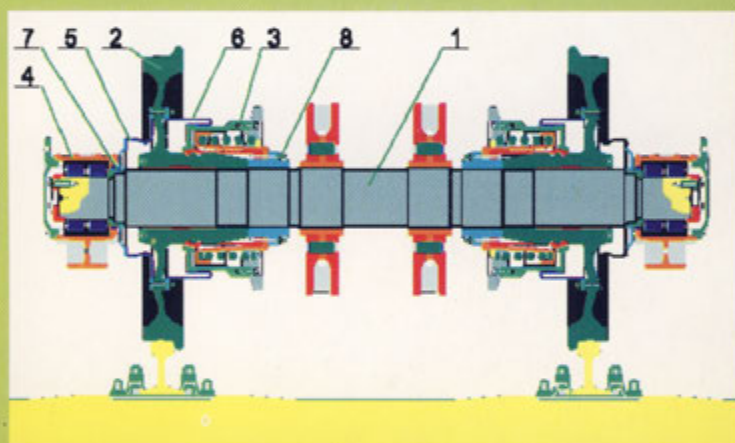


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RAILWAY WHEELSETS



59

MONOGRAPH



SILESIA UNIVERSITY OF TECHNOLOGY



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DESIGN AND INVESTIGATION OF RAILWAY WHEELSETS

Summary. In the present state-of-the-art the design process can be simplified and accelerated, using computer simulations based on finite element method (FEM). However, these methods are not error-free, since the model as well as computation algorithm are simplified. The error depends, therefore, on the software used. Noting the multitude of software present at the market, it seems reasonable to evaluate the discrepancy in the results between different types of software used. The representative data input into software should come from laboratory or service tests. The railway wheelsets designs in operation at present are the outcome of service experience and process engineering. Lately, the numerical analyses done in the design phase are also utilised. Emphasizing the extensive possibilities and usefulness of FEM, attention must be paid to the judicious use of this method. It is an approximate method, since the results relate to the models of real systems and not to the systems themselves. There exists a difference between real problem and computation results, which relate to the model. In the last stages of modelling the software used plays a significant role in the results obtained.

1. 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, underground subway) 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) means 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 railway wheelset is one of the more important elements of the rail vehicle, since it carries most of forces operating during train run. The prospective increase in speeds and specific loads causes a greater stress to be laid upon the role played by the wheelset as a load-carrying element. It becomes important to improve the run conditions of the wheelset, in particular where its reliability is concerned. The railway wheelsets designs used at present are the

outcome of service experience and process engineering. Lately, the numerical analyses of stress and strain done in the wheelset design phase are also utilised [5].

However, both at home and abroad the wheelset design numerical calculation results used are burdened with errors arising from [6-13]:

- lack of general methodology of railway wheelset design process
- not taking into account all loads of railway wheelsets (due to their production engineering and service parameters);
- absence of experimental confirmation of numerical computation results;
- lack of comparison of different computation procedures and software packages used in different research centres for the identical input data used in calculations.

These considerations justify the attempt to increase durability and reliability of the railway wheelset wheels during their design, using numerical calculations taking account of the above points.

2. The research centres at home and abroad working on wheesets

The railway wheelset is probably the least modified rail vehicle element in the whole of railways' history. During last few years, however, the research into the durability and reliability of the railway wheelset has been intensified. This research has been carried by groups of experts working in universities or large manufacturing plants.

Polish research is mostly conducted at universities. This is due to absence of prominent manufacturers of railway wheelset parts here.

Analysing books on the subject published lately and papers published in the years 1998-2003, four major academic centres working on railway wheelsets can be distinguished. These are: Politechnika Warszawska (Warsaw University of Technology) [14-16], Politechnika Krakowska (Cracow University of Technology) [9,17-28], AGH Kraków (Mining and Steel Academy in Cracow) [16,29] and Politechnika Śląska (Silesian University of Technology)[30-48]. Taking into account the number and diversity of publications, the Department of Railway Engineering of Silesian University of Technology is seen to be an outstanding research centre. The numerical and experimental investigations of rail vehicles wheels were started by Prof. Roman Bąk and his co-workers [6,44,49-55]. At present the subject matter is being researched into by the team led by Prof. Marek Sitarz [30-48]. In the mid-seventies the first Polish software package called KOŁO PC was worked out by this team. It has been constantly improved and modernised. This software, based on FEM, makes possible the calculations of strain and stress components at a given point of the railway wheelset. It was among the first European programs used by UIC for numerical calculations of railway wheelsets.

The research conducted in Silesian University of Technology can be sub-divided into the following topical ranges:

1. Application of mechanics in wheelset investigation concerned with wheelset quality [44,51,54,55];
2. Construction and identification of railway wheelset models during design, manufacture and service phases [39,42,56];
3. Service tests of wheels of railway wheelsets [50,47];
4. Metallographic and fatigue investigation of railway wheelsets [32,37,38];
5. Servicing and repair of railway wheelsets [30,57,58] [44].

The railway R&D centres such as CNTK in Warsaw [58,59] and OBRPS in Poznań [60-63] also work intensively on experimental investigation.

Abroad the research teams are often set up as a result of collaboration of academic centres with manufacturers of railway wheelsets. The best research results achieved are immediately implemented in the countries hosting the prominent wheelset manufacturers, such as: Italy [64-78], France [76,77,79,80], Czech Republic [81-88], Sweden [4,76,77,89-93], Russia [76,77,94-99], Ukraine [93,100-116].

Of course, other countries also significantly contribute to the development of railway wheelsets: UK [76,77,106,107,117-119], Germany [76,77,120-122], Spain [111], China [76,77,110], Japan [76,77,112,123], Australia [124-127], India [128].

Nowadays, European Centres of Excellence play important role in the European research. These Centres have set up interdisciplinary and international teams of researchers in order to solve specific problems. We can quote here CHARMEC Centre of Technical University in Goeteborg (Sweden) and TRANSMEC Centre of Silesian University of Technology in Katowice, Poland. These centres are partly financed by the European Union. They conduct basic and applied research; they also organise seminars, training courses and conferences aimed at knowledge and experience exchange between scientists from all Europe and scientific development of young researchers working on railway wheelset-related problems.

It may be noted that strength issues related to railway wheelsets cover a wide range of problems. It must be emphasised that the recent ten years are characterised by accelerated research into the railway wheelset issues in the different phases of its service. The resultant progress is due to implementing of numerical computational methods, which are used in the strength analyses more often.

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, we may conclude 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 a 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 elements),

- 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, based on finite element method (FEM), is used. FEM numerical calculations of different wheels of railway wheelsets are investigated at the Department of Railway Engineering. Three at the software packages are used – ANSYS, NASTRAN and COSMOS. This diversity of software should make it possible to compare 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 – web 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 web of a wheel model. Figure 2 shows the web of a wheel model together with physical and discrete models generated by the software.

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.

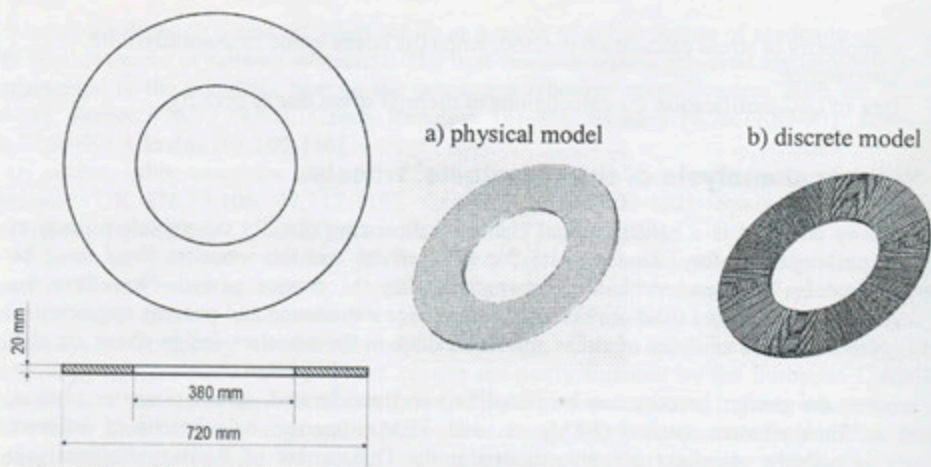


Fig. 2. Model of the web of a wheel

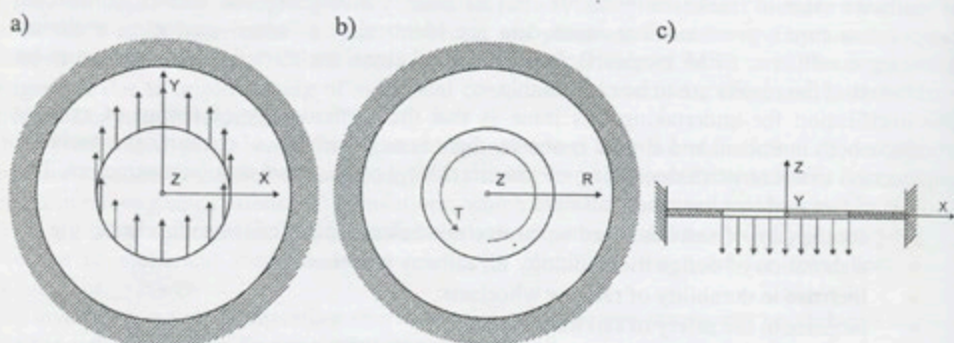


Fig. 3. Load models for the web a wheel

The maximum strains for the presented loads have been calculated analytically. The data: web thickness $h = 0,02$ [m], inner radius outline $r_1 = 0,019$ [m] and inner radius outer $r_2 = 0,36$ [mm] [103]:

