Shape optimization methodology for railway wheels subjected to thermo-mechanical loads

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Abstract: In railway industry a reduction of the wheel mass is at present required so as to achieve a higher energetic efficiency and a decrease of high-frequency vibrations. Wheel warming during the braking process is responsible for high thermal stresses which prove to be more relevant than stresses caused by wheel-rail contact loads. In this work an axisymmetrical finite element model is used to develop an optimization process of the wheel shape. This one is controlled by design variables defined by means of displacement vectors applied to nodes in the wheel web. The objective function is the minimization of mass subjected to constraints of maximum stress and maximum rim lateral displacement. The optimization process is iterative, and requires a thermo-mechanical transient analysis in Abaqus for each iteration to simulate the braking. At the end of each iteration, the values for the constraints and the objective function are read, and updated values are assumed for the design variables with the subsequent geometry modifications. The resultant model is then verified by performing a three-dimensional analysis in Abaqus in which rail contact loads are taken into account. Non-linear material behavior and temperature dependent material properties are considered in the analysis. As a result, a 20% mass reduction is obtained for the wheel only by acting in the wheel web. A stricter convergence criterion, a higher number of design variables and a wider variability for these ones provide better results.

Keywords: Brakes, Design Optimization, Coupled thermal stress analysis, Minimum-Weight Structures, Optimization, Railway industry, Shape Optimization, Railway wheel.

1. Introduction

Railway industry is experiencing a remarkable expansion, which is largely motivated by the continued implantation of high speed trains, tramcars and underground. An increased competitiveness is the consequence in railway market, and in fact local laws are now allowing the privatization of companies operating in the rail network to stimulate this competitiveness.

In this context, as any decrease of the involved costs is desirable, the optimization of the wheel shape and weight clearly means a higher energetic efficiency. Moreover, high speed development has also implied more demanding vibration conditions. It is thus foreseen that a reduction of the wheel mass will lead to a decrease of high-frequency vibrations.

Two different optimization methods could be used to make wheels lighter: topology optimization and shape optimization. The first one, although it is a non-conditioned process, is not easy to implement in this kind of analysis: HyperWorks topology optimizer (OptiStruct) is not compatible with heat transfer analysis and temperature-dependent material properties. On the other hand, shape optimization provides a determined geometric solution, with no need of delineation.

The optimization process applied to railway wheels has been analyzed in several works (Nielsen, 2006). On the other hand, the adaptive surface method has also been considered in design problems by some authors (Wang, 2003).

In this paper a specific methodology is presented to optimize the wheel mass by keeping functional and resistance features of the wheel. Particularly, wheel mass will be the objective function, and lateral displacement (functional feature) and stress (resistance feature) the variables on which the constraints of the optimization process are to be imposed.

2. Design conditions

Figure 1 shows the different parts in a three-dimensional wheel model. There are several design conditions to be taken into account in the optimization process. The first one is that only the wheel web can be modified (see Figure 2). On the other hand, the rim shape is composed of seven different circumference arcs and has been analyzed here in detail. Moreover, the dimensions of the hub region are determined by the joining process between the wheel and the axle.

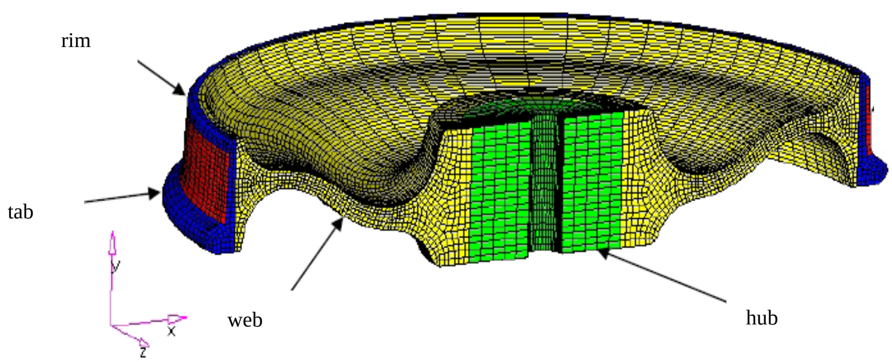

Figure 1. Wheel regions in a three-dimensional model.

