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Numerical - experimntal strenght analysis of wheels of railway wheelsets

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Summary: 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 is absent; there is no possibility of optimising wheelsets' construction characteristics depending on manufacturing process and service parameters. The analysis of this problem has made possible: comparison of software used so far in strain calculation of railway wheelsets; elaboration of design methodology for railway wheelsets; increase in durability of railway wheelsets; increase in the safety of rail transport; decrease in manufacturing and service costs of railway wheelsets.

Index Terms: finite elements method, numerical analysis, wheelsets.

1. NOTATION AND UNITS

c	coefficient
c_1	coefficient
D	coefficient
E	Young's modulus
h	mesh thickness
$M(r)$	bending moment
P	force
r_1	inner radius of mesh a wheel
r_2	outer radius of mesh a wheel
T	pressing force
T_r	coefficient
$W(r)$	displacements distribution
$\sigma_{r\,max}$	maximum strain in radial direction
μ	Lame coefficient
Δ_y	pre-set displacement in y direction
Δ_z	pre-set displacement in z direction
κ	Lame coefficient
v	Poisson's ratio
$\tau_{r\,\phi\,max}$	maximum strain in circumferential direction
ς	torsion of inner contour
$\Delta\vartheta$	torsion length in circumferential direction
σ_r	strain in radial direction
η	coefficient
μ	friction coefficient

2. INTRODUCTION

In spite of growing competitiveness of the road transport, the rail transport is still the principal long-distance transport system. In some countries (France, UK, Germany) the rail transport experiences its renaissance in city agglomerations (trams, rail bus, metro) and in long-distance travel (high-speed trains). Increased service demands as well as environmental and traffic safety requirements set on rail vehicles in the majority of European states explain the necessity of manufacturing rolling stock fulfilling high quality standards. The modern trains should be faster, cheaper and safer. The travel conditions, i.e. passengers' comfort, should also be improved. All these factors depend most of all on the design of rail vehicle.

The demands set on fast modern rail vehicles and their parts, including the railway wheelsets, can be enumerated as follows [1,2]: decreasing weight; increasing vehicle elements mechanical strength; lessening the noise and vibrations level; increasing travel comfort; decreasing dynamical interaction between vehicle and the track.

The development of computers and software provides possibilities of modelling new designs of different structural elements of rail vehicles. The phenomena occurring during the vehicle service can be adequately described, The error generated during design or prototype tests can lead to tragic occurrences during the service itself. These may be related to additional financial outlay and lengthening the prototype testing time, but they may also cause incalculable loss of human health or life.

According to the experts, the rail transport is one of the most ecological (Table 1) and safe (Table 2) eans of transport. Still, accidents happen [3]. They are not very frequent, but they are highlighted by the media. The most famous accident to date happened in Eschede, Germany, on 3rd June, 1998. Incorrectly designed railway wheelset has caused a tragic derailment and death of 101 persons (Fig.1 [4]). Similar, though less tragic accidents happened lately all over Europe [3]: Rickerscote, 8 March 1996, Sandy, 17 June 1998, Hatfield, 17 October 2000, Potters Bar, 10 May 2002.

Table 1. Comparison of different means of transport

Transport means	Energy consumption in kcal/passenger-km	Carbon dioxide emission in g of carbon /passenger-km
Car	642	45
Bus	196	19
Train	106	5
Airplane	393	30

Table 2. Mortality of the passengers per a hundred million

Transport means	Kilometres	Travels	Hours
Car	0,31	4	13
Bus	0,06	0,5	1
Train	0,08	2	4
Airplane	0,009	20	5

Fig. 1. Derailment of train in Germany - Eschede, 3 June 1998.

The advancement in the computers computational speed and the elaboration of complex software based on finite element method (FEM) and devoted to the railway industry demands, results in running calculations and simulations, which have not been previously possible.
There are many issues, which so far have been only experimentally/analytically investigated. The use of numerical methods limits or wholly eliminates the need for some tests or calculations.
The investigation time is therefore decreased, the complex test stands can be replaced with suitable software; hence, financial advantages are gained.
On the basis of the above analysis related to state-of-the-art in the railway wheelsets subject matter, conclusion may be drawn that computation by numerical methods might play significant role in each of the above issues.
It is certain that numerical calculations should be the starting point in the design, assembly and service stress analysis and thermal analysis of the material.
However, the numerical methods are saddled with errors due to the imperfect transformation of the real model into the virtual model. Still, if the investigation methods are used jointly, i.e. experimental tests are backed by numerical analysis, the results obtained may be close to reality.
Basing on the references, the present state-of-the-art of the numerical calculations of the railway wheelset wheels can be summarised as follows:

- lack of universally accepted computational algorithm of railway wheels,
- inadequate experimental confirmation of the correctness of computational procedures used at present,
- absence of comparison of numerical calculations of railway wheels done with the help of different software,
- discrepancies in set boundary conditions,
- absence of precise algorithm for creating FEM model for railway wheels (type, distribution and size of element)
- complexity of stress calculation method, when the stress is due to assembly-time interference,
- lack of UIC certification for calculations of thermal stress due to braking.

3. NUMERICAL ANALYSIS OF THE WHEELSETS' WHEELS

The railway wheelset is a constructional element influencing directly the vehicle motion as well as passengers' safety. That is why the axle, wheel and the wheelset itself must be characterised by adequate mechanical strength during the service period. Nowadays the railway wheelsets designs used are the effect of service experience and process engineering. Lately, the numerical analyses of stress and strain done in the wheelset design phase are also utilised.