Case # 1

$$\sigma_{r_{\max}} = \frac{\mu \cdot \Delta_y}{c} \left[\frac{3 \cdot \kappa (r_1^2 + r_2^2)}{r_1} + 2r_1 + \frac{\kappa^2 (r_1^2 + r_2^2)}{r_1} - 2 \frac{\kappa r_2^2}{r_1} \right] \quad (3.1)$$

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

$$\mu = \frac{E}{2(1+\nu)}$$

$$\kappa = \frac{3-\nu}{1+\nu} \quad (3.2)$$

E – Young's modulus

ν – Poisson's constant

Δ_y – pre-set displacement = 0,0001 [m]

Coefficient c has been determined with the help of formula:

$$c = r_1^2 - r_2^2 - \kappa^2 (r_1^2 + r_2^2) \ln \frac{r_1}{r_2} \quad (3.3)$$

For this equation

$$\sigma_{r \max} = 143,9 \text{ MPa} \quad (3.4)$$

Case # 2

$$\tau_{\varphi \max} = \frac{2 \cdot \mu \cdot r_1^2 \cdot \zeta}{r_2^2 - r_1^2} \quad (3.5)$$

where ζ - torsion of inner contour, torsion angle is

$$\zeta = \frac{\Delta \theta}{r_1} \quad (3.6)$$

for $\Delta \theta = 0,0001$ [m]

$$\tau_{\varphi \max} = 110,5 \text{ MPa} \quad (3.7)$$

Case # 3

$$\sigma_r = 6 \frac{M(r)}{h^2} \quad (3.8)$$

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

$$M(r) = D \left[\frac{d^2}{dr^2} w(r) + \frac{\nu}{r} \frac{d}{dr} w(r) \right] \quad (3.9)$$

depends on the displacements distribution of the web

$$w(r) = \frac{T r_2}{8\pi D (\eta^2 - 1)} \left\{ \left[\ln(\eta) - 1 + \frac{\eta^2 + 1}{2} \right] \left[1 - \frac{r_1^2}{r_2^2} \right] - \left[2 \ln(\eta) + (\eta^2 - 1) \frac{r_1^2}{r_2^2} \right] \ln \left(\frac{r_2}{r} \right) \right\} \quad (3.10)$$

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

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

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

$$c_1 = \frac{\left[\ln(\eta) - 1 + \frac{\eta^2 + 1}{2} \right] \left(1 - \frac{r_1^2}{r_2^2} \right) - \left[2 \ln(\eta) + (\eta^2 - 1) \frac{r_1^2}{r_2^2} \right] \ln \left(\frac{r_2}{r_1} \right)}{8\pi D (\eta^2 - 1)} r_2 \quad (3.11)$$

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

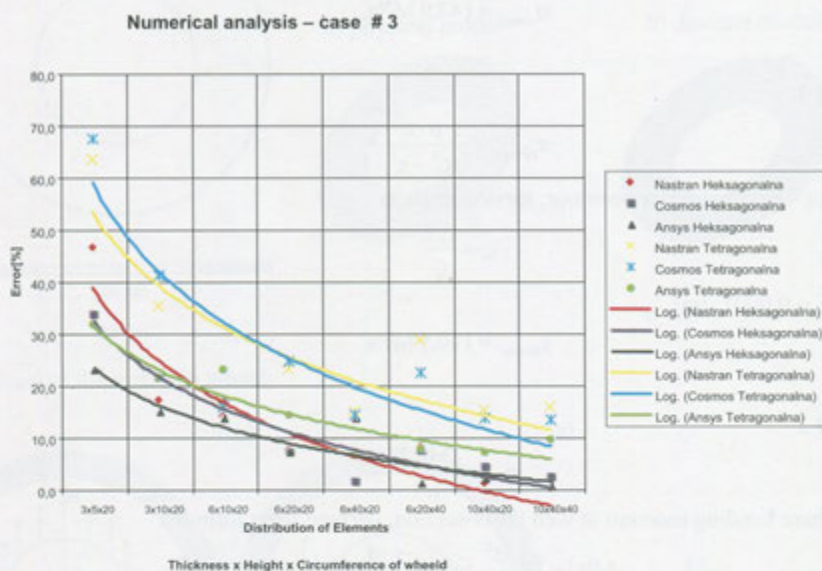
$$\sigma_{r \max} = 60,5 \text{ MPa} \quad (3.12)$$

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.

a)



b)

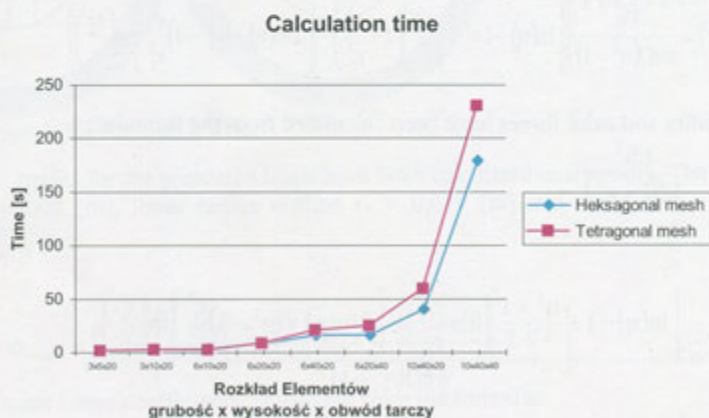


Fig.4. Results of comparison of analytical and numerical calculations - case #3

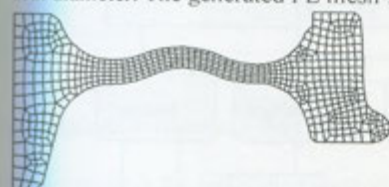
The analysis shows that the minimum number of elements along the lateral cross-section of the wheel web should be equal to six.

These investigations have led to generating an optimum mesh for the web with respect to computation accuracy and time.

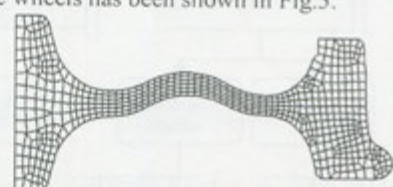
3.2. Subject of investigation – wheel design of railway wheelset

The work was principally aimed at elaborating methods of selection of constructional parameters for the railway wheelset wheels. The conducted calculations of railway wheelset 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.

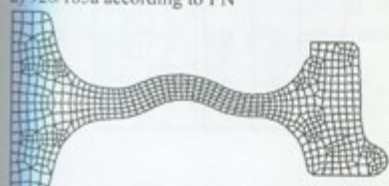
The numerical investigation has been run for ten different geometrical designs of wheels \varnothing 920 and \varnothing 920 – h (worn out), differing in geometrical parameters of the web, hub and wheel seat diameter. The generated FE mesh for these wheels has been shown in Fig.5.



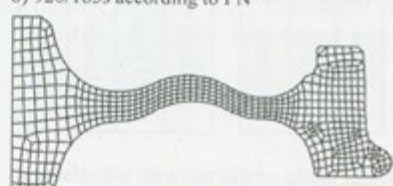
a) 920/185a according to PN



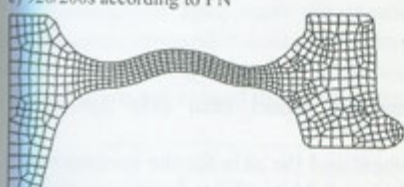
b) 920/185s according to PN



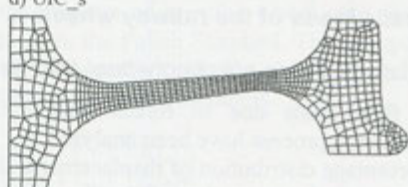
c) 920/200s according to PN



d) UIC_s



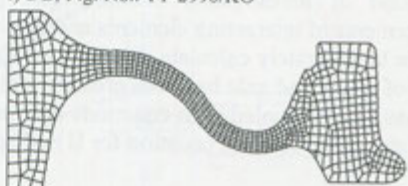
e) UIC_a



f) DB, Fig. No.. TT-2990/KO



g) FS, Stabilimento di Lovere Lucchini



h) ČD, Bohumin Fig. No.. 455.0.212.000.12



i) ČD, Bohumin Fig. No..FWG\302.0.02.001.007



j) ГОСТ 9036-88

Fig. 5. Discrete models of the investigated monoblock wheels

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.

