

## MECHANICAL STRENGTH TEST OF TRAIN WHEEL BASED ON VIRTUAL PROTOTYPING

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**Abstract:** *Virtual prototyping becomes more and more a less-expensive alternative of physical mockups. Its applicability to product certification procedures is already included in several standards where it is possible to replace completely physical prototyping and testing. A sample is given based on the standard for assessment of mechanical behavior of railway applications – wheel sets and bogies – EN 13979-1:2003+A1:2009. This study includes developed assessment approach for wheel mechanical strength check using engineering analyses, based on virtual prototyping technology. The approach contains 6 steps that allow user to determine required by the standard dynamic stress maximal values and to compare them against permissible ranges stated in the standard. Developed approach is demonstrated by an example of monoblock wheel for train vehicle up to 50t load capacity. This example shows an effective way to reduce expenses and obtain a cost-effective solution as well as to shorten time to reach market. Additional advantage is design evaluation at early stage of product life cycle and possibility to explore structural behavior in detail as to improve product performance. Virtual prototyping application combines overall product performance optimization, time and expenses reduction and detailed diagnosis of ongoing physical phenomena.*

**Key words:** *virtual prototyping, engineering analyses, train wheel, FEA.*

### 1. INTRODUCTION

The genesis of Virtual Prototyping (VP) is preconditioned by the idea to replace physical mock-ups by software prototypes [2]. This is an aspect of information technology that facilitates communication between different engineering disciplines during the early design process, and also provides abilities to assess design-under-development at very early stage. One of the most popular definition of VP is as "A computer-based simulation of a system or subsystem with a degree of functional realism comparable to a physical prototype" and virtual prototyping as "The process of using a virtual prototype, in lieu of a physical prototype, for test and evaluation of specific characteristics of a candidate design" [1, 4]. In general, VP prototyping can replace the expensive physical mockups constructed to test designs.

Traditional design process already consists of several steps, where mechanical engineers produce models, analyze their behaviors under operating conditions, and pass physical prototypes "over the wall" for test engineers to evaluate in a pass/fail mode. Next step is to eliminate physical prototyping and testing where is possible.

Certain standards already permit to skip physical testing and go for certification based on virtual prototypes only. Such a standard is connected to assessment of mechanical behavior of railway applications – wheel sets and bogies – EN 13979-1:2003+A1:2009.

A railway wheel, together with an axle, is one of the crucial parts that support the safe operation of railway vehicles. Wheels support the entire weight of cars; however, they cannot be designed as a failsafe structure where a backup system by other parts can be applied in case of a serious problem. Therefore, absolutely high reliability is demanded in terms of strength. Accordingly, the most important and fundamental characteristic in designing wheels is strength.

This assessment may comprise two stages. The second stage is carried out depending on the results of the first stage. The purpose of this assessment is to ensure that there will be no risk of fatigue cracking either in the wheel web or in its connections with the hub or the rim during the service life of the wheel.

This study presents an approach for completing first stage of the assessment – mechanical strength test, based on virtual prototyping.

### 2. ASSESSMENT APPROACH

General concept of mechanical assessment, including second stage of physical testing, is defined by EN 13979-1:2003+A1:2009. The flowchart for this assessment is shown on Fig. 1.

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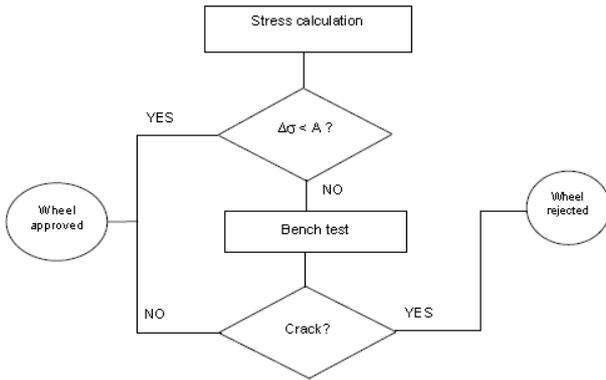


Fig. 1. Flowchart of the mechanical behavior assessment.