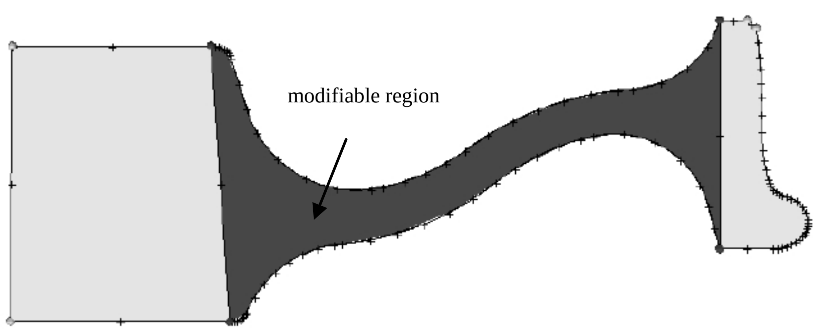

Figure 2. Design region in an axysimmetric model.

The software used in the model is Simulia Abaqus/CAE and Altair HyperWorks. Abaqus CAE is used as the solver of the optimization, and also in the preprocessing. HyperStudy and HyperMesh modules of HyperWorks are used respectively as optimizer and to define the design variables by the appropriate "morphing" tool.

American standard (AAR Standard S660, 2004) or European standard (EN 13979-1, 2003) for Railway applications have to be taken into account, depending on the destination of the wheel. In this work American standard is applied.

3. Analysis

3.1 FE model

Wheels must resist different load cases specified in the Standard for each geometry (new and old). The most critical situation for the previous wheel models corresponds to the old geometry subjected to a V2+Th load case. Th means a constant thermal load applied for twenty minutes onto the rim (an axisymmetric load in consequence). V2 is a mechanical load applied at a certain position of the wheel, so it is not axisymmetric and a three-dimensional model is required for the analysis. Application regions of these loads are shown in Figure 3.

The FE analysis of previous wheel models shows that approximately 90% of the stress is due only to the thermal load. For this reason and considering the much lower computational cost of an axisymmetric model, it is proposed here the following solution: an axisymmetric FE model of the old wheel subjected to a Th load case is used in the optimization process; then, a three dimensional FE model subjected to a Th+V2 load case is used to verify the validity of the best obtained optimized models.

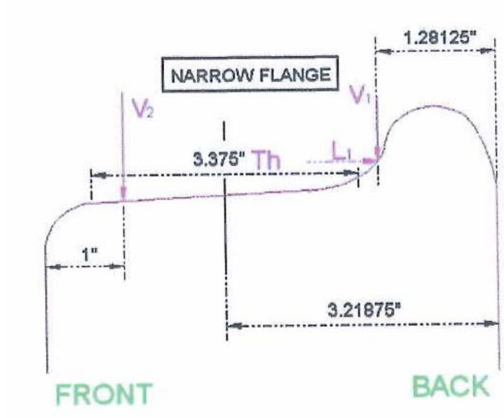

Figure 3. Application points and regions of loads (American Standard).

A transient fully-coupled thermal stress analysis is carried out in the axisymmetric model. CAX6MT and CAX8T second order elements are used. This analysis provides temperature and stress distribution at the same time. In the fully-coupled analysis, it is assumed that temperature has an effect on stress, and stress on temperature. A sequentially coupled analysis has been rejected in part for the fact that by using a single model simplifies the process.

Input loads are the thermal load simulating the braking process for twenty minutes (DFLUX), the convection in the rim and web (FILM), and the constraints of fixed displacements of the hub end.

The analysis is carried out in only one step with the following specifications:

Analysis type: HEAT TRANSFER

Initial Conditions: Initial temperature

Load types: DFLUX, FILM

Element types: DC3D6, DC3D8

The used model is shown in Figure 4. Non-linear material behavior and temperature dependent material properties are considered in the analysis.