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. Three software packages are used – ANSYS, NASTRAN and COSMOS. This diversity of software should make possible the comparison of convergence of results (tolerance, computation time), provided that input data are identical (e.g. wheel geometry, material. boundary conditions, FEM mesh). Before the calculations are started, these data must be established, if the results are to be comparable.
The justification for undertaking this issue is that the methodology of design of railway wheelsets both in Poland and abroad is absent; there is no possibility of optimising wheelsets' construction characteristics depending on manufacturing process and service parameters. The analysis of this problem has made possible:

- comparison of software used so far in strain calculation of railway wheelsets;
- elaboration of design methodology for railway wheelsets;
- increase in durability of railway wheelsets;
- increase in the safety of rail transport;
- decrease in manufacturing and service costs of railway wheelsets.

3.1. Initial investigations – mesh of a wheel model

In order to make FEM calculations more accurate, the comparison of analytical and numerical methods of calculations has been conducted on the basis of mesh of a wheel model. Figure 2 shows the mesh of a wheel model and physical and discrete models generated by the software.

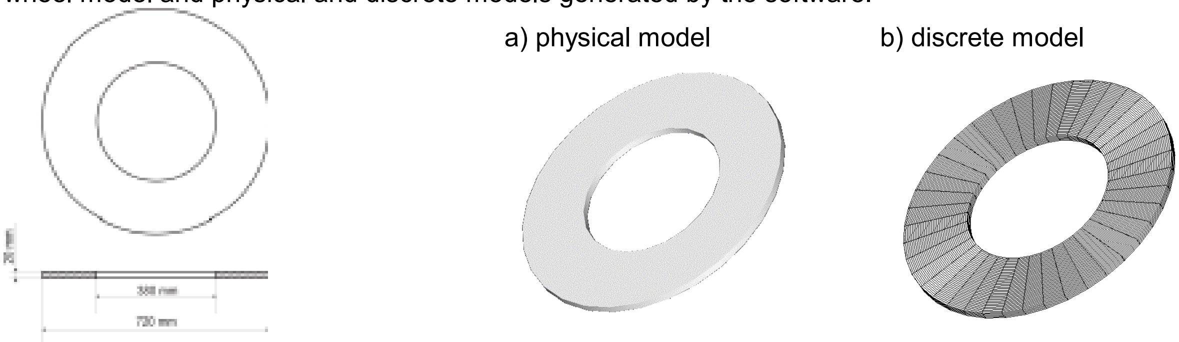


Fig. 2. Model of the mesh of a wheel

Three possible loads have been considered – see Fig.3. In case of the first load, the initial displacements of the nodes of inner surface generator parallel to y-axis of the global Cartesian co-ordinate system have been defined (Fig. 3.a). In the case of second load, the initial displacement of the nodes has been alongside T-angle of polar co-ordinate system – Fig. 3.b. In the case of third load, the nodes have been linearly displaced parallel to the z-axis of Cartesian co-ordinate system – Fig.3.c. The maximum strains for the presented loads have been calculated analytically. The data: mesh thickness h = 0,02 [m], inner radius outline r_1 = 0,019 [m] and outer radius r_2= 0,36 [mm] [5]:

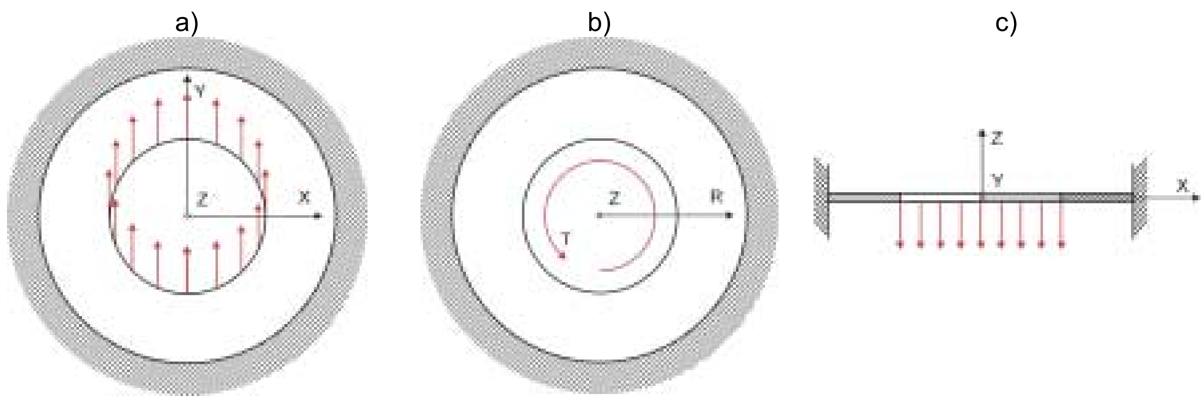


Fig. 3. Load models for the mesh a wheel

Case # 1.