Major point here is the evaluation of the amplitude of the calculated stress  $\Delta\sigma$  against the permissible limit of dynamic stress  $A$ . Further, stress calculation is needed to be developed, based on virtual prototype application for engineering analyses that is not clearly defined in the standard. This approach comprises of the next major steps:

- **Step 1: Geometry model development:** Usually, design process includes development of 3D model for the purposes of technical documentation, but this model is not organized properly to engineering analyses requirements. The wheel consists of four major parts: a hub, a web, a rim and a flange. Three of them need to correspond to standard dimensions, and only the web varies for different manufacturers. Thus, the web is in the focus of this assessment and the model needs to be separated in three components, as just the middle – colored in red on the Fig. 2 – will be examined in detail. Another important feature is the axis symmetry of the wheel. It allows modeling just a half of it.
- **Step 2: Mesh model generation. Material properties:** The mesh for the wheel must ensure a good correlation between the calculated nominal permissible stresses and their measured equivalents.

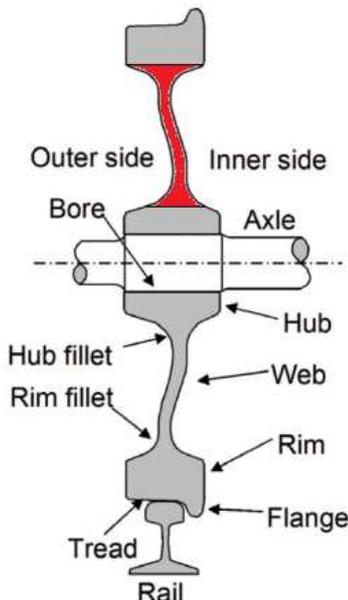


Fig. 2. Designations of each part of a solid wheel [5].

Used types of elements should allow detailed geometry reproduction, with dominant type having mid side nodes. The mesh should be hexagonal, structured, nevertheless of geometry curvatures. It should correspond to quality requirements (aspect and Jacobian ratio, warping, skewness, etc.).

- **Step 3: Boundary conditions:** Constraints are applied on the internal hub surface, according to the scheme on Fig. 3. Loads correspond to three load cases:

- Case 1: straight track (centered wheel set);
- Case 2: curve (flange pressed against the rail);
- Case 3: negotiations of points and crossings (inside surface of flange applied to the rail).

In fact, they differ by forces placement and values. The load vectors positions are shown on the Fig. 3.

Examined wheel is designated for overall load capacity of the vehicle up to  $Q_v$ . Thus, the load applied on single wheel  $P$  (for two bogies with two wheel sets each) is defined as:

$$P = \frac{Q_v \cdot g}{2 \cdot 2 \cdot 2} \tag{1}$$

Next, applied loads values are calculated according to the standard as follows:

- Case 1: straight track (centered wheel set):  
 $F_z(1) = 1.25P$ ;  
 $F_y(1) = 0$ ;
- Case 2: curve (flange pressed against the rail):  
 $F_z(2) = 1.25P$ ;  
 $F_y(2) = 0.7P$  (for guiding wheel set as worst case);

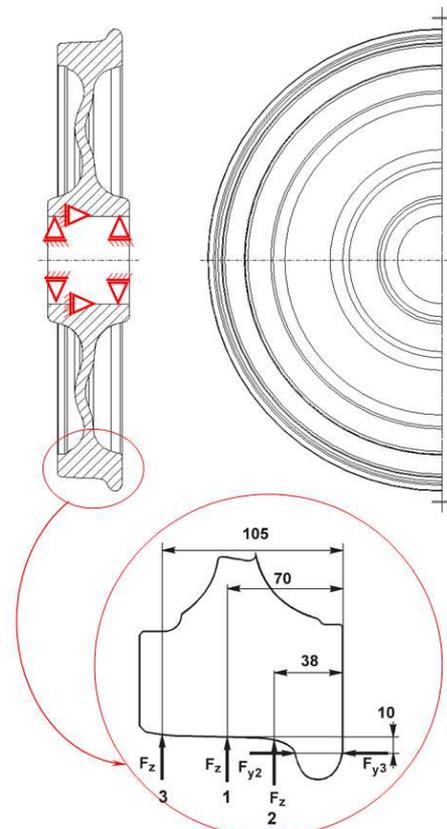


Fig. 3. Boundary conditions.

