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Designing of railway wheels - Part 2. Comparison of numerical analysis and experimental research

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Abstract: At present, the design process can be simplified and accelerated, if computer simulation basing on finite element method (FEM) is used. FEM numerical calculations of different wheels of railway wheelsets are investigated in Department of Railway Engineering. The justification for undertaking this issue is that the methodology of design of railway wheelsets both in Poland and abroad In order to assess the calculation model experimental tests of static loads (due to carriage weight) and thermal loads (due to braking) have been run. The results of these tests have been compared with results of numerical analysis. The proposed calculation model has been compared with results of test stand investigation. Comparison of the results has confirmed the correctness of the proposed discrete model and calculation algorithm.

In the framework of the common scientific-didactic projects of Railway Engineering Department and Central Rail Research Institute in Moscow and producer of railway wheelsets – BONATRANS a.s. in Bohumin (Czech Republic), the method was worked out, and the experimental researches were conducted on specialistic certified stands for railway wheelset researches [1,2,3].

Keywords: finite elements method, numerical analysis, experimental tests wheelsets, static loads, dynamic loads, thermal loads, stress, deformation

NOTATION

E	Young's modulus
v	Poisson's number
ε_R	radial deformation
ε_T	circumferential deformation
σ_R	radial stresses
σ_T	circumferential stresses

Abbreviations

UIC International Union of Railways

1. INTRODUCTION

The experimental investigation of the temperature, deformation and stress fields of the railway wheels has been conducted in order to check the accuracy of numerical calculations. They have covered:

- static loads simulating the carriage weight for different load values and different load models,
- thermal loads for 13 braking cycles, braking power of 20, 30, 40 and 50 kW.

In the tests German and Russian wheels have been used.

2. STATIC EXPERIMENTAL INVESTIGATION OF A RUSSIAN WHEEL

The experimental tests have been conducted in the Rail Transport Institute ВНИИЖТ in Moscow. Test stand for strain measurements of railway wheels deformations has been used – Fig. 1a.

Strain gauges have been placed on the internal and external surface of the web of a wheel in the vertical plane (Fig.1b). The wheel has been loaded with the vertical force (Fig.1b) of 800 kN. The previously acquired experience of the Rail Transport Institute ВНИИЖТ arising from analysis of fractures of wheels investigated in the test stand and wheels broken down in service has led to a conclusion that the interaction of two forces (horizontal and vertical – simulating interaction between track and wheel) can be exchanged for one vertical force.

The result has been obtained in the form of diagrams showing the courses of radial stresses at the internal and external wheel surfaces – Fig.2.

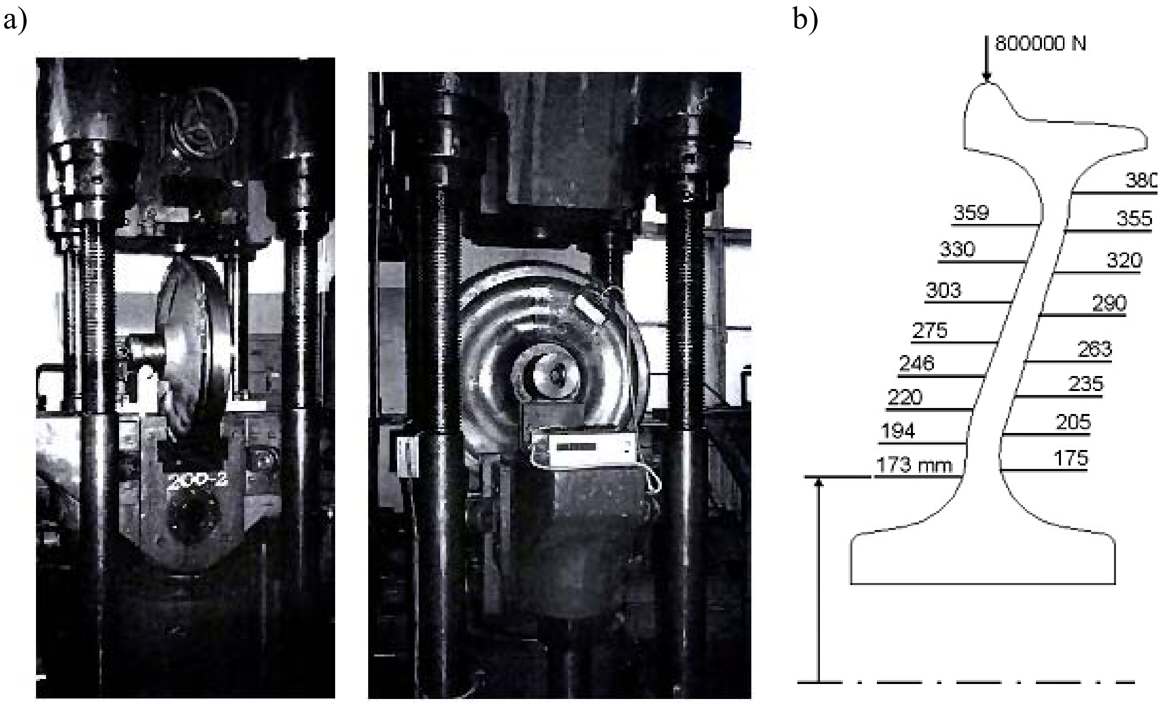
In order to facilitate the comparison of results of experimental tests and numerical computations, the values of radial deformation of the web of a wheel have been calculated with the help of the following formulas (the calculations have been conducted for the strain gauges locations):

$$\sigma_R = \frac{E}{1-\nu^2}\left(\varepsilon_R + \nu\varepsilon_T\right), \qquad \sigma_T = \frac{E}{1-\nu^2}\left(\varepsilon_T + \nu\varepsilon_R\right) \tag{1.1.}$$

where:

E –Young‘s modulus (2.1 e11 Pa),
ν –Poisson’s number (0.28)
ε_R – radial deformation
ε_T – circumferential deformation
σ_R – radial stresses
σ_T – circumferential stresses.