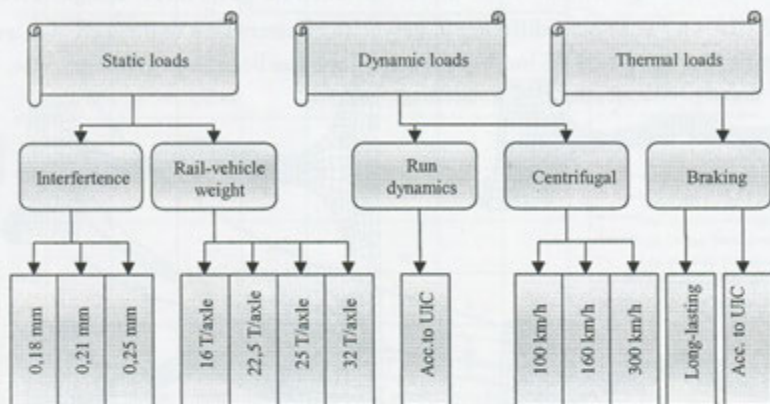


Fig. 6. Wheel loads of the railway wheelset

3.3. Static loads of the railway wheels

3.3.1. Loads due to pressing wheel onto axle

At first, loads due to forces arising from pressing wheel onto axle during the manufacturing process have been analysed. The percentage distribution of displacements in the wheel and the axle for the minimum and maximum interference value set by the standards (0.18 – 0.25 mm) has been investigated. This kind of investigation should make possible finding out of the proportion of displacements of interacting elements and this in turn should greatly simplify the analysis. In order to accurately calculate displacements and stresses due to assembly, an asymmetrical model of wheel and axle has been created, with the adequate number of nodes. Next the node pairs have been coupled with equations determining their displacements. Figure 7 shows this model and corresponding equation for IJ node pair (3.13):

$$\begin{aligned}
 v_i &= v_j \\
 u_i - u_j &= \Delta \\
 1 \cdot v_i - 1 \cdot v_j &= 0 \\
 1 \cdot u_i - 1 \cdot u_j + v_k &= 0 \\
 &\Downarrow \\
 &\Delta
 \end{aligned} \tag{3.13}$$

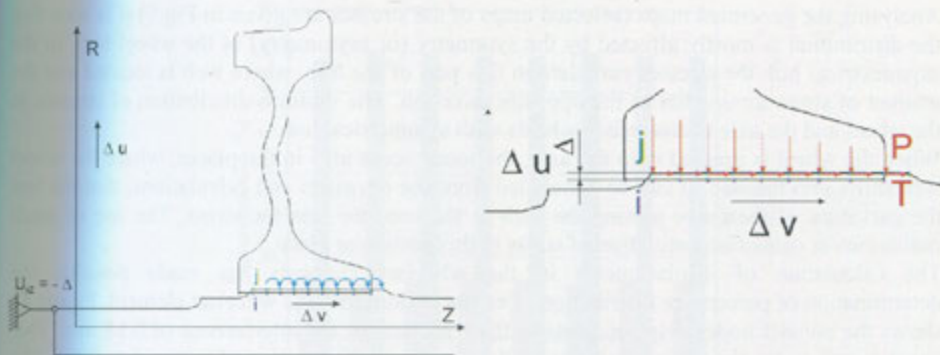


Fig.7. Axially symmetrical load model

In order to check the calculation method for the calculated force P value (reaction in the radial direction) and friction coefficient μ determined during pressing tests conducted in Huta Gliwice Steel Plant, the pressing force T has been determined for subsequent calculation phases:

$$T = \mu \cdot P$$

$$T_z = \sum_{i=1}^n T_i \quad (3.14)$$

with the contact nodes from i to n, where T_i is the value of the axial force in the node. These calculations have made the creation of the diagram of pressing wheel onto the axle possible and compared it with the diagram set down in the Polish Standard. This comparison has confirmed the correctness of the calculation method used – Fig.8. The results of analytical calculations do not exceed the chart range specified by the Polish Standard.

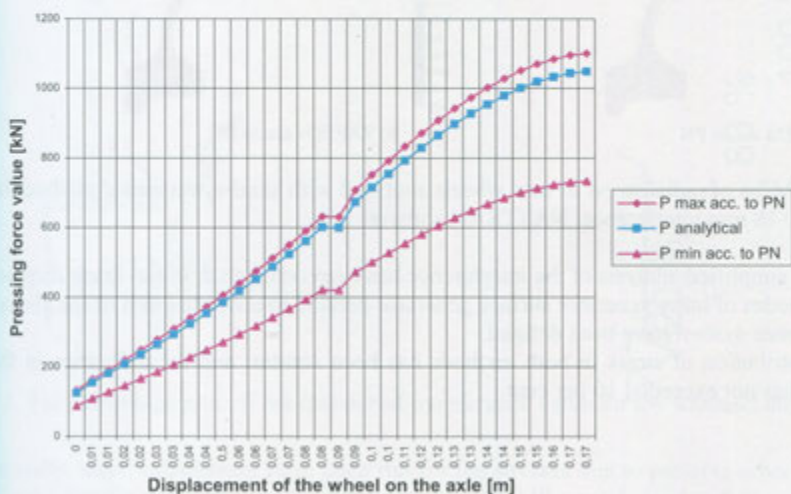


Fig. 8. Theoretical diagram of pressing wheel onto the axle – wheelset PN 920/200s

Analysing the generated maps (selected maps of the stresses are given in Fig.9) it is seen that the distribution is mostly affected by the symmetry (or asymmetry) of the wheel hub. In the asymmetrical hub the stresses cumulate in this part of the hub, where web is located and the amount of stress diminishes in the opposite direction. The uniform distribution of stresses in the wheel and the axle characterises wheels with symmetrical hub.

When the wheel is pressed onto the axle, the loads occur also in the places, where the wheel web shifts into the hub. It can be concluded from the drawings and calculations, that the less the curvature of the curve joining the web to the hub, the less the stress. The use of small radius curves causes accumulation of stress in this particular place.

The calculation of displacements in the wheelset elements has made possible the determination of percentage distribution of contact nodes into the wheelset element. Figure 10 shows the contact nodes displacements in the wheelset for the interference of 0,18 mm. This value seems to be almost independent of the shape of wheel web and is equal to 80%. The axle takes up the remaining 20% of the displacement.

The conducted analysis has led to the comparison of the influence of interference value on stress distribution and determination of percentage distribution of interference on different wheelset elements (wheel, axle):

- distribution of the stresses in the wheel does not change (the maximum stresses occur in the wheel hub and web bend);
- the stress value changes in proportion to interference value
- the percentage ratio of the interference transfer is constant for both interference values

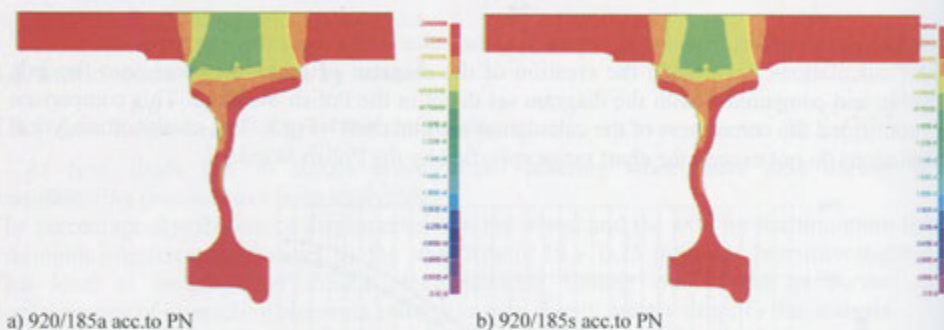


Fig. 9. Maps of radial stress of new wheels analysed with axial-symmetrical method for 0.18 mm interference. NASTRAN software

Next, a simplified analysis of the interference has been conducted. Initial linear displacements of the nodes of inner generator surface generator parallel to the radial axis of the global polar co-ordinate system have been defined.

The distribution of stress in both methods has been similar, and the difference in the stress values has not exceeded 10 per cent.