$$\sigma_{r\max} = \frac{\mu \cdot \Delta_y}{c}\left[\frac{3\cdot\kappa\left(r_1^2 + r_2^2\right)}{r_1} + 2r_1 + \frac{\kappa^2\left(r_1^2 + r_2^2\right)}{r_1} - 2\frac{\kappa r_2^2}{r_1}\right] \quad (1)$$

where μ and κ are Lame coefficients, calculated from the formulas:

$$\mu = \frac{E}{2(1+\nu)}$$
$$\kappa = \frac{3-\nu}{1+\nu} \quad (2)$$

E – Young's modulus
ν –Poisson's constant
Δy – pre-set displacement = 0,0001 [m]
Coefficient c has been determined with the help of formula:

$$c = r_1^2 - r_2^2 - \kappa^2 (r_1^2 + r_2^2) \ln \frac{r_1}{r_2} \quad (3)$$

For this equation

$$\sigma_{r\max} = 143{,}9 \text{ MPa} \quad (4)$$

Case # 2

$$\tau_{r\phi\max} = \frac{2 \cdot \mu \cdot r_1^2 \cdot \varsigma}{r_2^2 - r_1^2} \quad (5)$$

where ς - torsion of inner contour, torsion angle is

$$\varsigma = \frac{\Delta\vartheta}{r_1} \quad (6)$$

for $\Delta\vartheta = 0{,}0001 \ \ [\mathrm{m}]$

$$\tau_{r\phi\max} = 110{,}5 \text{ MPa} \quad (7)$$

Case # 3

$$\sigma_r = 6\frac{M(r)}{h^2} \quad (8)$$

where bending moment at mesh cross-section, derived from formula:

$$M(r) = D\left[\frac{d^2}{dr^2} w(r) + \frac{\nu}{r}\frac{d}{dr} w(r)\right] \quad (9)$$

depends on the displacements distribution of the mesh

$$w(r) = \frac{T_r r_2}{8\pi D(\eta^2 - 1)}\left\{\left[\ln(\eta) - 1 + \frac{\eta^2 + 1}{2}\right]\left(1 - \frac{r_1^2}{r_2^2}\right) - \left[2\ln(\eta) + (\eta^2 - 1)\frac{r_1^2}{r_2^2}\right]\ln\left(\frac{r_2}{r_1}\right)\right\} \quad (10)$$

Bending rigidity and axial forces have been calculated from the formulas:

$$D = \frac{Eh^3}{12(1-\nu^2)}$$

$$T_r = \frac{\Delta z}{c_1}$$

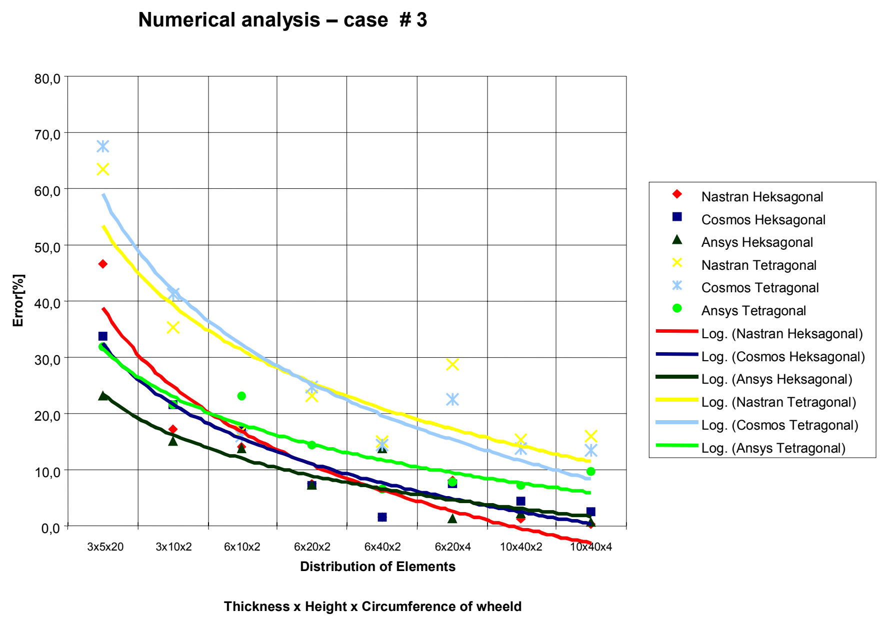
$$c_1 = \frac{\left[\ln(\eta) - 1 + \frac{\eta^2 + 1}{2}\right]\left(1 - \frac{r_1^2}{r_2^2}\right) - \left[2\ln(\eta) + (\eta^2 - 1)\frac{r_1^2}{r_2^2}\right]\ln\left(\frac{r_2}{r_1}\right)}{8\pi D(\eta^2 - 1)} r_2 \quad (11)$$

$$\eta = \frac{r_2}{r_1}$$

$$\sigma_{r\max} = 60{,}5 \text{ MPa} \quad (12)$$

At the same time, meshes have been generated, differing one from the other by the size and distribution of elements as well as the type of element used.
The investigation has resulted in determination of the percentage error of numerical analysis, which depends on the size and distribution of the elements and on type of element used.
Figure 4a illustrates graphically the computation error for the third and most typical load; computation time is shown in Fig.4b.
The analysis shows that the minimum number of elements along the lateral cross-section of the wheel mesh should be equal to six.
These investigations have led to generating an optimum mesh for the mesh with respect to computation accuracy and time.

a)



b)