Calculation results have been shown in Fig.3. The outcome of the numerical analysis consists in deformation values of wheel in places corresponding to placing of the strain gauges – Table 1. Table 1 and Figs. 4 and 5 show that modelling the load as a single force does not exactly correspond to the experimental results. The discrepancy of the results is great


a)

b)

Fig. 1. Test stand for strain investigation of railway wheels deformation: a) the stand, b) placing of strain gauges, way of modelling the wheel

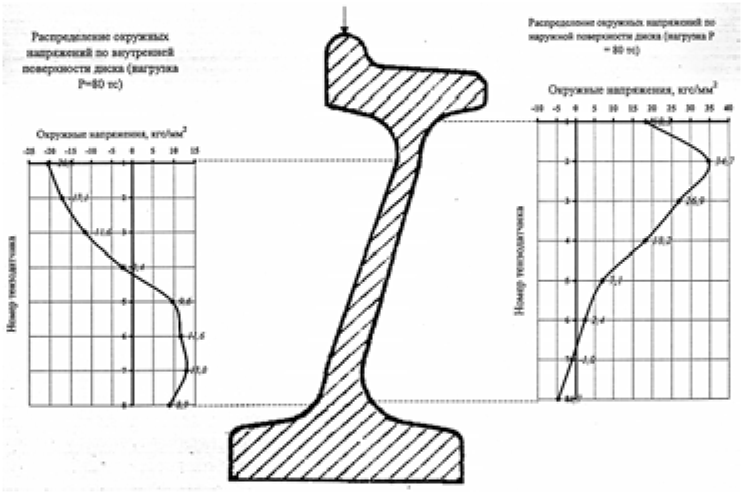


Fig. 2. The diagrams of the radial stresses at the internal and the external side of the web of a wheel

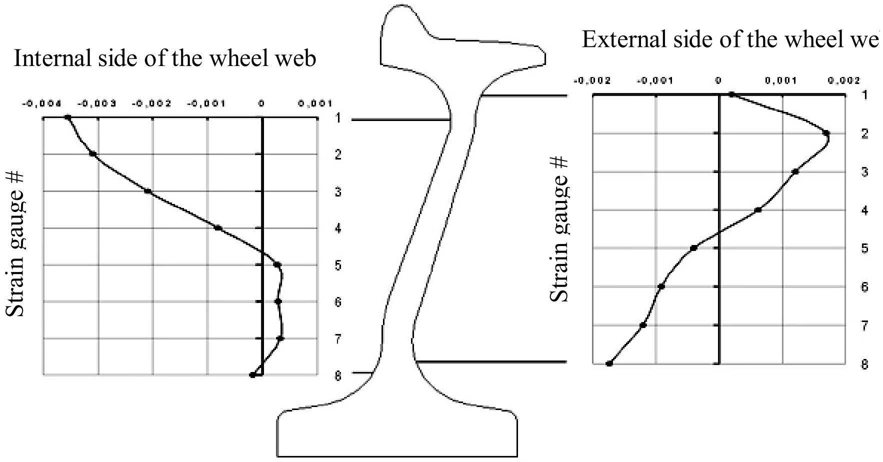


Fig. 3. The diagrams of the deformations at the internal and the external side of the web of a wheel

Table 1. Results of numerical analysis of a Russian wheel loaded with static force

Internal side of web of a wheel				
Strain gauge #	**Test**	**NASTRAN**	**ANSYS**	**COSMOS**
1	-0,0035457	-0,0048274	-0,0048251	-0,0048215
2	-0,0030958	-0,0027569	-0,0027543	-0,0027657
3	-0,0020882	-0,0016704	-0,0016692	-0,0016766
4	-0,00081086	-0,00094006	-0,00093562	-0,00094418
5	0,000272	-0,00041947	-0,00041157	-0,00041238
6	0,00028343	-2,9587E-06	-0,000012691	-0,000012735
7	0,0003219	0,00017478	0,00016852	0,0001693
8	-0,00017105	-0,00010708	-0,00010614	-0,00010578
External side of web of a wheel				
Strain gauge #	**Test**	**NASTRAN**	**ANSYS**	**COSMOS**
1	0,00019886	0,0006373	0,00065132	0,00064611
2	0,001704	0,0019544	0,001939	0,0019406
3	0,001208	0,0010549	0,0010513	0,0010547
4	0,00061924	0,00022312	0,00022503	0,00021953
5	-0,00040419	-0,00027328	-0,00027197	-0,00025934
6	-0,00091771	-0,00067952	-0,00068453	-0,00068513
7	-0,0012105	-0,0011688	-0,0011702	-0,0011695
8	-0,0017421	-0,00097488	-0,00097394	-0,00097337

a)

