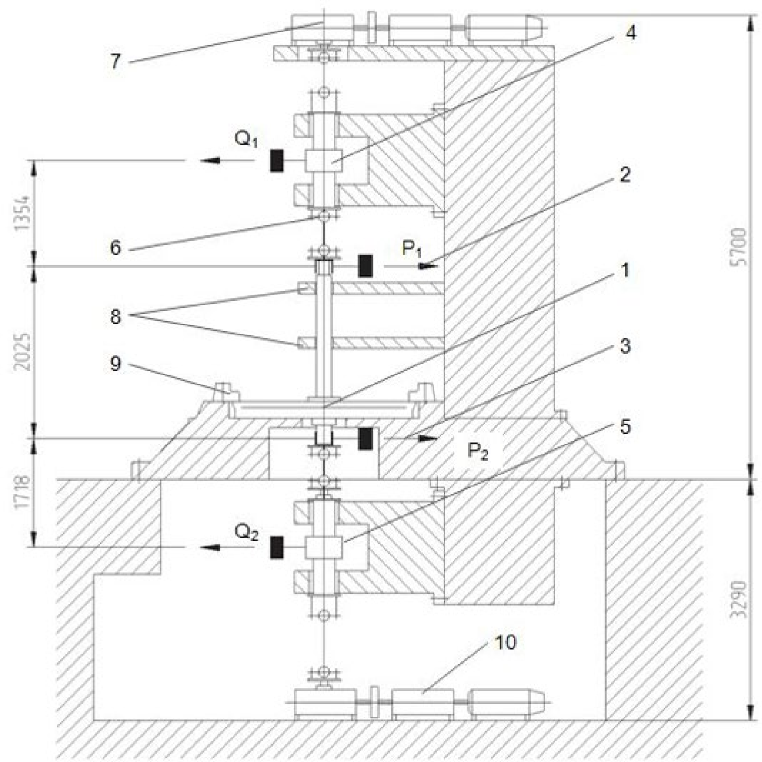
The structural system of the fatigue machine used for testing, shown in figure 2, uses the centrifugal forces of masses rotating with a constant angular velocity. A sample of the axle with actual dimensions was attached vertically to the machine body, and the rolling wheel mounted on the axle was attached with special anchor holders. Forces P_1 and P_2 (values: 37.4 and 81.0 [kN]), which directly loaded the sample, were caused by centrifugal pulsators mounted at its ends. The pulsators that caused the Q_1 and Q_2 forces were used to dynamically balance the forces that acted on the foundation of the machine. Both the upper and the lower pulsator are driven by a separate DC motor, powered by a common drive system from the control panel. In the middle part of the column there were safety sensors, which, in the event of contact with the axle sample, immediately switched off the drive system and stopped the machine. The machine was made at H. Cegielski in Poznań according to the construction documentation of the Research and Development Center of Rail Vehicles (now Łukasiewicz – PIT).



The machine includes:

1 - tested sample

2 - upper loading pulsator

3 - lower loading pulsator

4 - upper relief pulsator

5 - lower relief pulsator

6 - cardan shaft

7 - angular gear

8 – protection

9 - sample mounting

Fig. 2 Scheme of a fatigue machine for testing axles and wheels of wheelsets [7]

The tested set was mounted on the fatigue machine shown in figure 3 and statically loaded in the axis of the upper pulsator with a concentrated force corresponding to the centrifugal force of the pulsator. This procedure is necessary to determine the important parameters for setting the loads required in the fatigue test. During two measurement cycles, the axes were loaded with a force of 40 kN and the deflection arrow was measured in the axis of the upper pulsator and in the axis of the safety sensor, and strain gauges were measured in the control zones. In this way, the rotating masses of the loading pulsators were determined.

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Fig. 3 View of the fatigue machine for testing axles and wheels of wheelsets (Łukasiewicz-PIT archive)

The results obtained in experimental studies are presented in table 2.

Table 2 Deformations and stresses occurring at the locations of strain gauges

Strain gauge no	Distance	Deformation		Average amplitude	
		ε_{max}	ε_{max}	$\varepsilon_{śr}$	σ_a
	[mm]	[µm/m]	[µm/m]	[µm/m]	[MPa]
1 os	300	585.5	596.05	590.775	122
2 os	250	608.37	619.48	613.925	127

Analytical calculations

Analytical tests were carried out in accordance with the European standard EN 13103-1 [6]. As a standard, calculations are made considering two types of forces coming from masses in motion and braking. The concentration of stresses in the cross-sections of the axles most exposed to overloads is checked. Nevertheless, the purposes of comparison with other test methods, two cross-sections in which the strain gauges were placed during the bench test were analyzed analytically.

The input data needed for the calculations are given in Table 3.