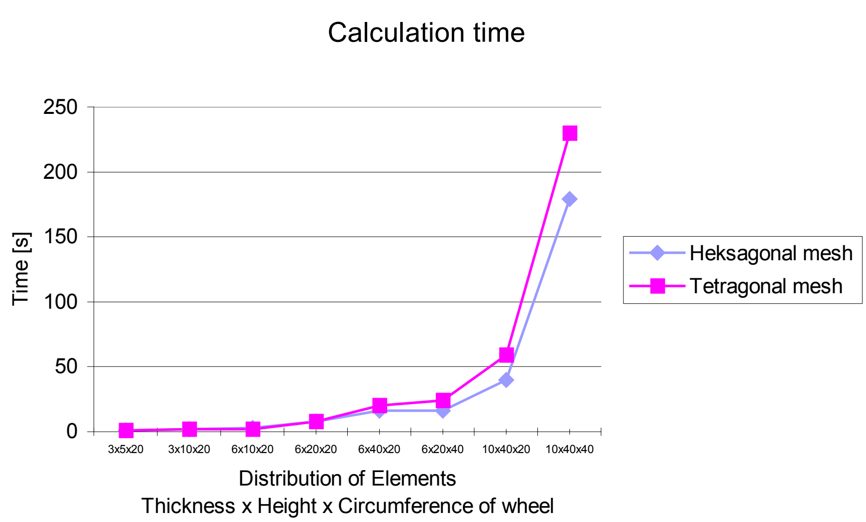


Fig.4. Results of comparison of analytical and numerical calculations - case #3

3.2. Subject of investigation – wheel design of railway wheelset

The work was principally aimed at elaborating methods of selection of constructional parameters for the railway wheelset wheels. The conducted calculations of railway wheelsets strains should improve the accuracy of presently used numerical methods and should also be helpful in working out the guidelines for designers, since the calculation algorithm and FEM software are determined.
The numerical investigation has been run for ten different geometrical designs of wheels Ø 920 and Ø 920 – h (worn out), differing by geometrical parameters of the mesh, hub and wheel seat diameter. The generated FE mesh for these wheels has been shown in Fig. 5.