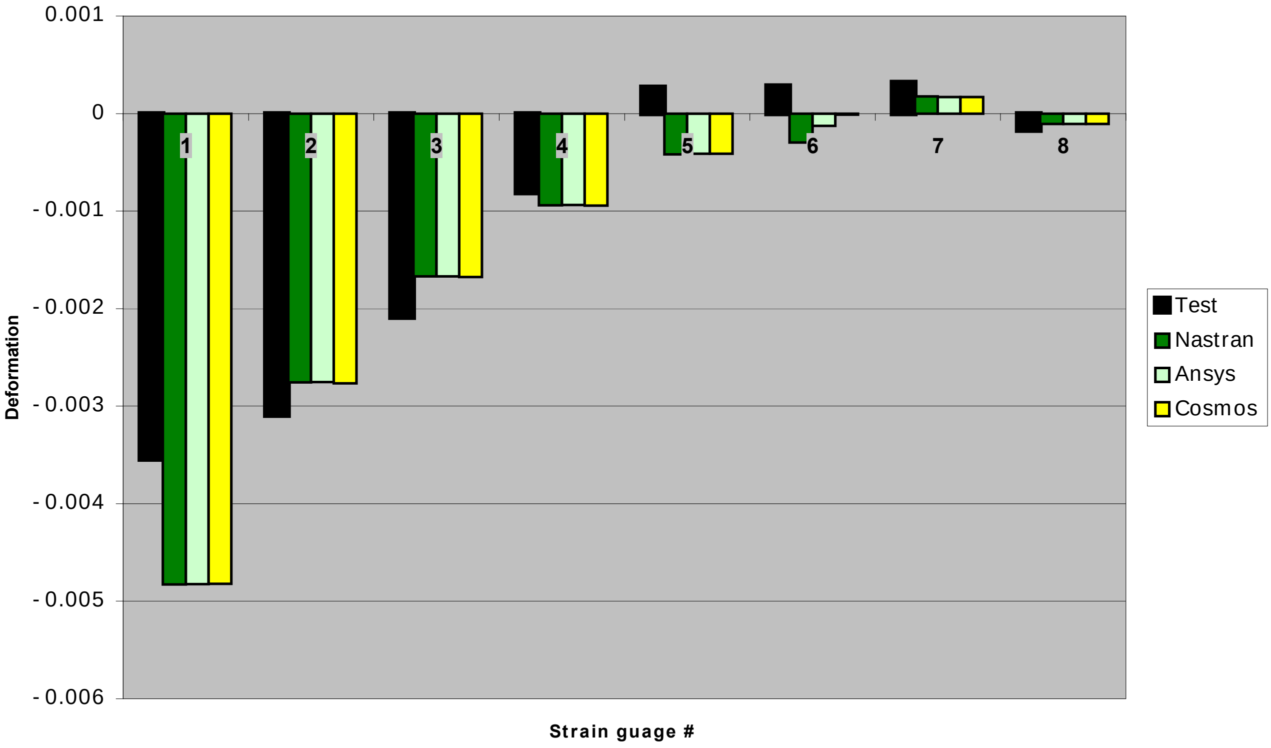


Fig. 4. Radial deformation – internal side of the wheel: a) bar chart, b) line chart

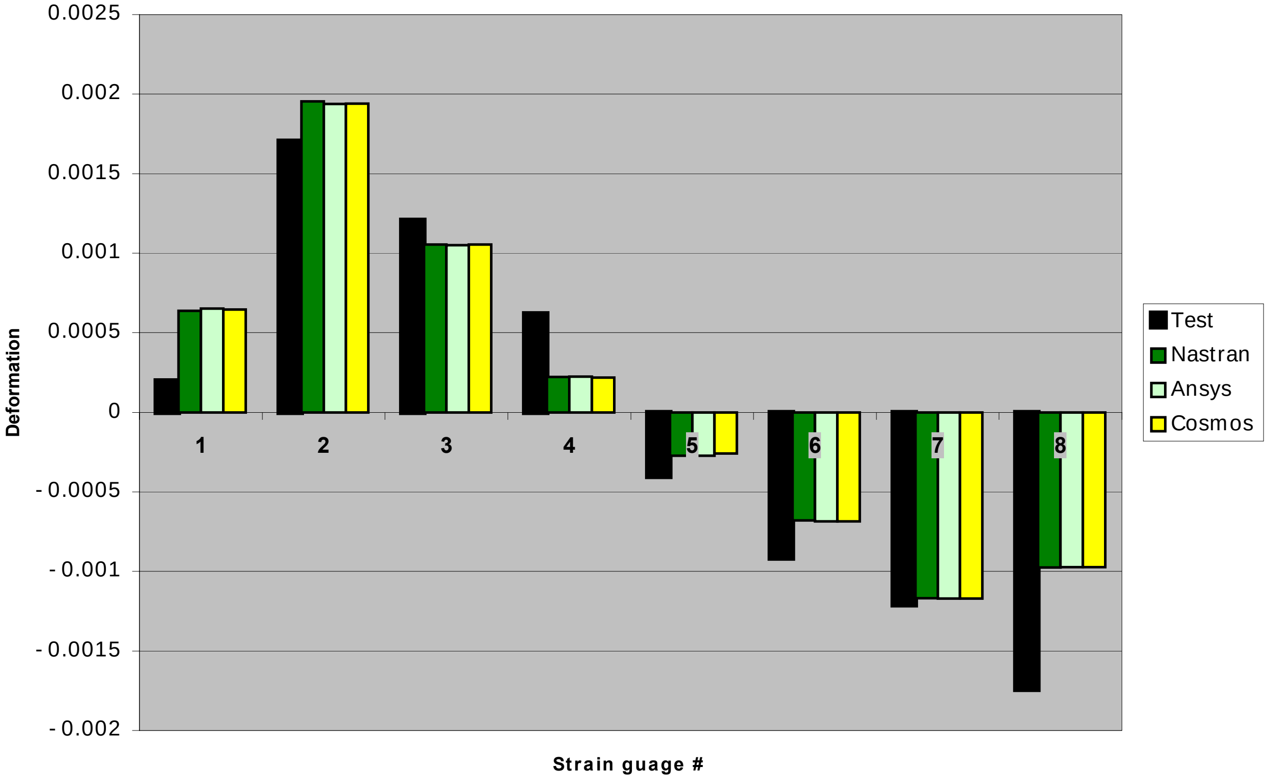


Fig. 5. Radial deformation – external side of the wheel: a) bar chart, b) line chart

3. STATIC EXPERIMENTAL INVESTIGATION OF A GERMAN WHEEL

The experimental tests have been conducted in the Bohumin Plant a.s. in Czech Republic. Test stand for strain measurements of railway wheels deformation has been used – Fig. 6.