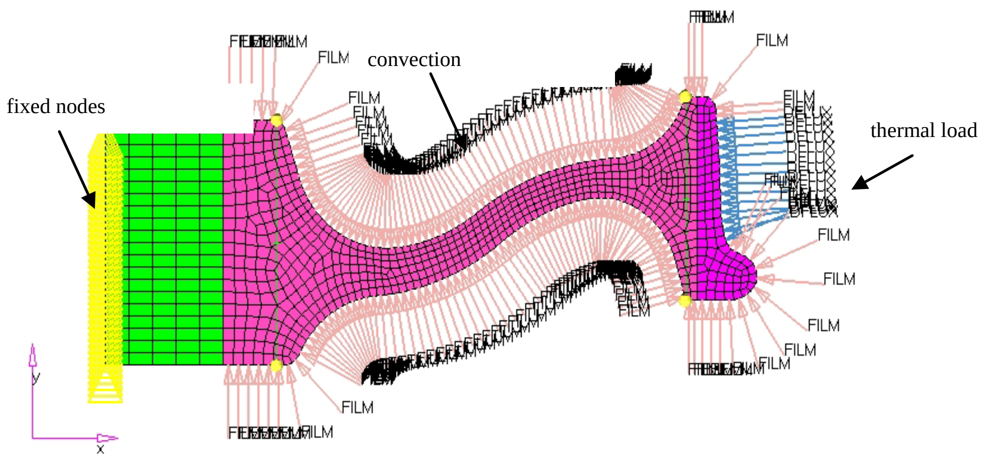

Figure 4. Axisymmetric model for fully coupled thermal stress analysis (American Standard).

3.2 Design variables definition

A design variable is a specification that is controllable by the designer during the optimization process. Each design variable has been defined as an amount of vectors that modify the position of the mesh nodes. When a design variable takes a value of one, each node displaces as specified by the designer. A value of 0.5 implies a node displacement of only 50% of the defined quantity.

Six different design variables have been used. The first four variables (Figures 5-8) are intended to eliminate mass from the fillets, and the other two (Figures 9-10) to the same purpose by moving down the upper contour or by moving up the lower one.

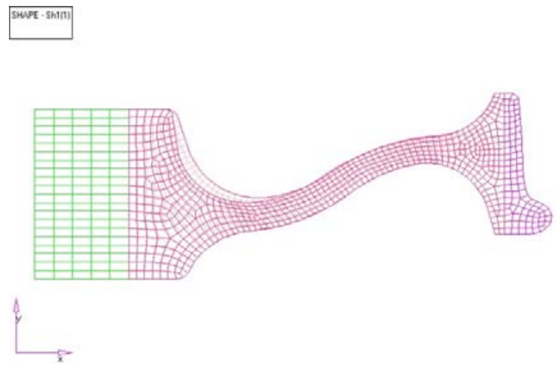

Figure 5. Design variable 1 (DV1).

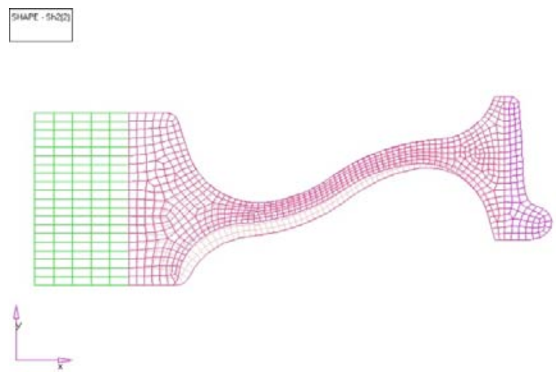

Figure 6. Design variable 2 (DV2).

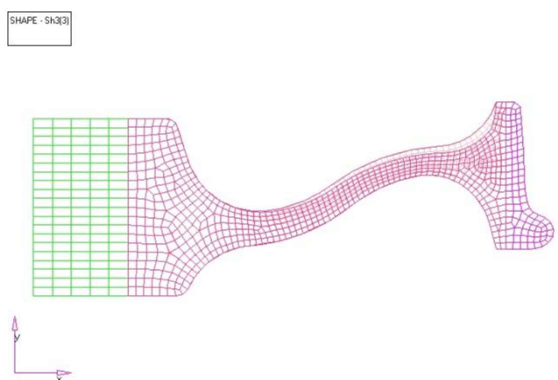

Figure 7. Design variable 3 (DV3).