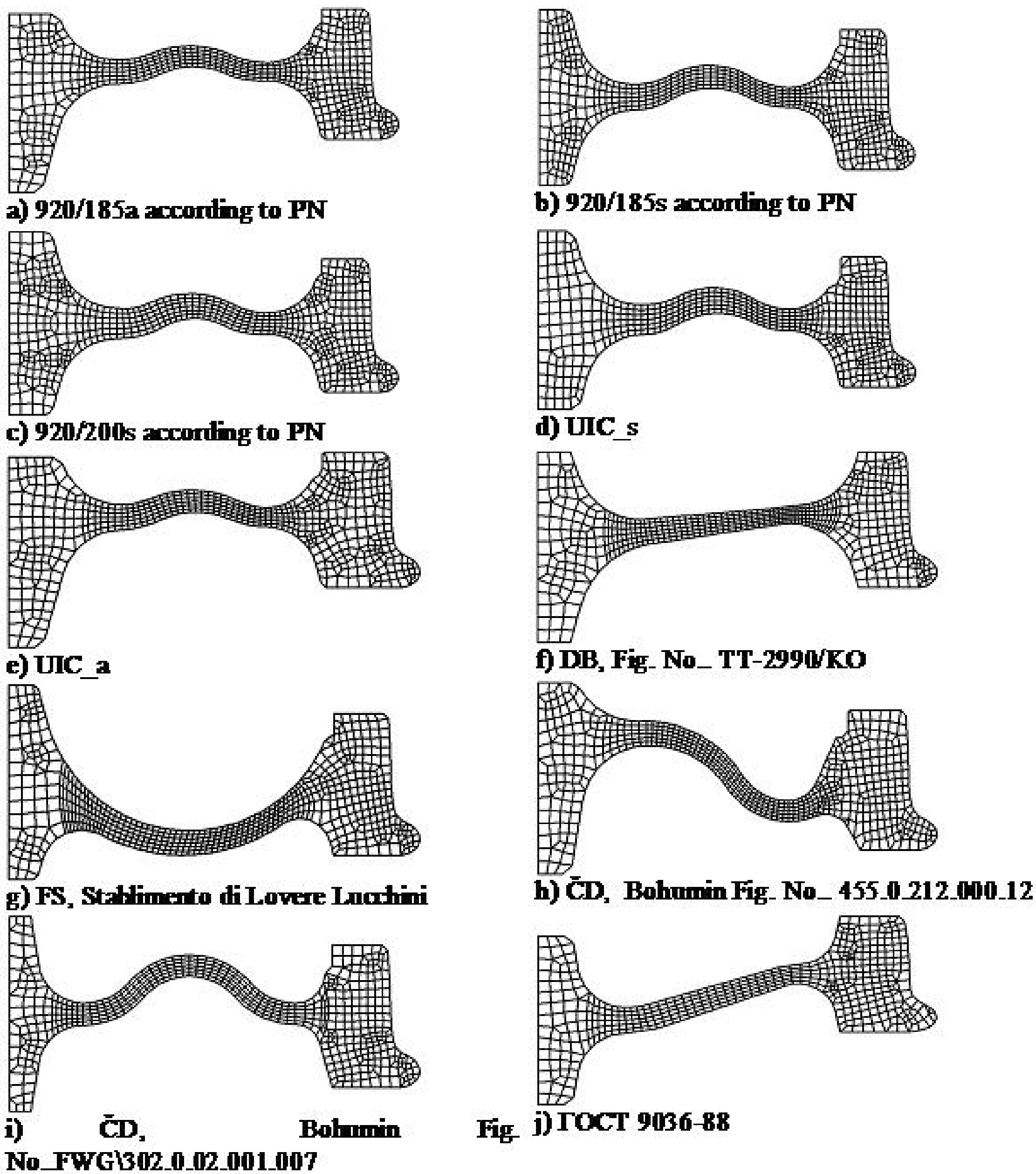


Fig. 5. Discrete models of the investigated monoblock wheels

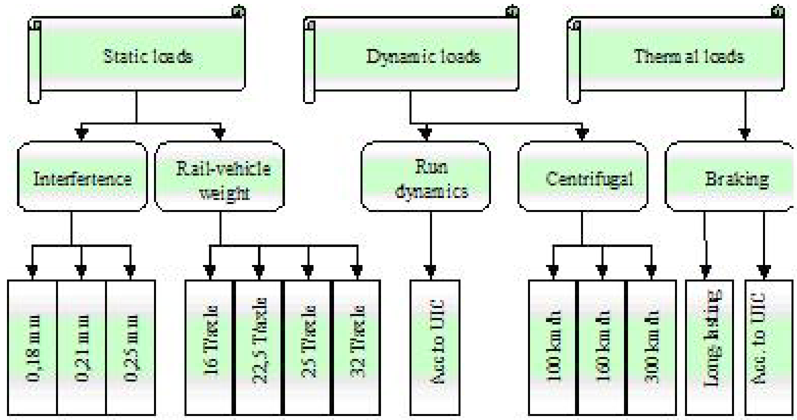


Fig. 6. Wheel loads of the railway wheelset

Next, assembly loads and service loads of the railway wheelset wheel have been determined. These have been used in subsequent research – Fig.6. These are static, dynamic and thermal loads. The static loads are due to interferential fit during the assembly and to carriage weight. The dynamic loads are related to the vehicle run over the track and centrifugal forces resulting from run speed. The thermal loads occur in the wheel during braking. The numerical analysis has been conducted for P52 material (R7 according to UIC classification), which is used for railway wheels in Poland. 3.3. Static loads of the railway wheels.

3.3. Remarks and conclusions numerical analysis of the wheelsets' wheels

After the numerical analysis of ten different wheel designs has been conducted with the help of three FEM software packages, the obtained results of reduced stress have been compared with respect to: mesh type and pattern, value-dependent stress distribution, wheel construction, degree of wear, used software.

The following conclusions may be drawn:

- mesh type and pattern have great impact on the temperature , displacement and stress fields values;
- geometry of the wheel mesh and hub bear influence on the distribution and values of the stresses. If the service conditions are known, the appropriate wheel geometry can be selected.
- analysis of the degree of wear of the wheel on the distribution and values of the stresses has shown that the influence is significant only in case of loads due to braking;
- comparison of software used has shown significant differences in computation of temperature field caused by braking;
- the conducted analysis has clearly demonstrated the importance of a correct FEM model
- numerical analysis has made possible comparison o software used in calculations.

The conducted analysis has shown the supremacy of modern-design wheels with respect to the mechanical strength (Italian wheel. Czech wheels 455.0.212 and302.0.02.001.007).

4. EKSPERIMENTAL INVESTIGATION OF DEFORMATION AND TEMPERATURE FIELDS OF RAILWAY WHEELS

4.1. 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. 7.

Strain gauges have been placed on the internal and external surface of the web of a wheel in the axial and radial directions (Fig. 8).

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. 8). The maximum steady-state force values have been: radial force 160 kN, axial force 60 kN (Fig. 9), 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. 10.

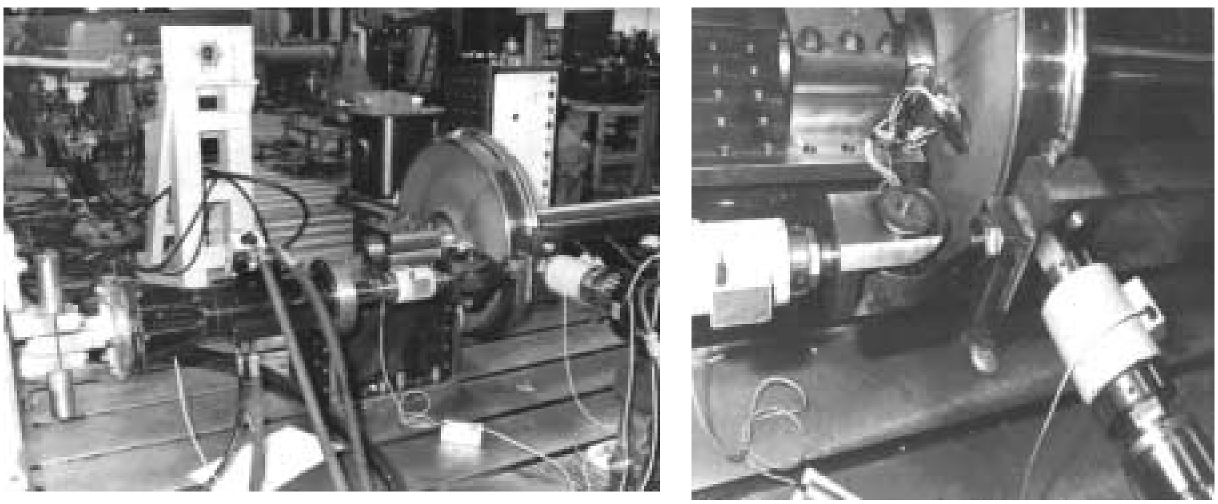


Fig. 7. Strain test stand for investigation of railway wheelsets' wheels deformations