Strain gauges have been placed on the internal and external surface of the web of a wheel in the axial and radial directions (Fig. 7). The maximum load generated by hydraulic actuator has corresponded to static load in the wheel-rail system. The strain gauges have been placed at the points of maximum stresses caused by static load due to carriage weight.

Strain gauges of KM120 type have been used, k=2,02. The conducted investigations have resulted in obtaining the deformation values of the web of a wheel of a railway wheel. The wheel has been loaded with axial and radial forces at the same time for 150 seconds (Fig. 7). The maximum steady-state force values have been: radial force 160 kN, axial force 60 kN (Fig. 8), which corresponds to the axle load of 32 T per axle. The maximum force values have also been adopted in the second stage of investigations, when FEM has been used. In order to compare the results of experimental tests with FEM numerical analysis results, the deformation values measured by strain gauges T3 and T5 have been selected. These gauges have been placed radially at both sides of the wheel and the deformation values measured by these gauges have been the highest of all.

Deformation values in the radial direction have been pre-dominant; they have been of a higher order than deformations in the axial direction. The maximum measured value of axial deformation of the web of a wheel at the internal side (T3 strain gauge) has been equal to 0,000412, and at the external side (T5 strain gauge) it has been equal to 0,000695 – Fig. 9

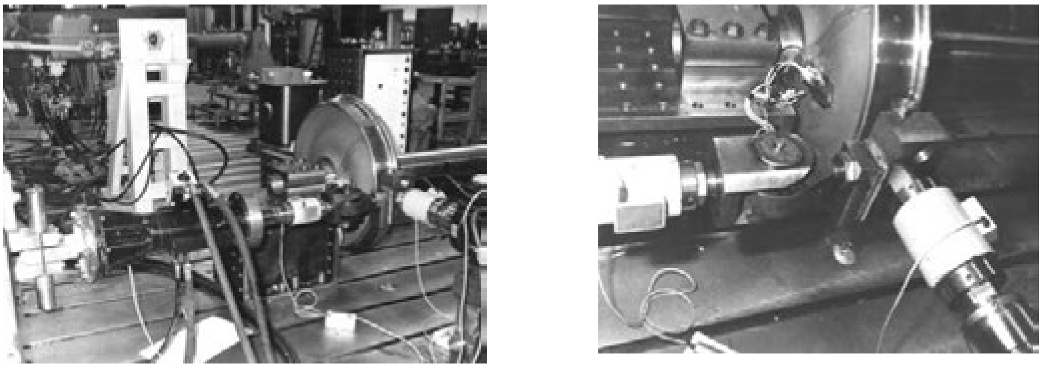


Fig. 6. Strain test stand for investigation of railway wheelsets' wheels deformations

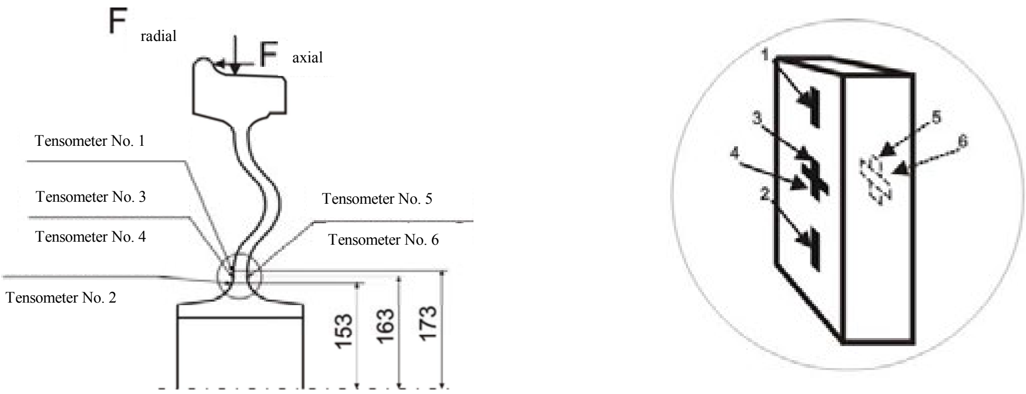


Fig. 7. Loading diagram for a wheel placed in the test stand; strain gauges placed on the wheel

F_{radial}	F_{axial}
-16,7	-9,3
-160,5	-30,2
-160,5	-34,9
-160,5	-39,7
-160,6	-44,8
-160,6	-49,8
-160,6	-55
-160,5	-60
-161,1	-54,7
-161,1	-49,8
-161,2	-44,9
-161,1	-39,8
-161,1	-34,9
-161,2	-30,1
-18,5	-10,1