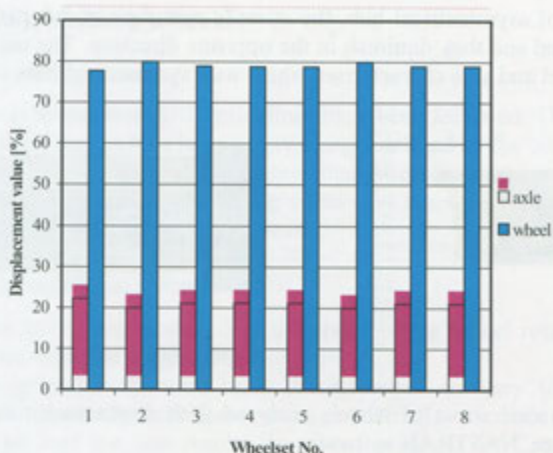


Fig. 10. Displacement of contact nodes in the wheelset

The finding out of the constant proportion of the displacements (i.e. independent of wheel geometry) has made possible the following:

- defining the load caused by pressing wheel onto the axle by the initial displacements
- shortening of model definition time;
- decreasing the computation time.

Further calculations will be simplified, by setting a pre-determined percentage value of the interference of the element of the railway wheelset – Fig. 11.

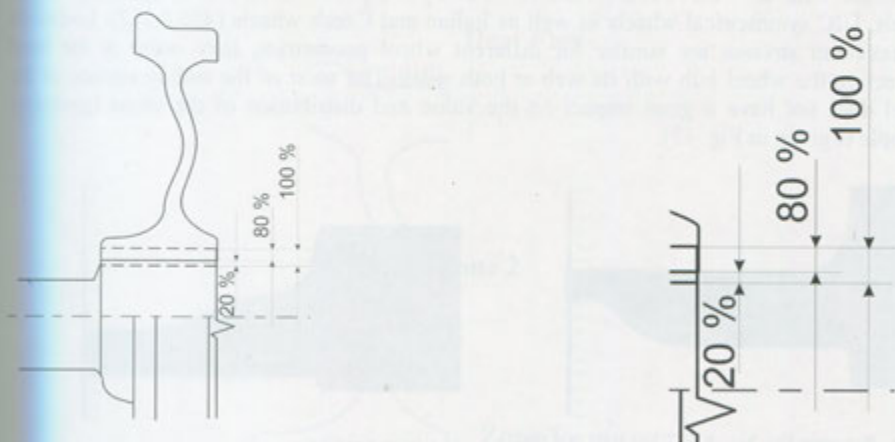


Fig. 11. The percentage ratio of the transfer of interference value for the wheelset elements

Additionally, some calculations have been run for the stresses due to pressing wheel onto the axle for the interference value used in Huta Gliwice Steel Plant and equal to 0,21 mm, for ten different wheel designs (the selected designs have been shown in Fig.12). The analysis has not proven any significant influence of the wheel geometry on the stress value (web geometry, the wear of wheel rolling surface). The impact of hub symmetry (or asymmetry) is

great. In the case of asymmetrical hub, the stresses cumulate in this part of the hub, where wheel web is located and they diminish in the opposite direction. The uniform distribution of stresses in the wheel and axle characterises wheel with symmetrical hub.

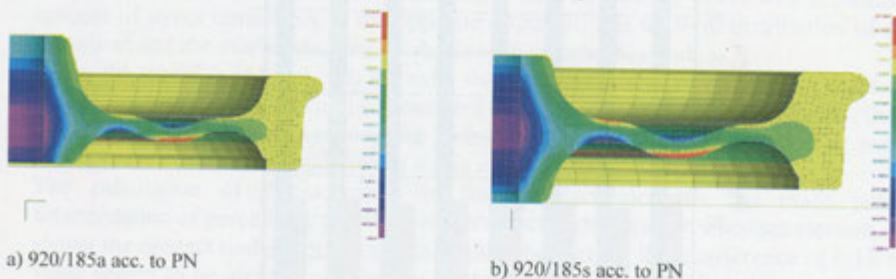


Fig. 12. Maps of radial stress of wheels analysed with displacement method for 0.18 mm interference. NASTRAN software

3.3.2. Calculations simulating carriage weight

Forces arising from carriage weight constitute another static load. The value and point of application of these forces have been simulated according to UIC guidelines set out in ERI Report B169.1 of 1998 [129].

Axle load of 22,5T per axle is the maximum load set down by Polish Standard for freight carriages. The analysis of maximum reduced stresses caused by the carriage weight has indicated wheels better adapted for carrying huge freights. It is obvious that the most advantageous results for monoblock wheels have been obtained for Russian wheels, since the wheel diameter is greater. However, this type of wheel has achieved the best results in case of maximum wear as well. Good results as to the yield point have been obtained by Polish wheels, UIC symmetrical wheels as well as Italian and Czech wheels (455.0.212). Locations of maximum stresses are similar for different wheel geometries; they occur at the bend connecting the wheel hub with its web at both sides. The wear of the rolling surface of the wheel does not have a great impact on the value and distribution of the stress (geometry example is given in Fig. 13).

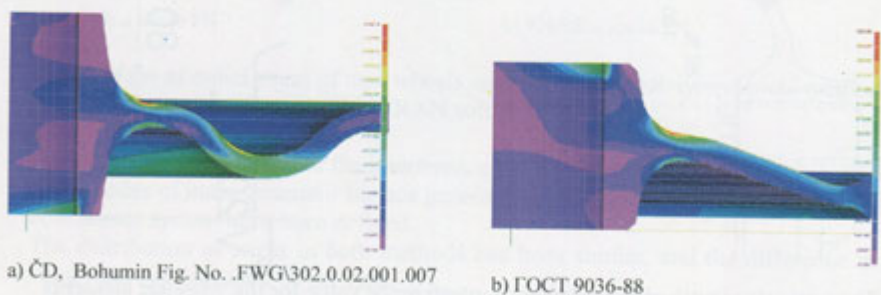


Fig. 13. Maps of reduced stresses due to carriage weight, axle load 22,5 T per axle. NASTRAN software

3.4. Dynamic loads of railway wheels

3.4.1. Loads due to run dynamics

Where run dynamics is in question, UIC guidelines have been followed. The loads have been modelled in a quasi-static way, with a huge safety margin defined by the coefficient [129]. The values of these stress are three times greater than stresses due to carriage weight, the distribution is similar. The wear of the rolling surface of the wheel does not have a great impact on the value and distribution of these stresses.

3.4.2. Loads due to centrifugal forces

Centrifugal forces have been modelled as the speed of the wheel rotating round z-axis. Depending on software used, the units are rpm or rad/s.

In relation to stress generated by other loads these stresses are very small. They can be neglected in further calculation for wheelsets running at normal run speeds (up to 160 km/h).

In case of this type of load the best results have been obtained for Czech wheel Fig. No. FWG\302.0.02.001.007 and Italian wheel. The maximum stress for every analysed design is centred in the wheel web. As before, the wear of the rolling surface of the wheel does not have a great impact on the value and distribution of these stresses

3.5. Thermal loads of railway wheels

3.5.1. Loads due to prolonged braking

The last type of load in question is braking. During analysis, an extreme case of prolonged braking has been modelled (identical to braking taking place in Gotthard Pass [130]). The length of this route is c. 30 km with the average slope of 20,7 and maximum slope of 27%. Figure 14 shows the convection zones and way of clamping the brake shoe.

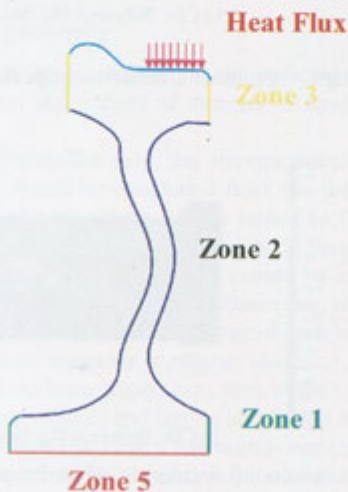


Fig. 14. Modelling thermal loads due to prolonged braking

The convection values and material properties have been taken from ERI Report 169 of 1987. This report describes investigation of different cases of braking UIC wheel, axle load of 22,5T per axle, at the run speed of 60 km/h (wind puff 0.6* run speed) and braking power of 30 kW. The braking time was 2700 s. During the analysis it has been assumed that 70% of the braking power is transferred to the wheel and the remaining 30 % is absorbed by the brake shoe. The value of the heat flux for the shoe 9- mm wide has been calculated according to the formula. Applying the above assumptions, calculations of the railway monoblock wheel have been conducted, simulating a prolonged braking.

As could be expected, the lowest maximum reduced stresses have been obtained for the Russian wheel (greater wheel diameter). However, when identical simulation has been run for the worn wheels, this wheel has been marked as the fifth. The overall good results have been obtained by modern-design wheels, with a considerable web bend: Italian wheel and Czech wheels (455.0.212 and 302.0.02.001.007). The wheel designed by Stablimento di Lovere Lucchini has shown the best results for the wheel with maximum wear of the rolling surface. The maximum stresses have always accumulated in the wheel web bends.