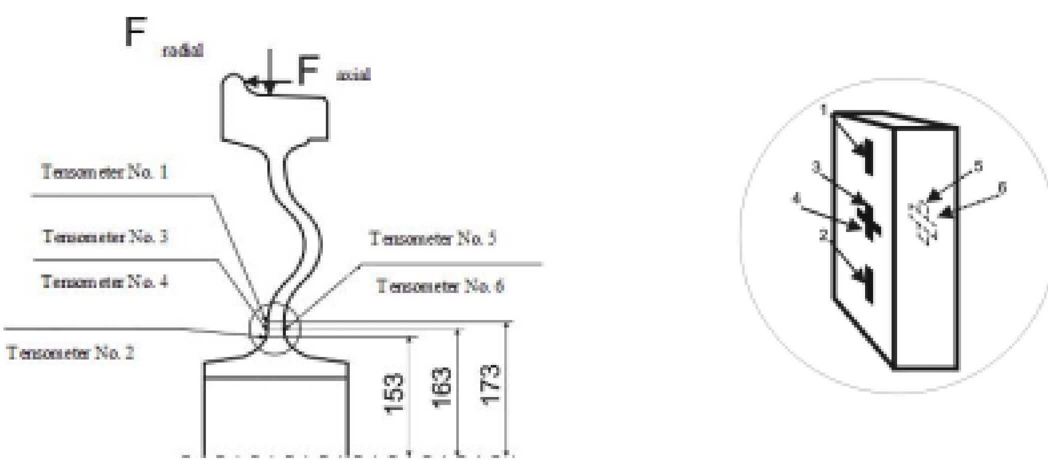


Fig. 8. Loading diagram for a wheel placed in the test stand; strain gauges placed on the wheel

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

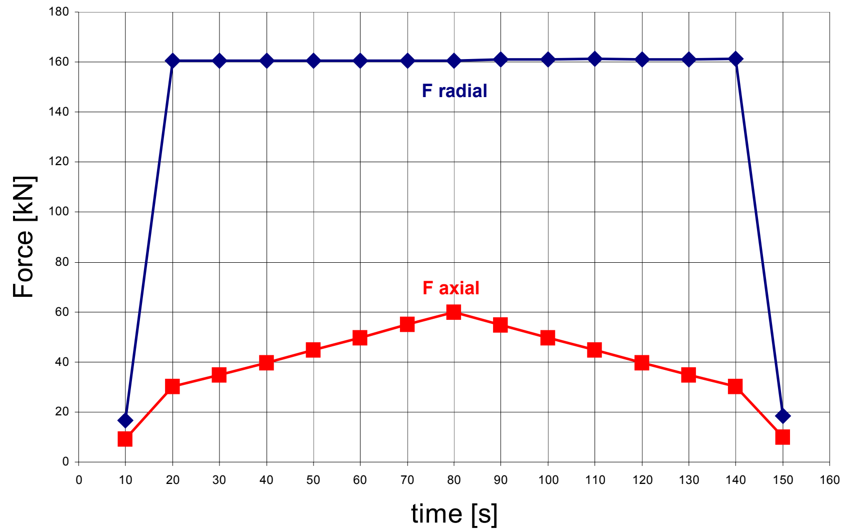


Fig. 9. Time cycles of forces loading the wheel at the test stand

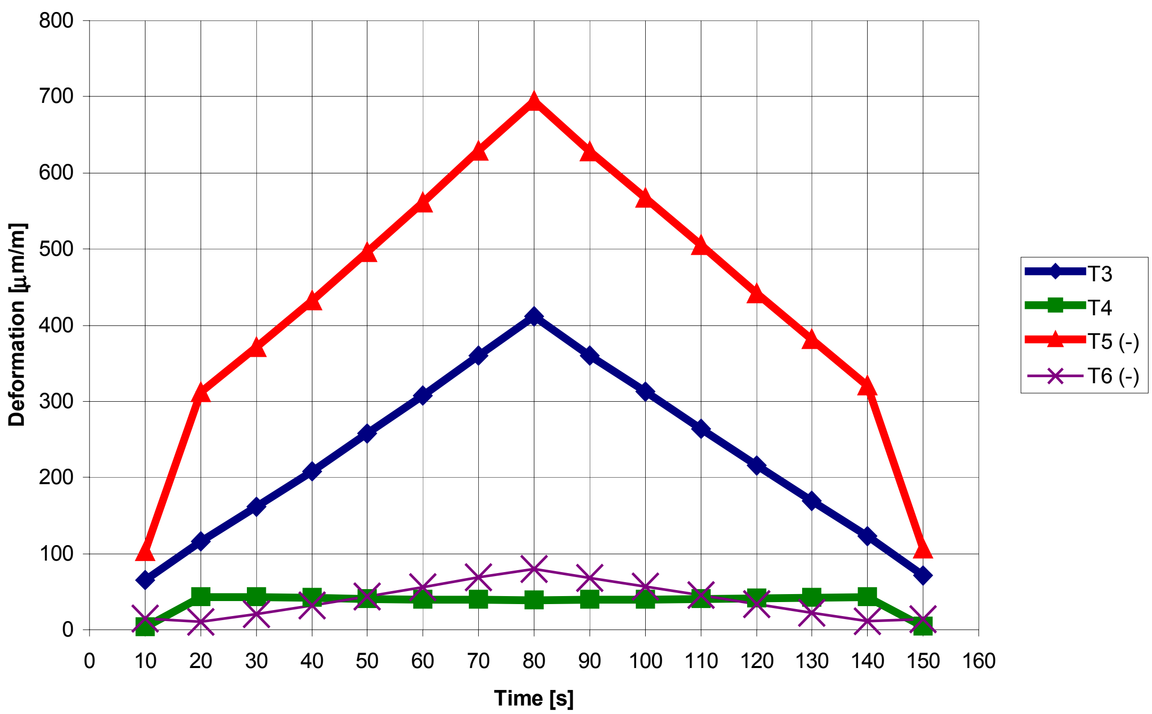
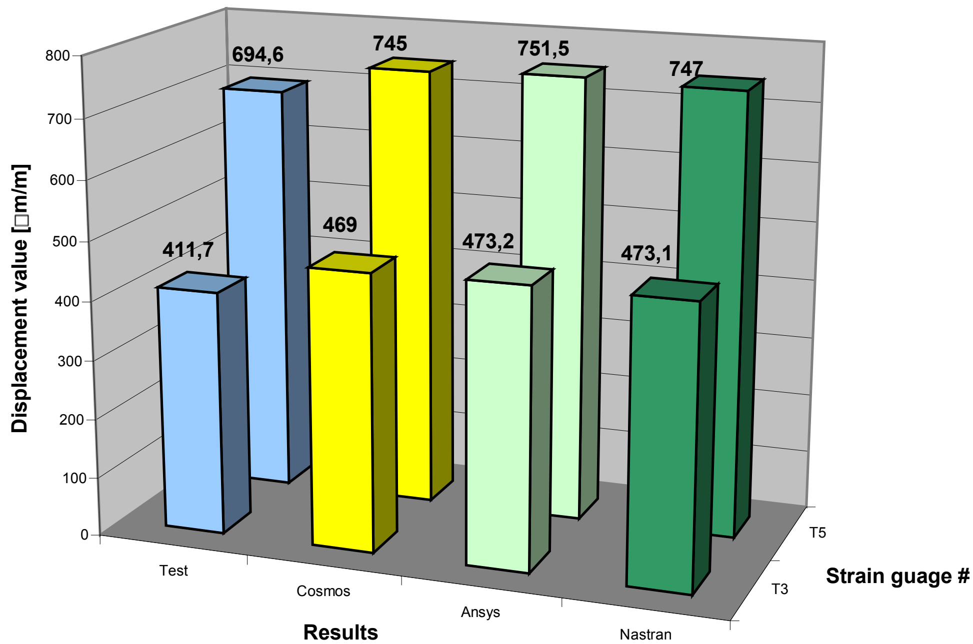