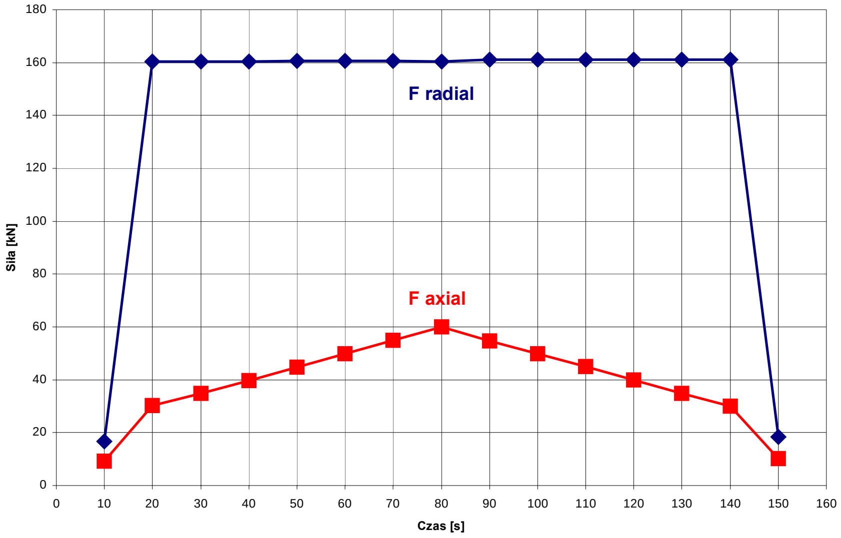


Fig. 8. Time cycles of forces loading the wheel at the test stand

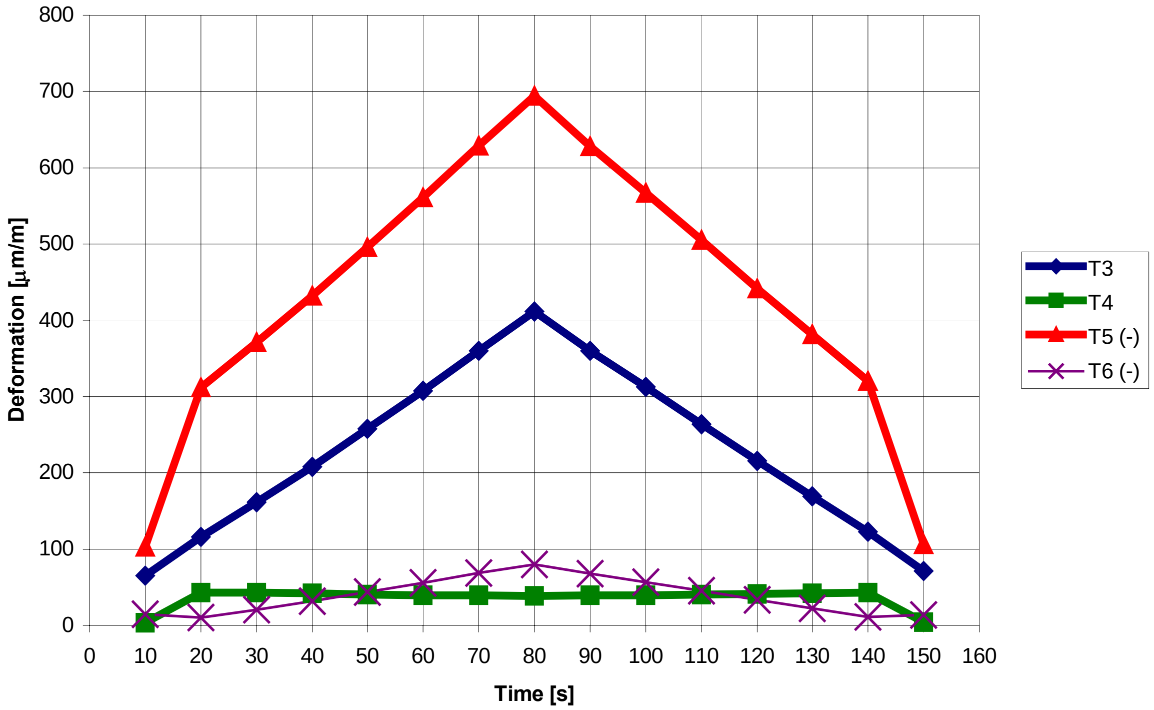


Fig. 9. Time cycles of wheelset wheel deformation measured at the test stand

After the FEM numerical analysis, the wheel deformation has been determined – Fig. 10b.

The lowest difference in the result as relates to experimental investigation has been generated in case of manual hexagonal mesh. Figure 10 shows comparison of the results.

The comparison confirms the correctness of the proposed model of loading the wheel with two forces – axial and radial.