In case of stresses due to braking, the impact of the wear of the rolling surface on the wheel strength has been observed. In extreme cases the difference in the maximum reduced stresses has been as high as 14 per cent (German wheel TT-2990/KO).

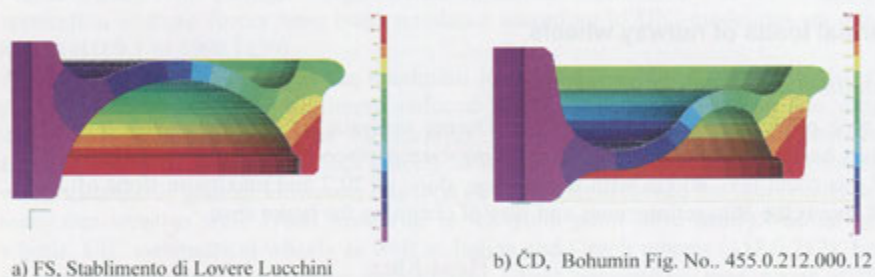


Fig. 15. Temperature distribution maps – results of simulation of prolonged braking. NASTRAN software

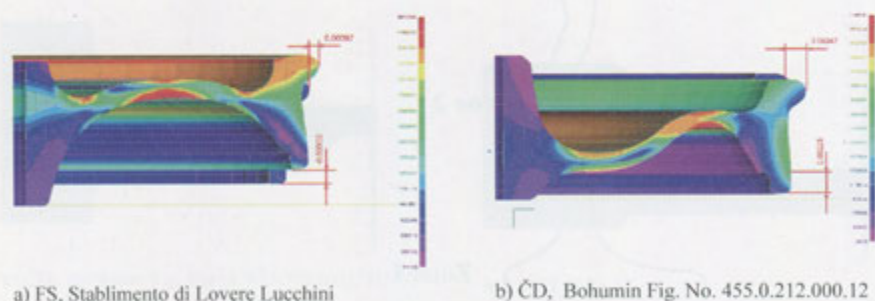


Fig. 16. Maps of reduced stresses – results of simulation of prolonged braking. Geometry of maximum displacements in both directions has been shown as well. NASTRAN software

3.6. Stress superposition

After the numerical calculations of the several stresses, the superposition of these stresses has been conducted for all the wheels, in accordance with the Polish Standard: PN-92/K-91019: Wagony. Kola Bezobrózowe. Typy i Wymiary (Carriages. Ringless Wheels. Types and Sizes).

The stress distribution has been shown for a Polish wheel PN 920/200s – Fig. 17. The concentration of stresses in this part of the web suggests that the decisive impact in the superposition is born by dynamic stresses and stresses due to braking.



Fig.17. Stress superposition for PN 920/200s wheel

3.7. The influence of design and service parameters on the stress distribution on the railway wheel

3.7.1. Impact of the web geometry

The numerical analysis has demonstrated an insignificant influence of the web of wheel geometry on the distribution and values of stresses in case of assembly and static stresses (carriage weight).

When the wheel is pressed onto the axle, the stresses occur in the place, where web of the wheel connects to the hub. It can be concluded from the drawings and calculations, that the smaller the curvature of the curve where web is joined to the hub, the smaller the stresses. Using small radius curves leads to stress accumulation in these places.

Places, where maximum stresses occur for loads caused by carriage weight are similar for all wheel geometries and are localised at the curve connecting web of a wheel to the hub, at both sides. It is obvious that the best results for the monoblock wheel have been obtained for the Russian wheel, since the wheel diameter is greater. However, this wheel has also achieved the best result in case of maximum wear. Good results as to the yield point have been achieved by Polish and UIC wheels (symmetrical) and Italian and Czech wheels (455.0.212).

In case of loads due to prolonged braking it has been noted that geometry of the wheel bears a significant impact on the distribution of reduced stresses and, in turn, on the increased wheel strength. The maximum stresses have accumulated in the web of a wheel bend. The best results have been obtained by modern-design wheels with a large web bend – Italian wheel and Czech wheels (455.0.212 and 302.0.02.001.007). The wheel designed by Stablimento di Lovere Lucchini has generated best results for the wheel with maximum wear of a rolling surface.

The values of the reduced stresses for modern-design wheels have been decreased by 70 Mpa (comparison of Italian wheel and UIC asymmetrical wheel, which is widely used – Fig. 18).

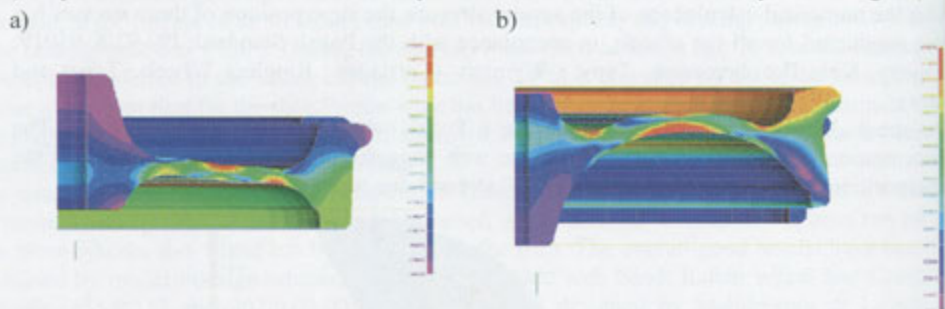


Fig. 18. Comparison of the stress distribution of reduced stresses due to prolonged braking: a) UIC symmetrical wheel, b) Lucchini wheel

3.7.2. Impact of the wheel hub geometry

In case of symmetry or asymmetry of the web of a wheel no noticeable impact on the reduced stress for any load type has been observed. However, change of geometry of the web influences the location of stress in case of load due to pressing wheel onto the axle. For the asymmetrical hub, more stress accumulates in this part of the hub, where the web is located; the stresses diminish in the opposite direction. Wheels with symmetrical hubs are characterised by uniform distribution of stresses in wheel and axle – Fig.19.

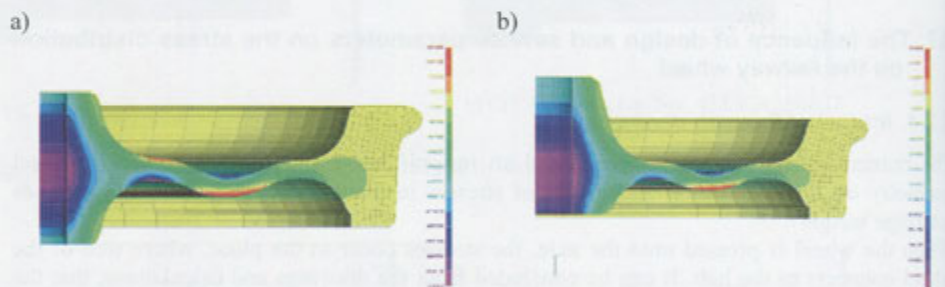


Fig. 19. Distribution of reduced stress caused by pressing wheel onto the axle for wheel in accordance with Polish Standard: a) 920/185 symmetrical wheel, b) 920/185 asymmetrical wheel

3.7.3. Influence of the wheel seat diameter

Significant impact of the wheel seat diameter can be observed in case of the loads due to carriage weight and run dynamics only. The wheels with greater wheel seat diameter (200-267 mm) have generated lower values of the reduced stresses during the numerical analysis. The distribution of the reduced stresses has not changed for all wheel geometries, and stress maximum values are located at the curve connecting wheel hub with web of the wheel on both sides – Fig.4.14.

A good example of the influence of wheel seat diameter is the comparison of two wheel types – German wheel (Technical Specification No. TT2990) and Russian wheel – acc. to ГОСТ 9036-88. There is a difference in the wheel seat diameter, while the shape of the web is similar. The Russian design is characterised by wheel seat greater by 80 mm (TT2990 =

180 mm, Γ OCT 9036-88 = 263 mm). This design has led to halving the stresses due to static carriage weight and run dynamics – Fig. 20.

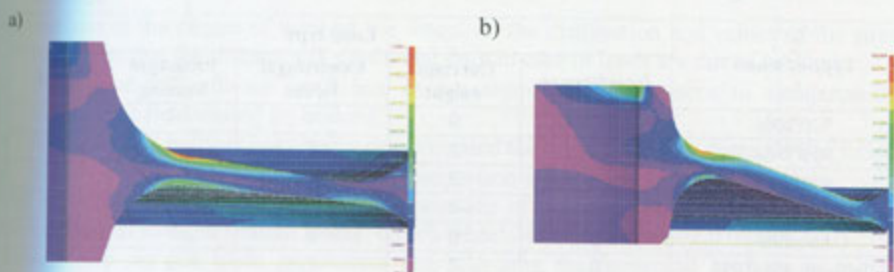


Fig. 20. Reduced stresses due to carriage weight: a) TT2990, b) Γ OCT 9036-88

3.7.4. Impact of wheel rolling surface wear

The impact on the wheel strength of the wear of wheel rolling surface has been observed in case of loads due to prolonged braking only (30 kW power, 45 s time interval). The numerical analysis demonstrates the need for investigation and numerical calculations for wheel designs with maximum wear of the rolling surface.