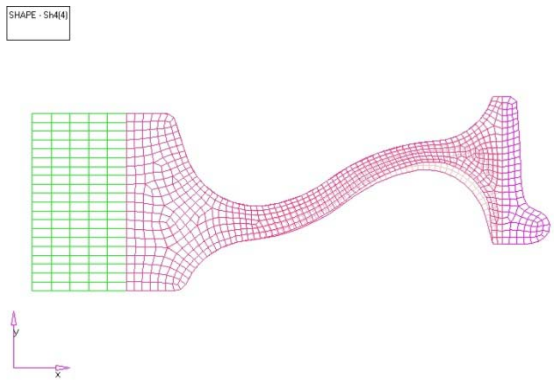

Figure 8. Design variable 4 (DV4).

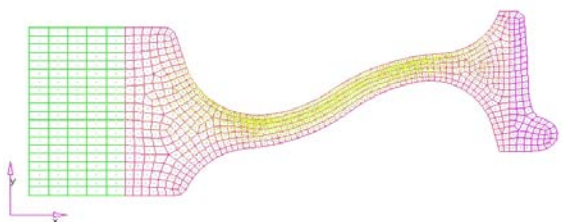

Figure 9. Design variable 5 (DV5)

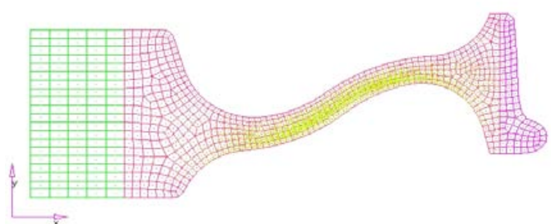

Figure 10. Design variable 6 (DV6)

In the definition process of the design variables the corresponding "morphing" tool of the HyperMesh module has been used.

3.3 Optimization programming

Optimization is programmed by means of the HyperStudy module. Three steps can be distinguished in this process: the optimization definition, the optimization running and the results output.

Firstly, the previously defined design variables are to be recognized and their upper and lower limits as well as their initial values are to be determined by the user in the HyperMesh module. Then, the initial model analysis is carried out (*nominal run*), and the required output throughout the optimization process are read, as for example: the maximum Von Mises effective stress in the wheel web and the mass of the model. The aim of this first analysis is to become aware of the starting model characteristics and to help defining responses. The objective function and the constraints are determined from these responses (Table 1). In our case, the objective function is to minimize the mass of the model, and it is subjected to the constraint of a maximum stress value in the web nodes (Table 2).

Table 1. Optimization Responses.

Response ID	Responses
Response 1	Mass
Response 2	Maximum VM stress in web

Table 2. Optimization variables.

Optimization Variables	
Objective Function	Minimize Response 1
Constraint	Response 2 ≤ Upper limit = 90 ksi

As Abaqus is the external solver, a script with the path of the solver should be introduced. When the output file size is too large in an iteration, HyperStudy may try to start the next iteration before having read the responses. This would lead to an execution error. “Sleep” MS-DOS program can be used to avoid it. A model of script could be the next one:

```
C:\ABAQUS\6.7-1\exec\abq671 job=rueda memory=200Mb
C:\sleep 250
```

A convergence criterium as well as a maximum number of iterations should also be determined by the user. Then, the optimization engine method should be selected by the user. Although there are some different methods, only the Adaptive Response Surface Method (ARSM) has been used.

The first iteration of the process is the nominal run. Throughout the next *n* iterations (where *n* is the number of design variables) an initial mapping is carried out by the optimization engine. In each iteration only one of the design variables changes its value and the process will stop according to the defined convergence criteria. After the initial mapping, different values are given to the different design variables, depending on their influence on the objective function and the constraint (Figure 11).