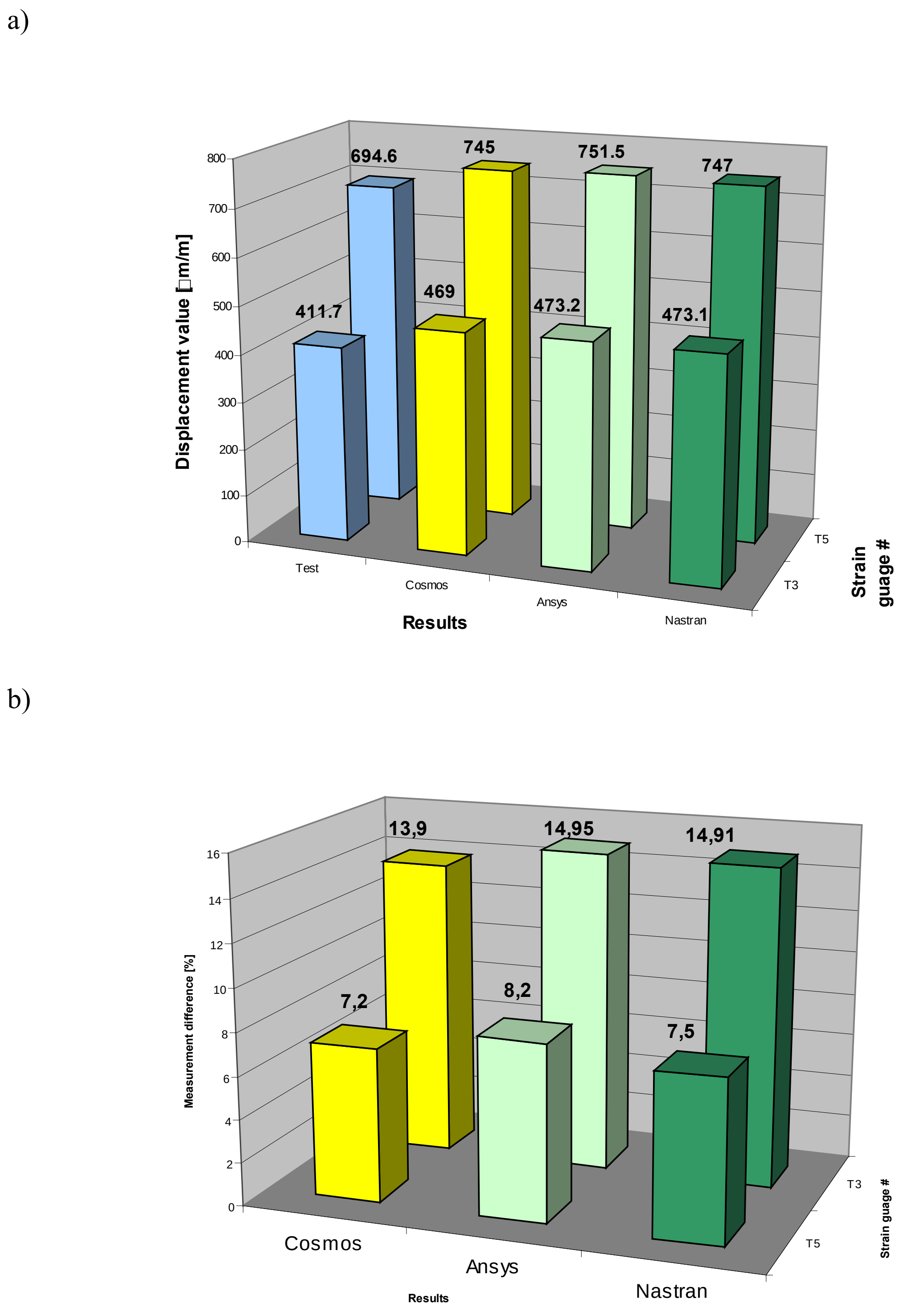


Fig. 10. Comparison of results of numerical analysis and experimental tests; load is due to carriage weight; a) deformation, b) results discrepancy

4. EXPERIMENTAL THERMAL INVESTIGATION OF A GERMAN WHEEL

A German wheel has been investigated (Technical Specification No. FWG302.0.02.001.007); its initial diameter has been equal to 920 mm. In order to simulate the effects of wear, this diameter has been machined down to 854 mm. The wheel has been subjected to braking in 13 cycles at the test stand of Machine Faculty of University of Zilina (Fig. 11). The investigation has been aimed at the temperature measurement in the railway wheel during braking.

Test stand scheme is shown in Fig. 12. The stand consists of a measurement frame affixed to the wheel shaft. Two brake sets are located on the frame. The braking force is determined by the air pressure measured with mechanical force gauge during calibration; this meter has been inserted between the wheel and brake casing. Temperature in the wheel of the web has been measured with the help of thermosensors of K type, manufactured by American company Omega. The sensors and ceramic casing have been symmetrically affixed at every 90º along the circumference, and 9 mm away from the wheel's rolling surface – Fig.13.

The thermosensor signal is amplified and transferred to data acquisition computer card.

The tests have been run in cycles in accordance with UIC 510-5:

- braking at 60 km/h, braking power of 60 kW, time interval 45 minutes – one cycle
- braking at 60 km/h, braking power of 30 kW, time interval 45 minutes – one cycle
- braking at 60 km/h, braking power of 40 kW, time interval 45 minutes – one cycle
- braking at 60 km/h, braking power of 60 kW, time interval 45 minutes – ten cycles.

013-P10-type shoe brakes have been used in the investigations. These are widely used in Czech rail transport. They are made of cast iron with 1 per cent phosphorus admixture. The shoes have been placed 10 mm away from the external face of wheel's rolling surface (320 x 80 mm).