In extreme cases the difference in the maximum reduced stresses has been as high as 14 % - German wheel TT-2990/KO – Figure 21.

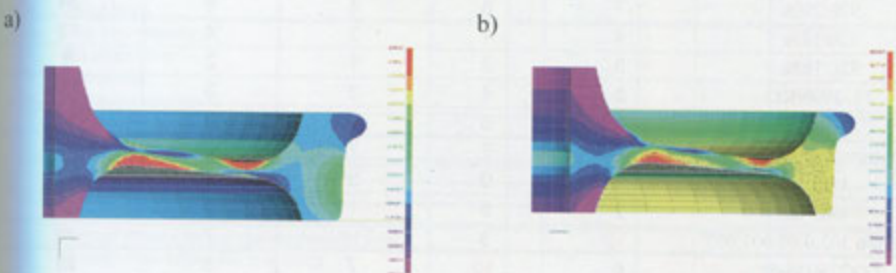


Fig. 21. Distribution of reduced stresses caused by prolonged braking of the German wheel TT2990: a) wheel \varnothing 920, b) wheel \varnothing 854

3.7.5. Summary comparison of ten different designs of railway wheels

In order to indicate the most advantageous design with respect to reduced stresses for given load condition, the wheels with lowest values have been given the highest marks (10. Ten different wheel designs). In the same way, inferior wheels received lower marks. The evaluation has been conducted for all load cases, differentiating between new wheels and worn wheels with maximum wear of the rolling surface – Tables 3 and 4.

The marks have been added up in order to select the best design in service.

New wheel – diameter \varnothing 920 mm

Type of wheel	Load type				
	Interference	Carriage weight	Centrifugal forces	Prolonged braking	Total
920/200s	7	9	1	4	21
920/185s	3	2	4	3	12
920/185a	3	3	1	1	8
TT-2990/KO	5	4	4	6	19
Lucchini	8	6	9	7	30
Bohumin 455.0.212	9	7	7	9	32
UIC_s	1	8	4	5	18
UIC_a	2	5	1	2	10
Bohumin 302.0.02.001.007	9	1	10	8	28

Wheel with worn rolling surface - \varnothing 854 mm

Type of wheel	Load type				
	Interference	Carriage weight	Centrifugal forces	Prolonged braking	Total
920/200s	7	8	3	6	24
920/185s	4	1	3	4	12
920/185a	3	2	1	2	8
TT-2990/KO	5	4	3	3	15
Lucchini	8	6	9	9	32
Bohumin 455.0.212	9	7	7	8	31
UIC_s	1	9	3	9	22
UIC_a	2	5	1	1	9
Bohumin 302.0.02.001.007	10	3	10	7	30
ГОСТ9036-88	6	10	7	5	28

The best results have been achieved by Czech wheels (455.0.212 and 302.0.02.001.007) and Italian wheel.

Taking into account that these are the wheels of modern design (still in the design stages or testing stage), it is important to conduct the numerical analyses and investigations, since these new designs may avail real financial advantages, and increase the safety and travel comfort of future high speed rail vehicles.

3.8. Remarks and conclusions

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 web 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 are 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 and 302.0.02.001.007).

Table 5 sets out the basic parameters characterising the numerical analysis in the three different software packages.

Table 5

Comparison of software used in strain analysis of railway wheelsets

Parameter	Method		
	NASTRAN	ANSYS	COSMOS
1. Facility of data preparation			
• creation of mesh	+	*	+
• input of material data	*	+	+
• time-consumption	+	+	+
• dependence on temperature	*	*	*
• setting boundary conditions in displacements	*	*	*
• setting loads	+	+	+
2. Time consumption of:			
• data preparation	+	+	+
• computation	*	+	+
• results visualisation	*	*	*
3. File import	*	+	+
4. Contact issues	*	*	*
5. Static issues	*	*	*
6. Dynamic issues	*	*	*
7. Nonstationary heat flow	-	*	*
8. Thermal stresses for a given temperature field	*	*	*
9. Thermal stress issues due to braking	*	*	*
• time consumption	+	+	+
10. Facility of accounting for assembly stresses (interference)	-	-	-
11. Facility of accounting for centrifugal forces	*	*	*
12. Simple wear analysis	-	-	*
13. Superposition of stresses and displacements	*	*	*

4. Experimental investigation of temperature, deformation and stress fields of railway wheels

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:

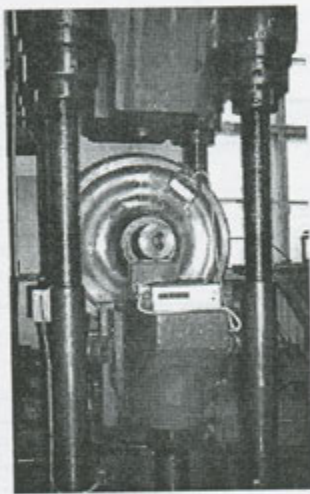
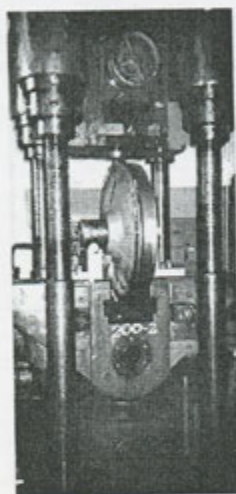
- static loads simulating the carriage weight for different load values and different load models,
- thermal loads for 13 braking cycles, braking power of 20, 30, 40 and 50 kW.

In the tests German and Russian wheels have been used.

4.1. 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. 22a.

Strain gauges have been placed on the internal and external surface of the web of a wheel in the vertical plane (Fig. 22b). The wheel has been loaded with the vertical force (Fig. 22b) 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 action



b)

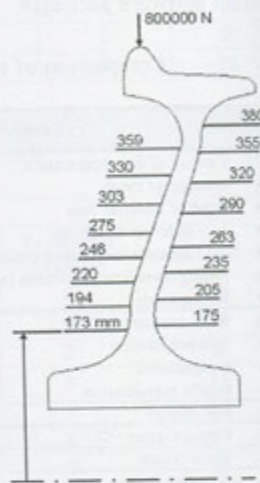


Fig. 22. Test stand for strain investigation of railway wheels deformation: a) the stand, b) placing of strain gauges, way of modelling the wheel

The result has been obtained in the form of diagrams showing the courses of radial stresses at the internal and external wheel surfaces – Fig. 23.

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):

$$\sigma_R = \frac{E}{1-\nu^2} (\varepsilon_R + \nu\varepsilon_T), \quad (4.1)$$

$$\sigma_T = \frac{E}{1-\nu^2} (\varepsilon_T + \nu\varepsilon_R)$$

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 are shown in Fig.24.

The outcome of the numerical analysis persists in deformation values of wheel in places corresponding to placing of the strain gauges – Table 6.

Table 6 and Figs. 25 and 26 show that modelling the load as a single force does not exactly correspond to the experiment results. The discrepancy of the results is great.

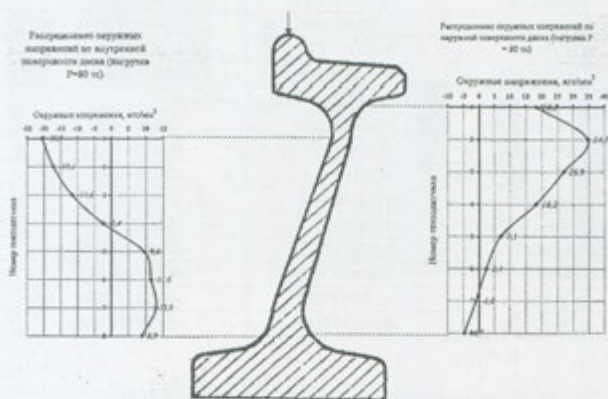


Fig. 23. The diagrams of the radial stresses at the internal and the external side of the web of a wheel