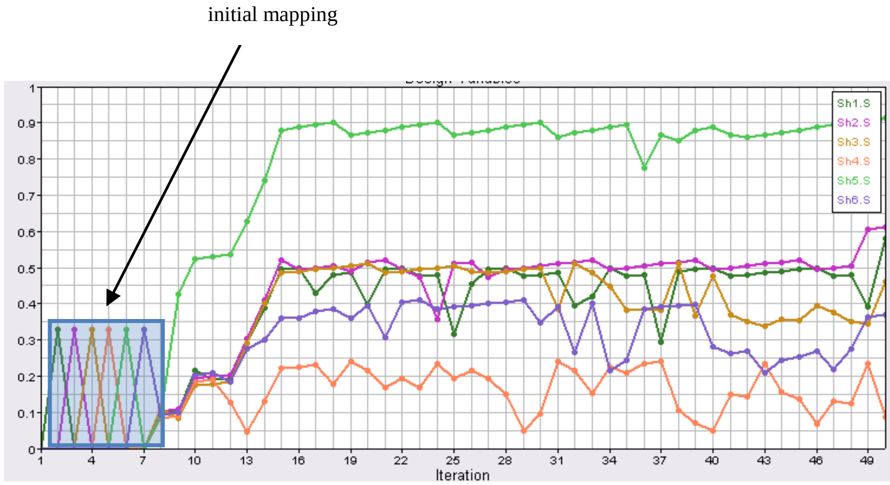

Figure 11. Design variable history throughout the optimization process.

Finally, a history of iterations is obtained: the mass value of the model (objective function) and the maximum stress in the wheel web (constraint). Then, the best model is selected as the solution of the problem (Figure 12).

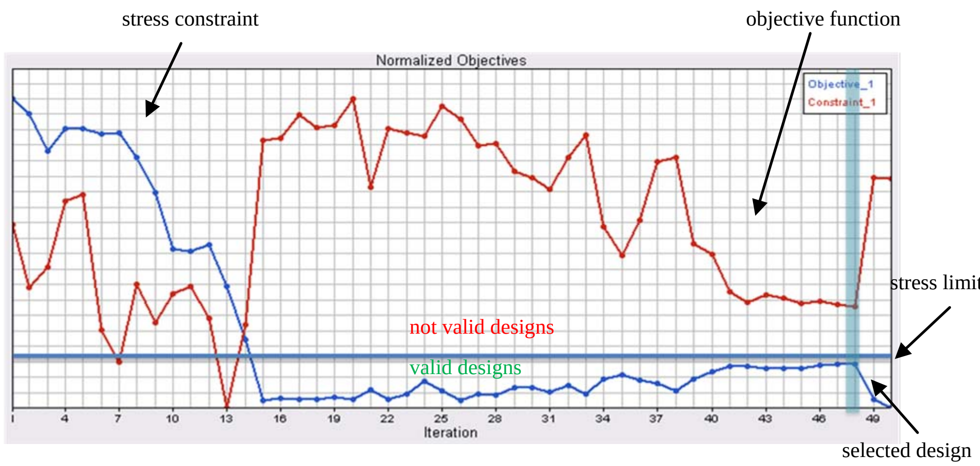

Figure 12. Best design selection.

3.4 Design verification

As mentioned above the V2 load corresponding to the most critical load case was not taken into account in the previous optimization process. Now a design verification of the proposed solution must be carried out by a three-dimensional FE model.

A sequentially-coupled thermal stress analysis is recommended. A first model is used to perform a transient HEAT TRANSFER analysis, in which the wheel is subjected only to the thermal load. The thermal field is obtained as an output and applied along with the V2 mechanical load in a second model by using a STATIC analysis to obtain the maximum stress in the wheel web (Figure 13).

For the first model DC3D6 and DC3D8 first order elements are used, and for the second one C3D15 and C3D20 second order elements. The MIDSIDE option is applied in the second model to interpolate temperature values for middle nodes that do not exist in the first model. The analysis conditions are summarized as:

- *First model (1 step)*

  Analysis type: HEAT TRANSFER

  Initial Conditions: Initial temperature

  Load types: DFLUX, FILM

  Element types: DC3D6, DC3D8

- *Second model (1 step)*

  Analysis type: STATIC

  Initial Conditions: Initial temperature, Hub nodes fixed

  Load types: CLOAD, TEMPERATURE

  Element types: C3D15, C3D20

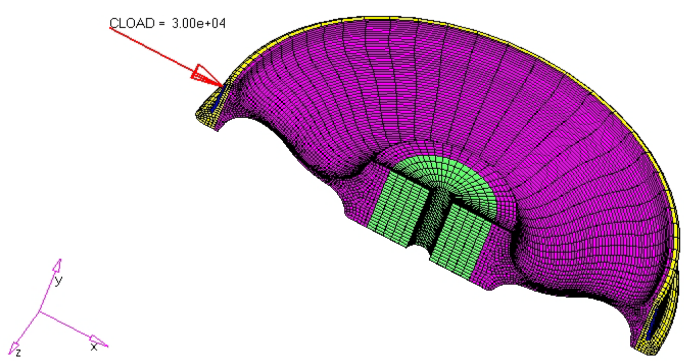

Figure 13. 3D static stress model (*second model*).