- Case 3: negotiations of points and crossings (inside surface of flange applied to the rail):  
 $F_z(2) = 1.25P$ ;  
 $F_y(2) = 0.42P$  (for guiding wheel set as worst case).
- **Step 4: Simulation results:** Results for all load cases are to be presented in means of distributions of:
  - principal stresses;
  - equivalent (von Mises) stresses.
 Major target is to review loaded structure and to determine maximal values.
- **Step 5: Dynamic stress calculation:** This step consists of three substeps, defined by the standard as follows:
  - Substep 5.1: Assessment, for each node, of the maximum principal stress for the three load cases ( $\sigma_{\max}$ ) and of the direction of this principal stress;
  - Substep 5.2: Assessment, for each node, of the minimum stress equal to the lowest normal stress in the direction of  $\sigma_{\max}$ , for the three load cases ( $\sigma_{\min}$ );
  - Substep 5.3: Calculation for each node of requested dynamic stress value:

$$\Delta\sigma = \sigma_{\max} - \sigma_{\min}.$$

- **Step 6: Design evaluation:** The range of dynamic stress  $\Delta\sigma$  shall be less than the permissible stresses at all points of the web. The permissible ranges of dynamic stresses,  $A$ , are as follows:
  - for wheels with a machined web:  $A = 360 \text{ N/mm}^2$ ;
  - for wheels with a non-machined web:  $A = 290 \text{ N/mm}^2$ .

### 3. ASSESMENT OF SAMPLE WHEEL

Proposed assessment is demonstrated through an example for a train wheel design for a vehicle having max capacity of 50 t. Developed approach is applied step by step below.

#### 3.1. Step 1: Geometry model development

Geometry model is built according to supplied technical documentation for its manufacturing. It includes all details of the examined design including relatively small rounds and other features. Due to the assessment approach specifics, built geometry model is separated in three zones, as it is shown on Fig. 4. Another specific feature is that all three volumes are connected by common surfaces. Thus, they act as a single body and are treated just like separate regions of it.

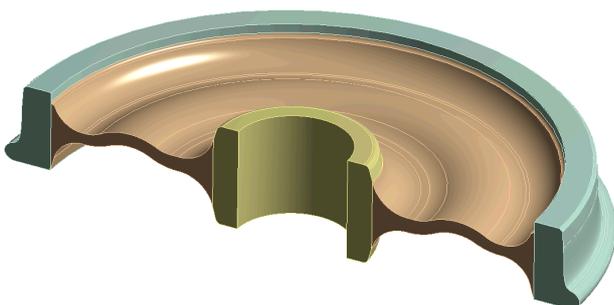


Fig. 4. Geometry model, used for engineering analyses.

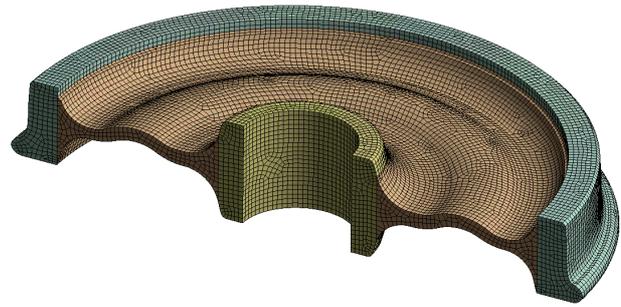


Fig. 5. Generated mesh model.

#### 3.2. Step 2: Mesh model generation. Material properties

Next step is connected to building a mesh model, using already developed geometry model. Hexagonal elements are used as to correspond to requirements for mesh quality. General view of the mesh is shown on Fig. 5. Material properties are set for standard wheel steel as defined by EN 13262, grade ER7.

#### 3.3. Step 3: Boundary conditions

Boundary conditions are applied according to specifications for step 3. Three separate analyses are formed, each presenting straight and curved track (Case 1 and Case 2) as well as negotiations of points and crossings (Case 3) as are defined by the standard.