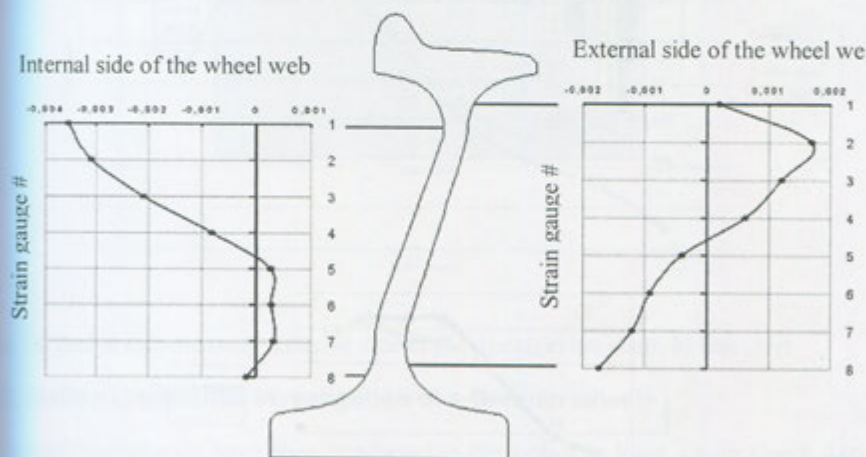
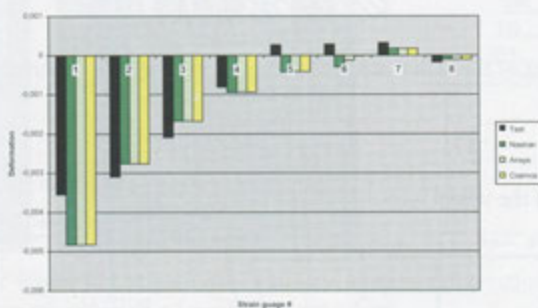


Fig. 24. The diagrams of the deformations at the internal and the external side of the web of a wheel

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)



b)

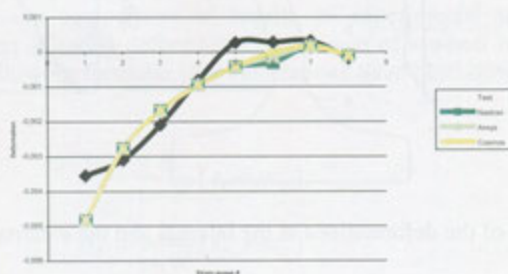
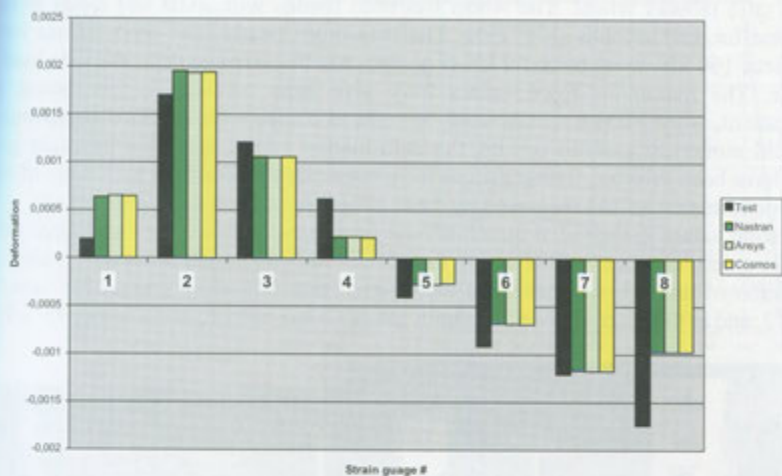


Fig. 25. Radial deformation – internal side of the wheel: a) bar chart, b) line chart

a)



b)

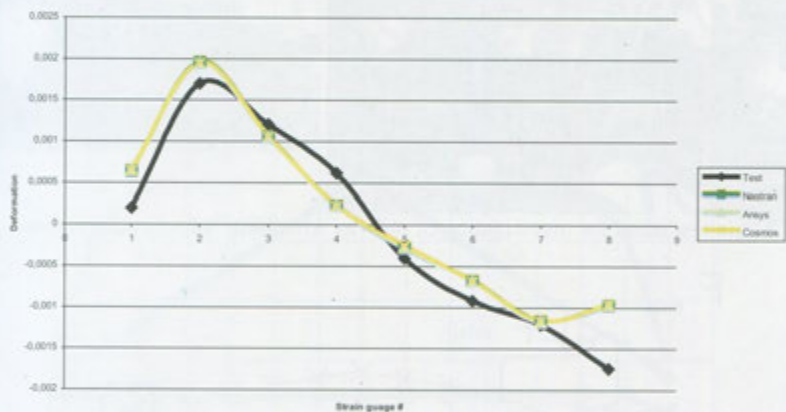


Fig. 26. Radial deformation – external side of the wheel: a) bar chart, b) line chart

4.2. 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. 27. Strain gauges have been placed on the internal and external surface of the web of a wheel in the axial and radial directions (Fig.28).

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. 28). The maximum steady-state force values have been: radial force 160 kN, axial force 60 kN (Fig. 29), 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. 30.

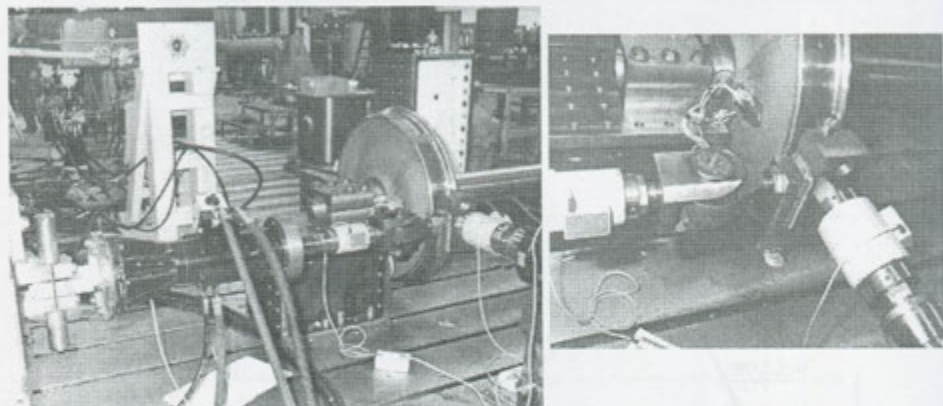


Fig. 27. Strain test stand for investigation of railway wheelsets' wheels deformations

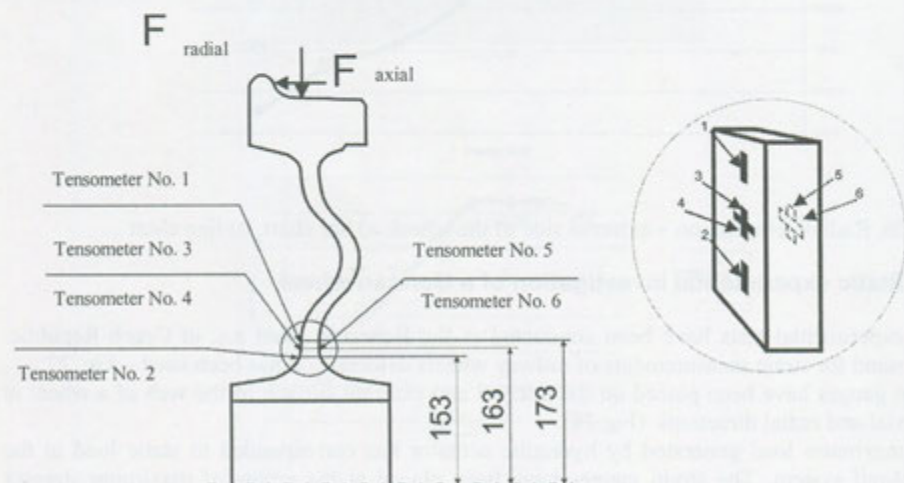


Fig. 28. Loading diagram for a wheel placed in the test stand; strain gauges placed on the wheel