4. Results

A complete set of analyses have been carried out to test the methodology and improve the optimization programming. Along these analyses the upper limit of the maximum stress, the number of design variables, the design variable range, the convergence parameters or maximum number of iterations and some advanced mapping parameters have been modified.

The results for the starting model are the following:

- Maximum VM stress in wheel web (V2 + Th load case): 85 ksi
- Maximum temperature after braking: 647 F
- Model mass: 892 lb.

The optimization analyses leading to the best solutions are presented in Table 3.

Table 3. Best optimization tests.

MNI: Maximum number of iterations; AC: Absolute Convergence; RC: Relative Convergence; CV: Constraint violation.

Opt. ID	DV Variability	Stress limit (psi)	Parameters
1 (15)	DV1: (0,1) DV2: (0,1) DV3: (0,1) DV4: (0,1)	82120	MNI: 25 AC: 0.001 RC: 1% RV: 0.5%
2	DV1: (-0.5,1) DV2: (-0.5,1) DV3: (-0.5,1) DV4: (-0.5,1)	82150	MNI: 25 AC: 0.001 RC: 1% RV: 0.5%
3	DV1: (-0.5,1) DV2: (-0.5,1) DV3: (-0.5,1) DV4: (-0.5,1)	77400	MNI: 25 AC: 0.001 RC: 1% RV: 0.5%
4	DV1: (-1,1) DV2: (-1,1) DV3: (-1,1) DV4: (-1,1)	77400	MNI: 25 AC: 0.001 RC: 1% RV: 0.5%
5	DV1: (-1.5,1) DV2: (-1.5,1) DV3: (-1.5,1) DV4: (-1.5,1)	77400	MNI: 25 AC: 0.001 RC: 1% RV: 0.5%
6	DV1: (0,1) DV2: (0,1) DV3: (0,1) DV4: (0,1)	82120	MNI: 25 AC: 0 RC: 0% RV: 0.5%
7	DV1: (-1,1) DV2: (-1,1) DV3: (-1,1) DV4: (-1,1) DV5: (-1,1) DV6: (-1,1)	77400	MNI: 25 AC: 0.001 RC: 1% RV: 0.5%

The results for the best solutions are shown in Table 4.

Table 4. Optimization results.

Opt. ID	DV Final Values	Results for selected model	Wheel Mass Reduction
1	DV1: 1 DV2: 1 DV3: 0.54 DV4: 0.43	Number of it.: 19 Max. VM stress in web: 90 ksi Max. Temperature: 701 F Mass: 726 lb	18,61%
2	DV1: 0.27 DV2: 0.63 DV3: 0.3 DV4: 0.21	Number of it.: 15 Max. VM stress in web: 86 ksi Max. Temperature: 666 F Mass: 802 lb	10,08%

Opt. ID	DV Final Values	Results for selected model	Wheel Mass Reduction
3	DV1: 1 DV2: 0.45 DV3: 0.1 DV4: 0.1	Number of it.: 19 Max. VM stress in web: 81 ksi Max. Temperature: 661 F Mass: 816 lb	8,52%
4	DV1: 1 DV2: 0.37 DV3: 0.1 DV4: 0.08	Number of it.: 18 Max. VM stress in web: 81 ksi Max. Temperature: 659 F Mass: 824 lb	7,62%
5	DV1: 0.25 DV2: 1 DV3: -0.16 DV4: 0.58	Number of it.: 23 Max. VM stress in web: 82 ksi Max. Temperature: 677 F Mass: 772 lb	13,45%
6	DV1: 1 DV2: 1 DV3: 0.63 DV4: 0.34	Number of it.: 25 Max. VM stress in web: 90 ksi Max. Temperature: 702 F Mass: 725 lb	18,72%
7	DV1: 0.29 DV2: 0.31 DV3: 0.29 DV4: 0.05 DV5: 0.72 DV6: 0.29	Number of it.: 25 Max. VM stress in web: 86 ksi Max. Temperature: 684 F Mass: 778 lb	12,78%

The hub and rim are taken into account when the model mass is calculated. Percentages of mass reduction are thus more significant. As mentioned above only the wheel web geometry can be modified.