Load applied on single wheel  $P = 65\,459.4 \text{ N}$ . Determined load values are as follows:

- Case 1:  
 $F_z(1) = 81824.2 \text{ N}$ ;  
 $F_y(1) = 0$ ;
- Case 2:  
 $F_z(2) = 81824.2 \text{ N}$ ;  
 $F_y(2) = 45821.6 \text{ N}$  (for guiding wheel set as worst case);
- Case 3:  
 $F_z(2) = 81824.2 \text{ N}$ ;  
 $F_y(2) = 27492.9 \text{ N}$  (for guiding wheel set as worst case).

A sample for the applied boundary conditions for Case 1 is shown on Fig. 6.

#### 3.4. Step 4: Simulation results

Major simulation data are shown by principal stress distributions and vectors – just for the web. A sample is shown for load case 1 on Figs. 7–10.

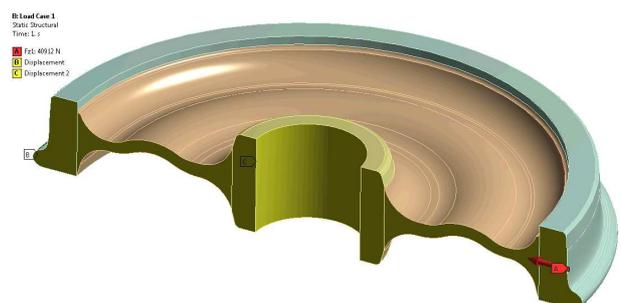


Fig. 6. Applied boundary conditions for Case 1.

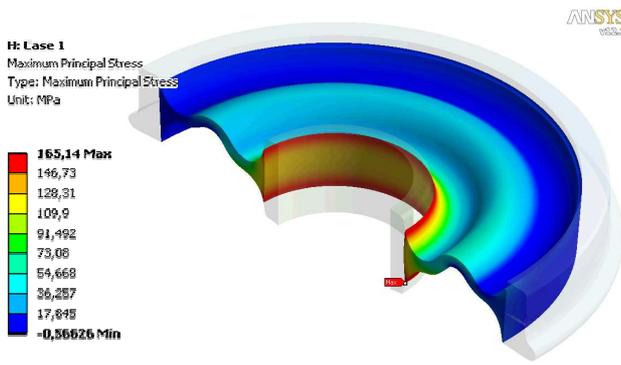


Fig. 7. Case 1. Maximum principal stresses,  $\sigma_1$ .

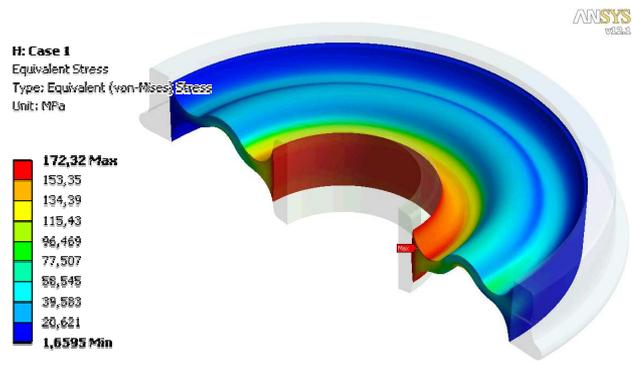


Fig. 11. Case 1. Equivalent stresses.

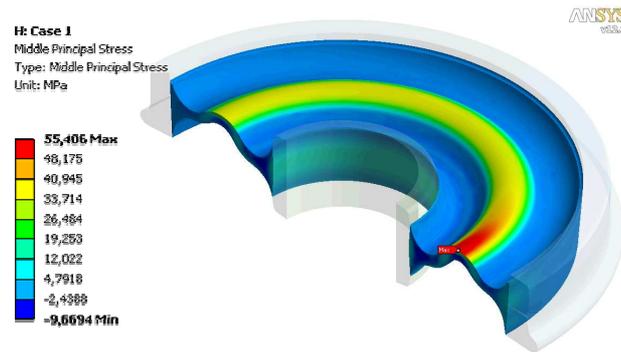


Fig. 8. Case 1. Middle principal stresses,  $\sigma_2$ .