Fig.10. Time cycles of wheelset wheel deformation measured at the test stand

After the FEM numerical analysis, the wheel deformation has been determined – Fig. 11b.
The lowest difference in the result as relates to experimental investigation has been generated in case of manual hexagonal mesh. Figure 11 shows comparison of the results.
The comparison confirms the correctness of the proposed model of loading the wheel with two forces – axial and radial.

a)



b)

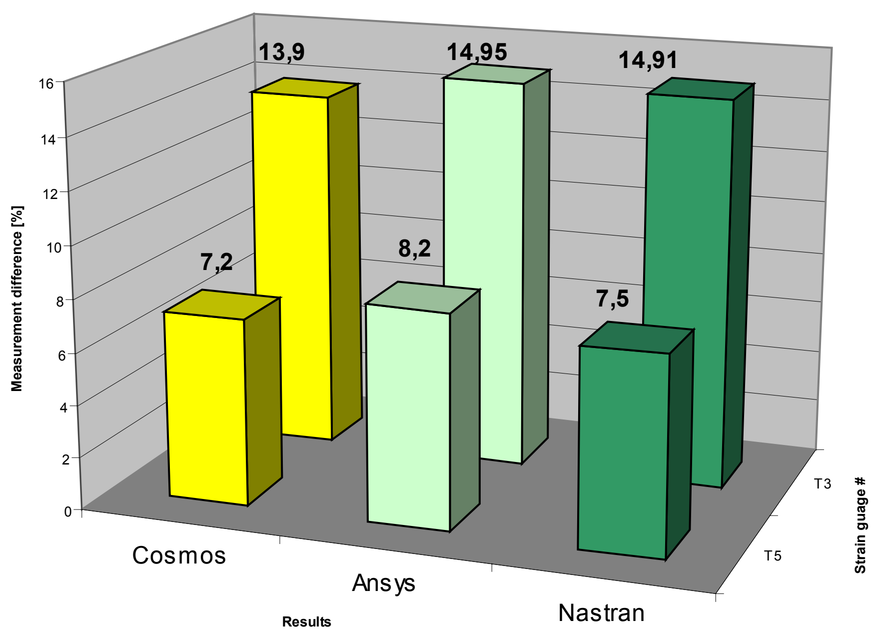


Fig. 11. Comparison of results of numerical analysis and experimental tests; load is due to carriage weight; a) deformation, b) results discrepancy

4.2. 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.12). The investigation has been aimed at the temperature measurement in the railway wheel during braking.

Test stand scheme is shown in Fig.13. 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.14.