Table 3 Results

Force on the upper pulsator	P_1	37.40	kN
Force on the lower pulsator	P_2	-81.00	kN
Wheel rolling radius	R	0.46	m
Distance of the upper pulsator axis to the center of the wheel	L_g	1.74	m
Distance of the lower pulsator axis to the center of the wheel	L_d	0.26	m
Top moment at the center of the rim width	$M_g = P_1 \cdot L_g$	65.15	kN·m
Bottom moment at the center of the rim width	$M_d = P_2 \cdot L_d$	-20.90	kN·m
Moment applied to the wheel disc M_1-M_2	$M_t = M_g - M_d$	86.05	kN·m
Force at the periphery of the circle	$P_t = \frac{M_t}{R}$	187.06	kN
Distance of the wheel from the „2os" strain gauge on the axle	L_{i1}	0.25	m
Distance of the wheel from the „1os" strain gauge on the axle	L_{i2}	0.30	m
Axle dimeter behind the wheel	d	0.160	m
Strength index	$W = \pi \cdot \frac{d^3}{32}$	0.000402	m^3
The bending moment of the axis in the place of the strain gauge „2os"	$M_1 = P_1(L_g - L_{i1})$	55.80	kN·m
The bending moment of the axis in the place of the strain gauge „1os"	$M_2 = P_1(L_g - L_{i2})$	53.93	kN·m
Stresses in the axis at the place strain gauge „2os" <166	$S_1 = \frac{M_1}{W \cdot 1000}$	139	MPa
Stresses in the axis at the place strain gauge „1os" <166	$S_2 = \frac{M_2}{W \cdot 1000}$	134	MPa
The ratio of the moments Mg/Md	t	-3.12	-

MES research

Currently, the Finite Element Method is the most used tool for numerical analysis of structures. It is a discrete method that solves differential equations in an approximate way. The use of FEM allows for significant savings by eliminating many bench tests. Thanks to it, it is also possible to properly prepare the model for experimental research by determining the appropriate measurement parameters or indicating the most favorable location of strain gauges.

The strength analysis was performed with the ABAQUS/Standard program based on the Finite Element Method at the Łukasiewicz Research Network - Poznań Institute of Technology.

The following units of measurement were used in the calculations: [mm], [N], [MPa].

Support conditions and loads P_1 and P_2 (Fig. 4) were introduced into the computational model, which are identical to the restraint of the wheelset during the bench test and the assumptions that

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were included in the mathematical model, so that the obtained results can be compared as precisely as possible from all conducted studies.

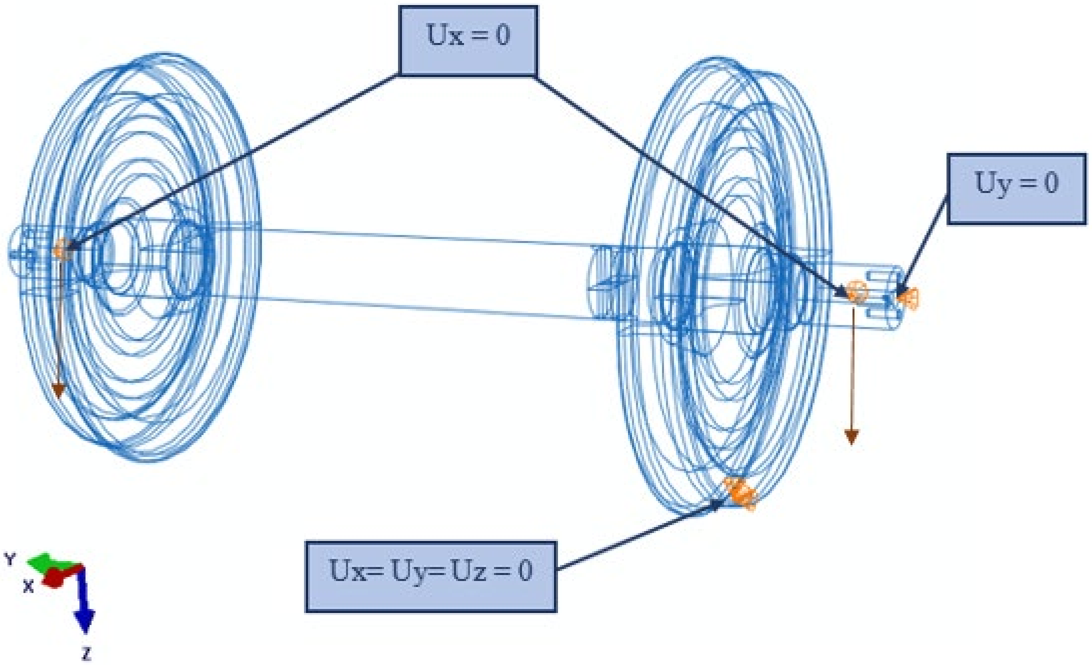

Fig. 4 Scheme of modeling the boundary conditions of the tested object

Before performing the calculations, a study of the size and type of the finite element was carried out. The stress calculation results were examined and compared by changing the types of finite elements on the axis. Various finite elements were checked and the results of the calculations turned out to be were very similar to each other. It was decided to use tetragonal square elements in accordance with the guidelines in [9]. This allows to reproduce all the details of the tested axis with the greatest possible accuracy and to easily determine the points where the strain gauges were placed during stand tests. The results obtained with the FEM method are shown in figure 5.