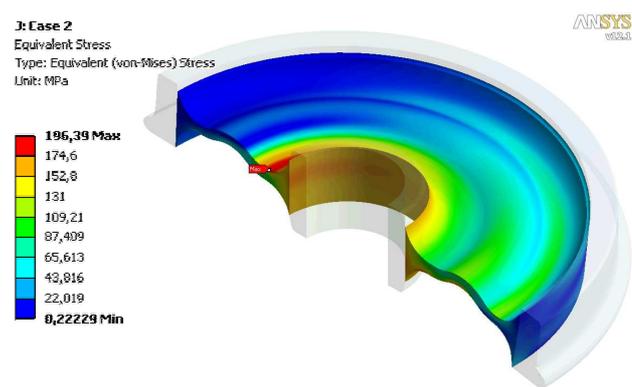


Fig. 12. Case 2. Equivalent stresses.

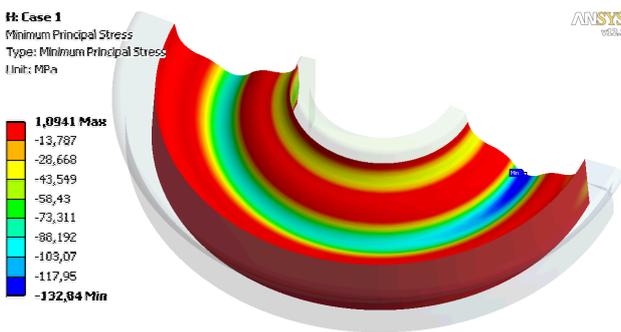


Fig. 9. Case 1. Minimum principal stresses,  $\sigma_3$ .

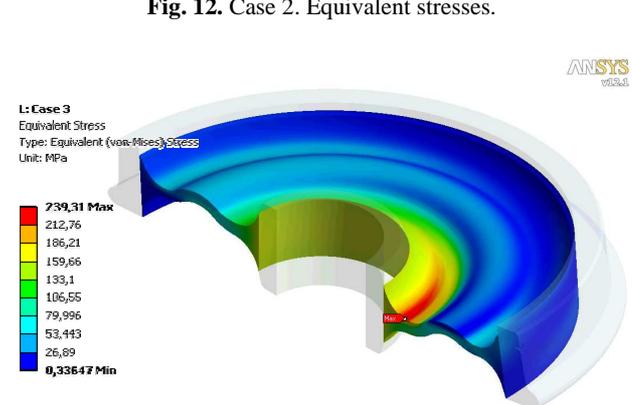


Fig. 13. Case 3. Equivalent stresses.

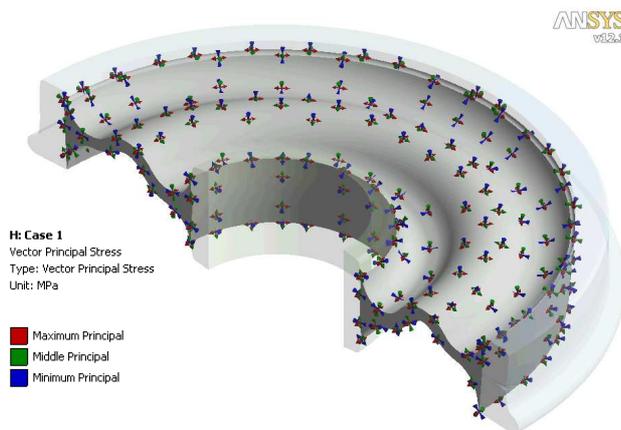


Fig. 10. Case 1. Principal stresses vectors directions.

Additionally, the equivalent (von Mises) stress distributions are shown for each load case, to be compared to material limits. They are shown graphically on Figs. 11–13.

Maximal values of equivalent (von Mises) stresses by load cases are as follows:

- Case 1: 172 MPa;
- Case 2: 196 MPa;
- Case 3: 239 MPa.

Briefly, the most loaded case is Case 3 (negotiations of points and crossings) with nearly 240 MPa equivalent stress. This is predefined by the standardized loads applied at maximal distance to the central membrane. The most loaded zone is the round near the hub. Further optimization of geometry is possible as to reduce stresses if needed.

### 3.5. Step 5: Dynamic stress calculation

Assessment of all nodes in every load case is performed to extract maximum principal stress value for each load case, its location (node) and vector direction. These data are stored in Table 1.