Final results are obtained from the three dimensional model, in which the V2 concentrated load is also taken into account. When the maximum stress in the wheel web is calculated, high values could appear due to the concentrated load. These values are rejected as they are caused by a stress concentration phenomenon.

The starting geometry is shown in Figures 14, 15 and 16 with the selected geometry for each optimization design.

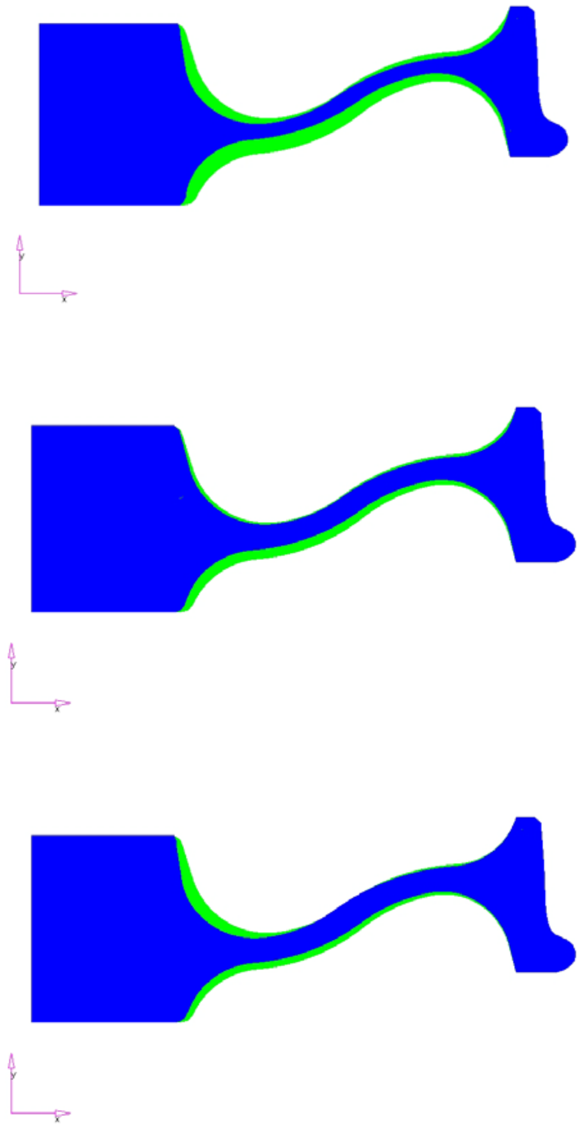

Figure 14. Designs 1, 2, 3.