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

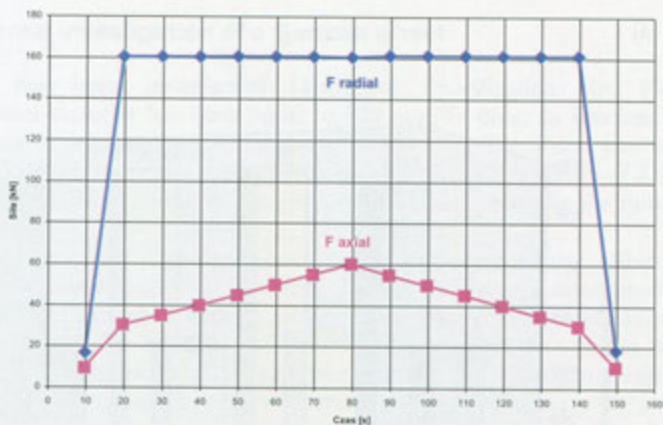


Fig. 29. Time cycles of forces loading the wheel at the test stand

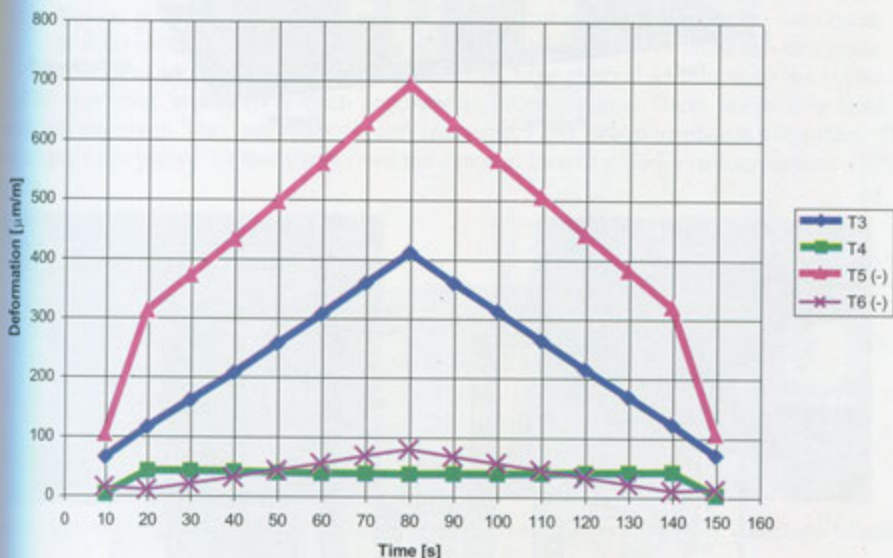
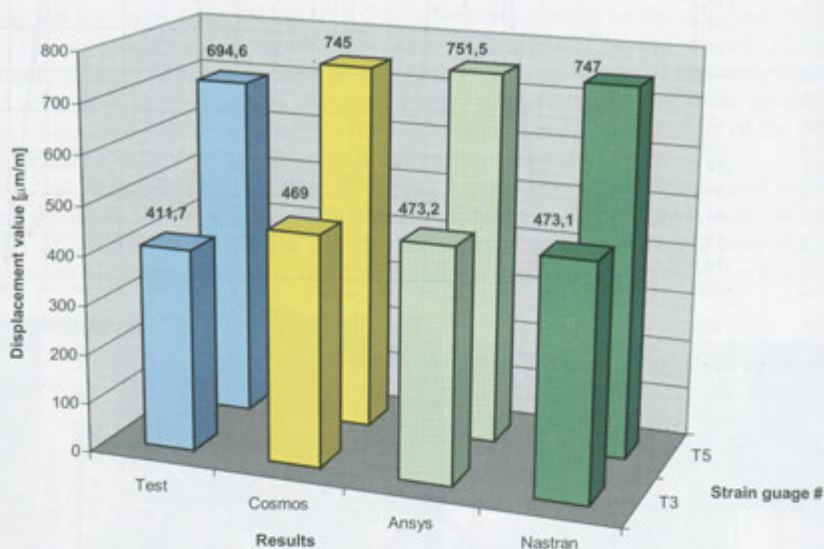


Fig.30. Time cycles of wheelset wheel deformation measured at the test stand

After the FEM numerical analysis, the wheel deformation has been determined – Fig. 31b. The lowest difference in the result as relates to experimental investigation has been generated in case of manual hexagonal mesh. Figure 31 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)

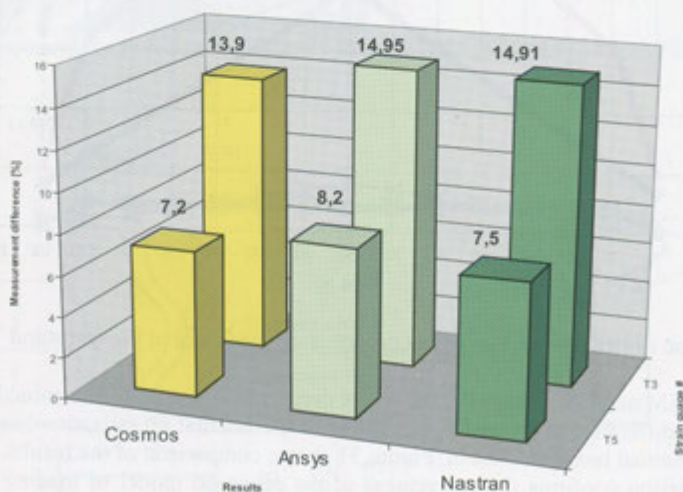


Fig. 31. Comparison of results of numerical analysis and experimental tests; load is due to carriage weight; a) deformation, b) results discrepancy

4.3. Experimental thermal investigation of a German wheel

A German wheel has been investigated (Technical Specification No. FWG 302.0.02.001.007); 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.32). The investigation has been aimed at the temperature measurement in the railway wheel during braking.

Test stand scheme is shown in Fig.33. 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 90° along the circumference, and 9 mm away from the wheel's rolling surface – Fig.34.

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

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 °C. 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. 50° 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).

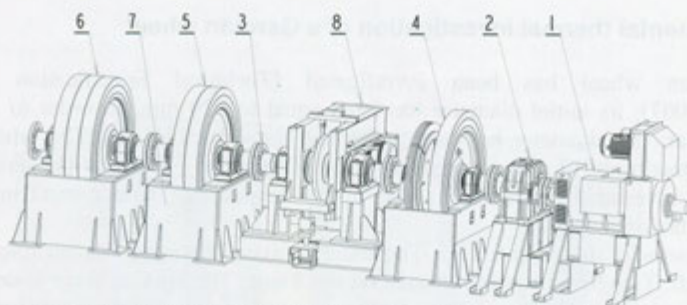


Fig. 33. Test stand for investigation of deformation and temperature of railway wheelsets wheels

- 1 Electric motor ($P = 265 \text{ kW}$, $P_{\max} = 400 \text{ kW}$, $n = 3200 \text{ rpm}$)
- 2 Toothed gear ($i = 1,5$, resp. $i = 1,72$ or $i = 4$)
- 3 Coupling
- 4 Flywheel $400 \text{ kgm}^2 = 280 + 120 \text{ kgm}^2$ ($2 \times 5 + 3 \times 10 + 4 \times 15$)
- 5 Flywheel 600 kgm^2
- 6 Flywheel 900 kgm^2
- 7 Stand casing
- 8 Wheel with railway shoe brake

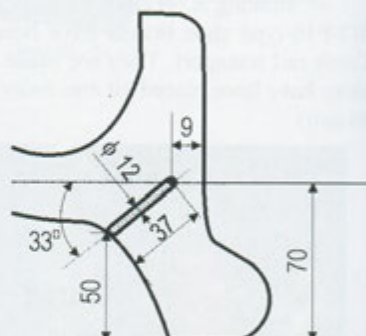


Fig. 34. Placing of thermocouples along the wheel circumference (9 mm away from the rolling surface)

The numerical thermal analysis has been conducted according to UIC Report [130], 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. 35) 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 %.

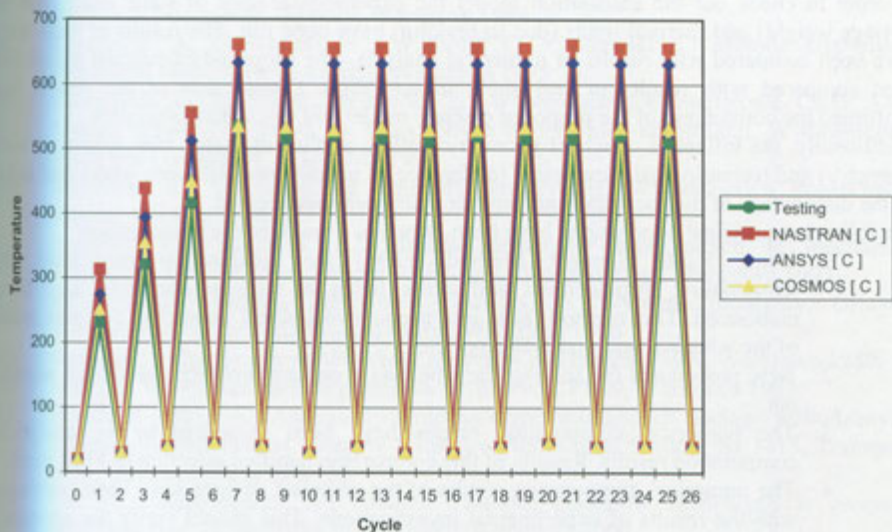


Fig.35. Temperature diagram for nodes corresponding to thermocouples placing

5. Remarks and conclusions

To summarise: presented work recapitulates the existing state-of-the-art related to railway wheelsets. Several basic direction of research on railway wheelsets are indicated:

- durability and reliability,
- design,
- load analysis (loads due to manufacture and service, and in particular to braking processes),
- unconventional materials,
- certification and commissioning tests,
- application of FEM in design,
- accuracy of numerical and experimental methods.

The analysis has confirmed the advisability of conducting planned research aimed at systemising the model generation and calculation by numerical procedures for railway wheelsets wheels.

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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