Table 1

Maximum values by load cases and principal stresses.  
Maximum principal stress location and vector direction  
(All stress units are in MPa)

Parameter \ Cases		Case 1	Case 2	Case 3
$\sigma_1$	max	165.1	163.4	166.5
	min	0.6	-3.9	-2.4
$\sigma_2$	max	55.4	99.7	116.3
	min	-8.6	-10.1	-12.4
$\sigma_3$	max	1.4	1.8	2.4
	min	-132.8	-190.6	-231.7
Max stress value		$\sigma_{\max} = \sigma_{1(\max)} = 165.1$	$\sigma_{\max} = \sigma_{3(\min)} = -190.6$	$\sigma_{\max} = \sigma_{3(\min)} = -231.7$
Node #		60311	39543	60491
Direction vector	X	-0.00007	-0.27020	0.13269
	Y	0.00041	0.96165	0.99116
	Z	1.00000	-0.04704	0.00009

Selected nodes with maximum principal stress values are used to obtain normal stresses in these directions for the three load cases. Thus, three normal stresses are obtained and minimum value of each is selected. This minimum value is combined with the maximum values to obtain the dynamic stress  $\Delta\sigma = \sigma_{\max} - \sigma_{\min}$ .

Three solution combinations are examined:

- Combination A:  $\sigma_{1(\max)}_{\text{Case 1}} - \sigma_{\text{normal}(\min)}_{\text{Case 2}}$ , based on node 60311 case, shown on Fig. 14;
- Combination B:  $\sigma_{3(\max)}_{\text{Case 2}} - \sigma_{\text{normal}(\min)}_{\text{Case 3}}$ , based on node 39543 case, shown on Fig. 15;
- Combination C:  $\sigma_{1(\max)}_{\text{Case 3}} - \sigma_{\text{normal}(\min)}_{\text{Case 2}}$ , based on node 60491 case, shown on Fig. 16.

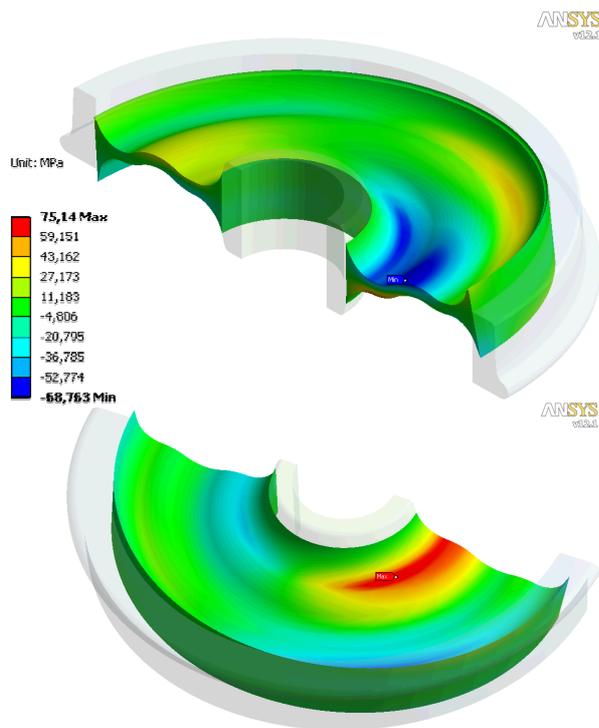


Fig. 14. Solution combination A of principal stresses – distribution fields on both sides of the web, Mpa.