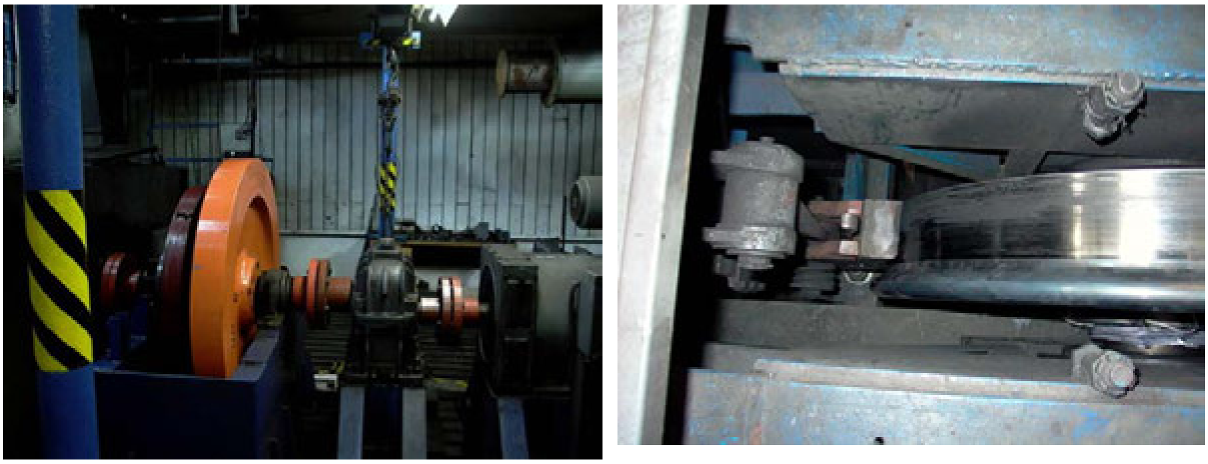


Fig. 11. Test stand for deformation and temperature investigation of railway wheelsets wheels

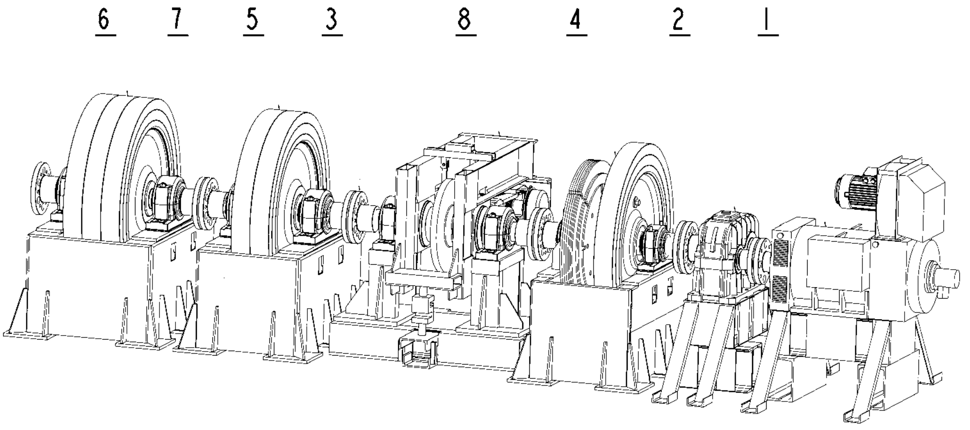


Fig. 12. Test stand for investigation of deformation and temperature of railway wheelsets wheels:

1. Electric motor (P = 265 kW, P_{max} = 400 kW, n = 3200 rpm)
2. Toothed gear (i = 1,5 , resp. i = 1,72 or i = 4)
3. Coupling
4. Flywheel 400 kgm^2 = 280 + 120 kgm^2 (2x5+3x10+4x15)
5. Flywheel 600 kgm^2
6. Flywheel 900 kgm^2
7. Stand casing
8. Wheel with railway shoe brake

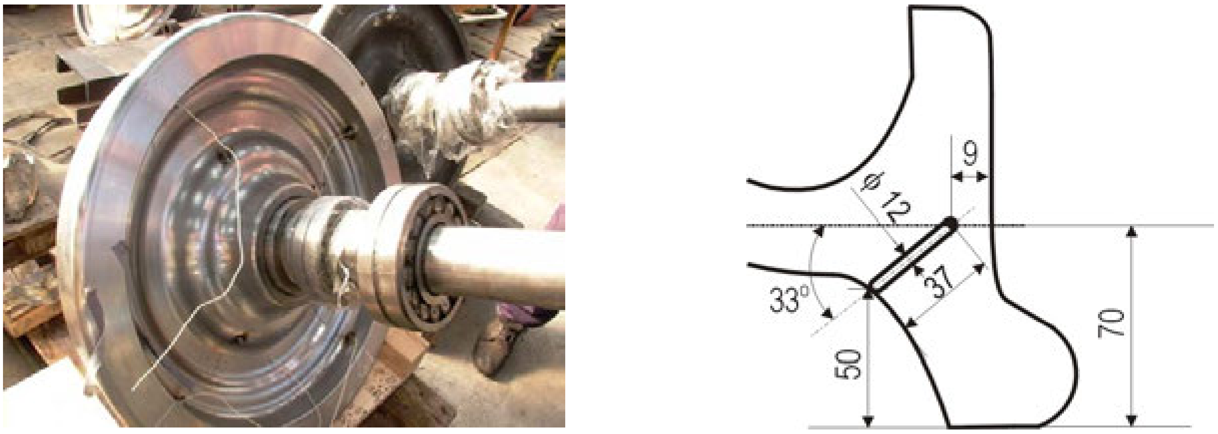


Fig. 13. Placing of thermocouples along the wheel circumference (9 mm away from the rolling surface)

During braking the wheel has been cooled by the air flowing at 30 km/h (half the run speed) . The cooling after braking has been done with forced air ventilation, air has been heated to 250 ºC. Lower temperatures have been obtained by using water sprinkling both sides of the wheel hub at 6 km/h. The water has been able to flow freely over the web and wheel's rolling surface. Total amount of water used has been equal to 165 l/h.

The initial wheel temperature in each cycle has been equal to max. 50º C. Investigations have resulted in a reading of average temperature during the whole cycle (thermocouples measurement). In order to verify the correctness of the used calculation algorithm for temperature field a comparison of numerical calculations with results of experimental tests has been conducted (braking cycle in accordance with UIC 510-5).