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

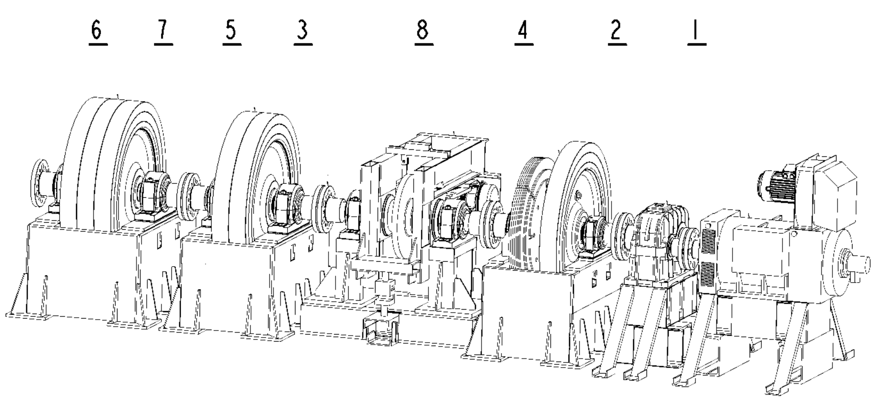


Fig. 13. Test stand for investigation of deformation and temperature of railway wheelsets wheels:

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

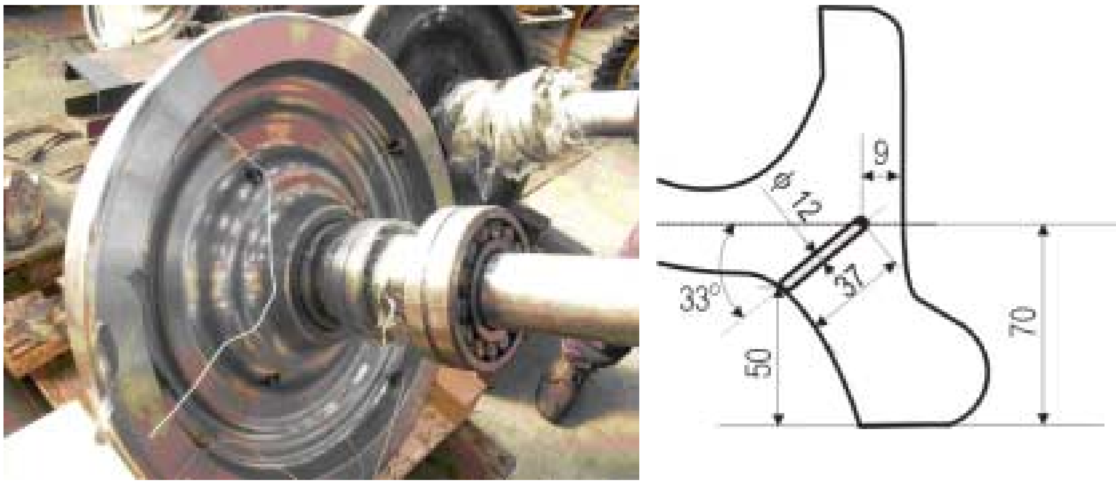


Fig. 14. Placing of thermocouples along the wheel circumference (9 mm away from the rolling surface)

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 [6,7], 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. 15) as well as temperature fields of the wheel.
The comparison of experimental investigation and numerical calculation of braking cycles leads to following conclusions:

- for NASTRAN and ANSYS software the discrepancy has been as 22 %.

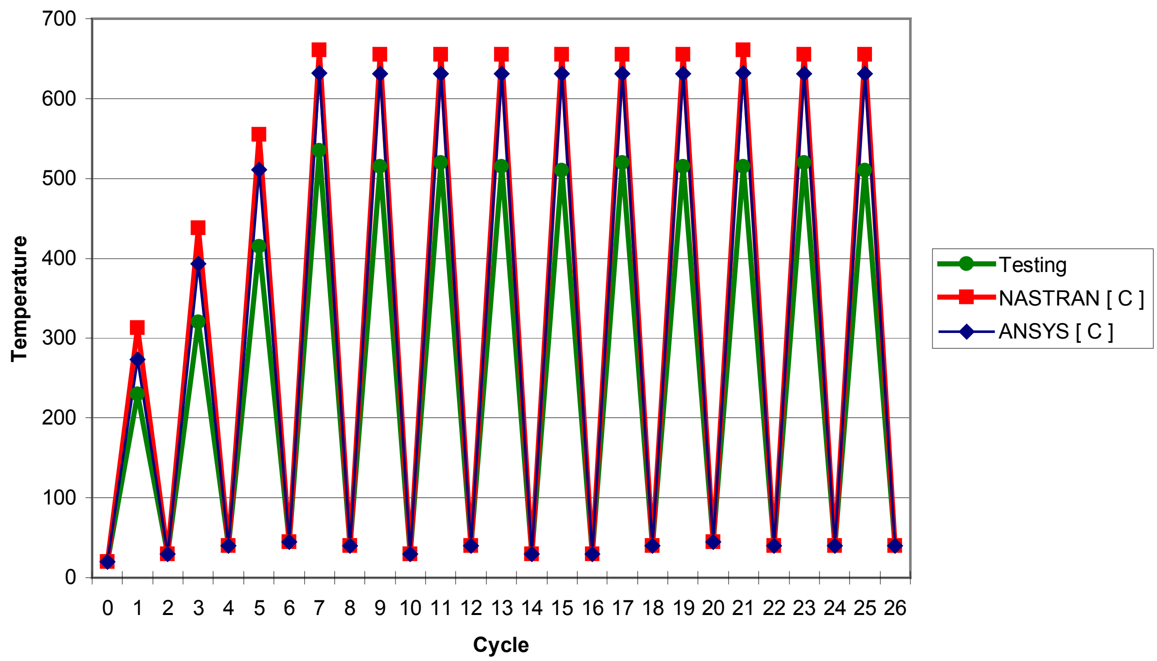


Fig.15. Temperature diagram for nodes corresponding to thermocouples placing

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.
7. 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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