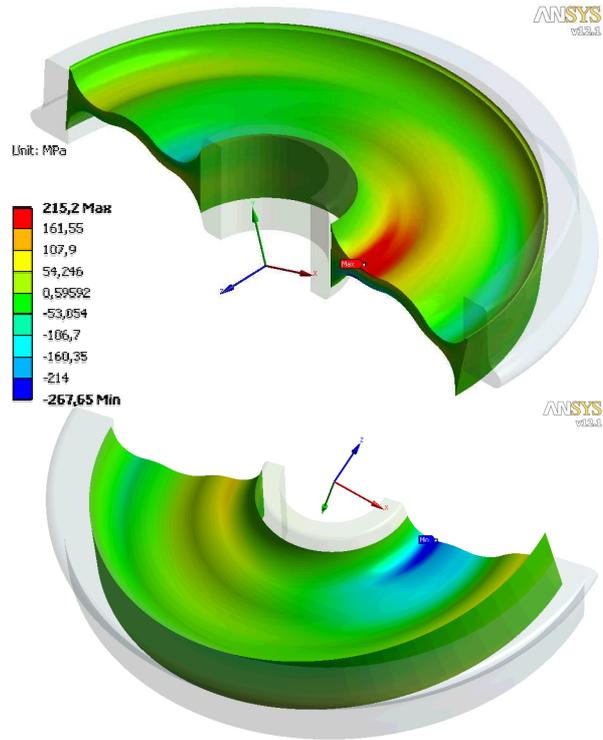


Fig. 15. Solution combination B of principal stresses – distribution fields on both sides of the web, Mpa.

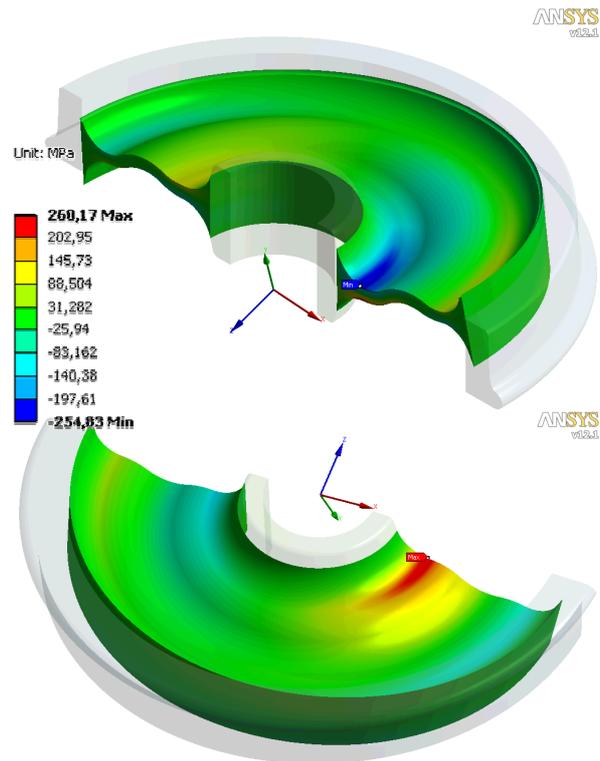


Fig. 16. Solution combination C of principal stresses – distribution fields on both sides of the web, Mpa.

3.6. Step 6: Design evaluation

Design evaluation is based on simulation results of the web of wheel and could be formed as follows:

- Maximum von Mises equivalent stress is 239 MPa are less than permissible steel limit of elasticity 355 MPa;

- Maximum dynamic stress range  $\Delta\sigma$  is 268 MPa that is lower than permissible value  $A = 290$  MPa for non-machined web;
- Obtained results correspond to the decision criteria in EN 13979-1: 2003 (E) standard, First stage – Calculation.
- The wheel could be approved and certified without subsequent benchmark over physical prototype included in the second stage of EN 13979-1: 2003 (E) standard.

#### 4. CONCLUSIONS

General conclusions could be stated that the developed approach is successfully demonstrated through an example of monoblock wheel for train vehicle up to 50 t load capacity. It completely eliminates the necessity of physical prototyping and testing and leads to the next major advantages:

- Product design is evaluated at its most early life cycle stage;
- Product structural behavior is examined in detail and possibilities to improve its performance are marked (critical zones are found that would help further engineering process);
- Development expenses are reduced by certification process based entirely on virtual prototyping and engineering analyses;
- Time to reach market is decreased significantly (normal certification process by physical tests takes about 1 month).

Virtual prototyping application combines overall product performance optimization, time and expenses reduction and detailed diagnosis of ongoing physical

phenomena. Important step, of course, is connected to the fact that this standard procedure allows to rely on virtual prototyping results that have results with sufficient accuracy for the target application to be obtained.

Additionally, the standard allows assessing acoustic behavior of the wheel as well, based again over virtual prototyping techniques. Thus, the entire certification process of the product is based on virtual prototyping, leading to above mentioned advantages and contributing for increased dynamics of the contemporary new product development process.

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