The numerical thermal analysis has been conducted according to UIC Report [4], for a wheel with the maximum wear of the rolling surface. The thermal flux has been determined for each braking phase. Between the braking processes the wheel has been left without any incoming thermal flux. The convection simulating cooling processes lasting 14400 seconds has been taken into account. After each braking, the temperatures in the points, where thermocouples have been placed in real life, have been calculated. As a result, the average values of the temperatures for selected nodes have been determined for several braking cycles - Fig. 14) as well as temperature fields of the wheel.

The comparison of experimental investigation and numerical calculation of braking cycles leads to following conclusions:

- the smallest difference of temperatures has been obtained with the help of COSMOS software (up to 5 %);
- for NASTRAN and ANSYS software the discrepancy has been as high as 22 %.

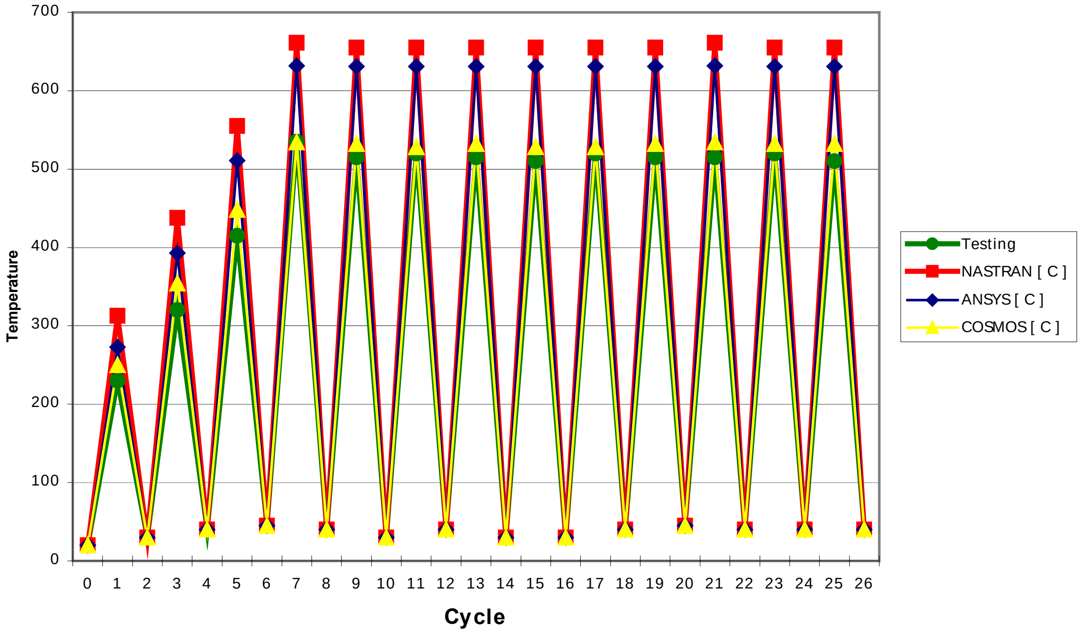


Fig. 14. Temperature diagram for nodes corresponding to thermocouples placing

5. REMARKS AND CONCLUSIONS

On the basis of web of a wheel model a rational mathematical numerical model has been established (rationality here is understood as reasonable accuracy and computation time). This model has made possible the comparison of numerical calculations with unshakeable analytical results.

The wheel model, which has been created, has made possible the subsequent creation of rational discrete models of selected railway wheels designs. These have been analysed with respect to loads arising during manufacture and operation.

In addition, simulation of railway wheel strain for different mesh types and patterns generated by different software has been conducted in order to establish possible results discrepancy. The analysis has shown the importance of accurate modelling of the load course and wheel mesh.

In order to check out the calculation model the experimental tests of static loads (due to carriage weight) and thermal loads (due to braking) have been run. The results of these tests have been compared with results of numerical analysis. The proposed calculation model has been compared with results of test stand investigation. Comparison of the results has confirmed the correctness of the proposed discrete model and calculation algorithm.

Additionally, the influence of wheel geometry (rolling surface diameter, hub geometry, web geometry) and technological parameters (difference of interferences between wheel and axle) on the distribution of displacements and stresses has been investigated.

The following principal conclusions have been drawn as a result of the investigation:

1. Using Finite Element Method (FEM) the computation procedures for temperature, displacement and stress fields in railway wheelsets have been elaborated. This method takes into account real wheel geometry, complex loads of the wheelset and material properties.
2. New procedures for solving the wheel-axle contact problems have been worked out.
3. The numerical computation results have been compared to the analytical computation results. Results of this comparison justified selection of FEM mesh.
4. The numerical computation results of the wheelset designs have been compared with the results of experimental investigations. This should verify the adequacy of algorithm method used. Until now, the discrepancies between numerical computation and field investigation results could be as high as 50 per cent. For the method proposed in this paper the difference was not higher than 22 per cent in case of thermal loads and 5 per cent for static loads.
5. The elaborated procedures of design and selection of wheelsets' wheels using FEM have made possible a complex strength analysis of ten different wheel designs; conclusions as to their design and service have been worked out.
6. The conducted numerical analysis of the influence of the most significant geometrical and service parameters of wheels of railway wheelsets should enable the designers to optimise and select the wheels according to their process engineering parameters and construction.

Further investigation will be directed at more precise definition of boundary condition for thermal loads in order to achieve less discrepancy in the results; real calculations of dynamic loads are also planned.

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