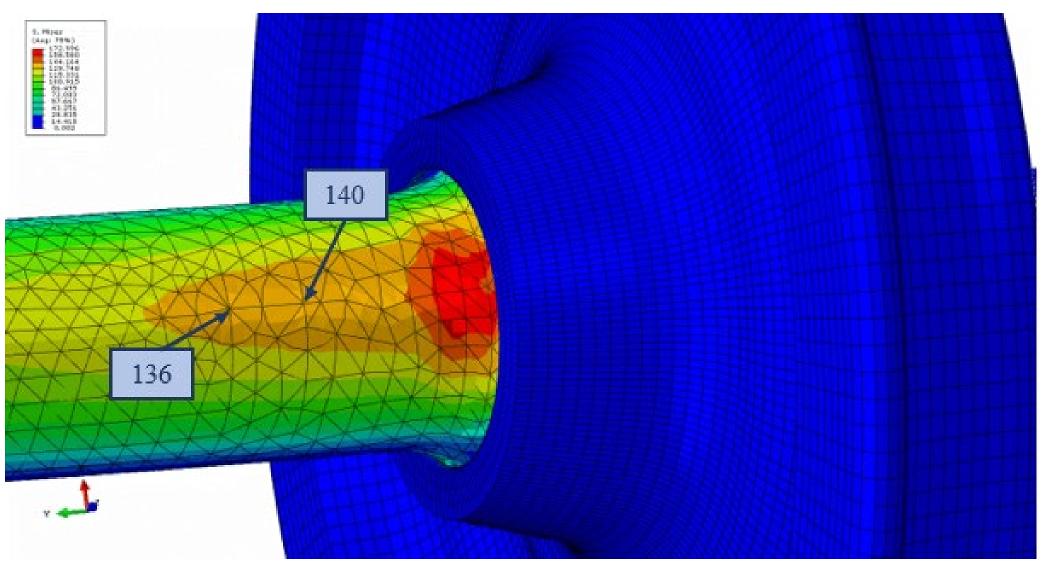

Fig. 5 Huber-Mises stress distribution in the wheelset model in the locations of strain gauges

Summary

Comparison of the results obtained from analytical calculations, FEM numerical tests and bench tests with limit values is presented in Table 4.

Table 4 Summary of research results

Strain gauge no	Distance	Bench research	Calculations analytical	FEM numerical research	Permissible stresses [6]
		σ_a	σ_a	σ_a	σ_{dop}
	[mm]	[MPa]	[MPa]	[MPa]	[MPa]
1 os	300	124	134	136	166
2 os	250	129	139	140	166

The stress values obtained from analytical calculations of the tested axle of the rail vehicle and during numerical tests based on the Finite Element Method, as well as bench tests, are almost identical, which suggests the correctness of determining the boundary conditions and loads in computer analysis using the Abaqus/Standard program. None of the obtained values exceeded the permissible stress (taking into account the safety factor equal to 1.2), i.e. 166 MPa.

Thanks to the conducted research, it can be concluded that numerical analysis based on the finite element method reflect the real results that were obtained by means of deformations determined on the tested axle sample on the experimental stand and during analytical calculations in accordance with the standard [6].

On the basis of the conducted research, it can be unequivocally stated that simulation studies can be an alternative or a great support for analytical methods. Thanks to the FEM analysis, we can check the distribution of stresses that arises over the entire tested structure, and not only in a few potentially most endangered cross-sections. Using computer simulation, we can additionally introduce various types of structural modifications much faster and check how a given change will affect the strength of the tested object. In many areas, the finite element method has completely supplanted analytical calculations. Also, in the case of axles of wheelsets in rail vehicles, it seems necessary, for example, to extend the existing guidelines for axle fatigue strength analysis with FEM simulations. The discussed extension of the strength analysis will significantly facilitate and accelerate the work on the design of the axles of rail vehicles and will enrich the methodology of approach to such issues.

To enrich the strength analysis of rail vehicle axles carried out so far, will be to carry out bench tests with more strain gauges. It gives us a more accurate picture of the stress distribution in the real axle located on the experimental stand and compare it with the results of analytical calculations and FEM simulations.

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