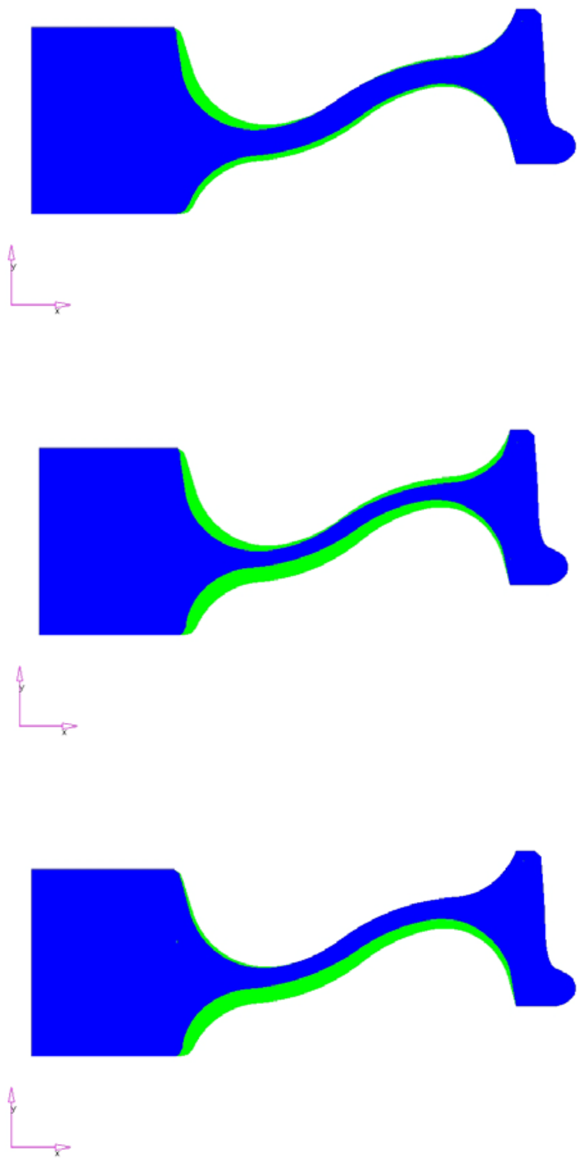

Figure 15. Designs 4, 5, 6.

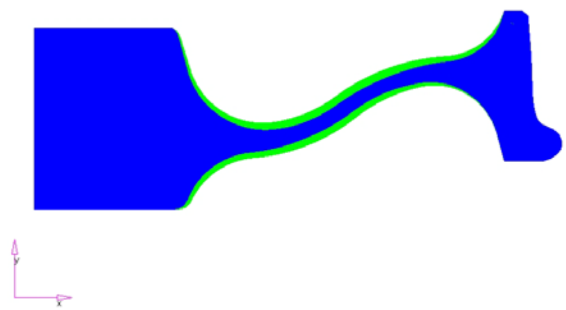

Figure 16. Design 7.

Von Mises stress and temperature distributions after braking for selected Design 1 are shown in Figures 17 and 18.

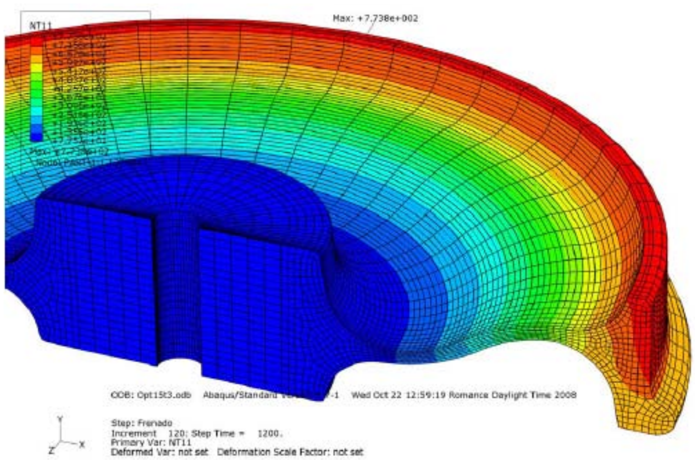

Figure 17. Temperature distribution in Design 1.

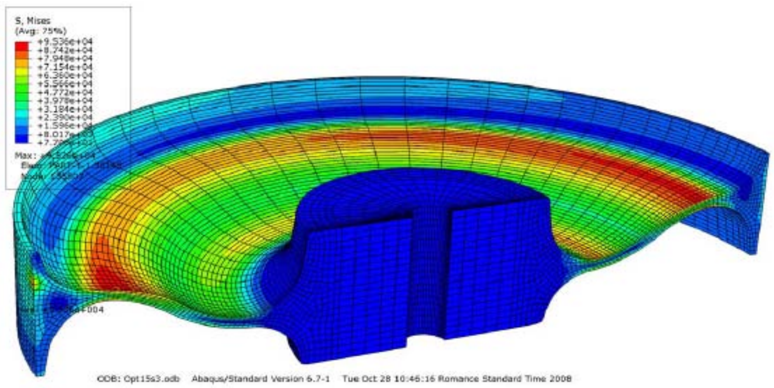

Figure 18. Von Mises stress distribution in Design 1.

5. Conclusions

An optimization methodology for railway wheels has been proposed. The wheel web geometry has been considered as the objective variable by carrying out first a fully-coupled thermal stress axisymmetric analysis and then a sequentially-coupled three-dimensional analysis. Simulia Abaqus/CAE capabilities as well as Altair HyperWorks software have been used.

The best solutions have been illustrated in the paper. It is concluded that a 20% mass reduction can be obtained by changing the web shape. It has also been proved that a higher number as well as a wider range of design variables lead to a better solution. A higher maximum number of iterations and stricter convergence criteria extend the process avoiding premature stops. Better models are obtained when negative values (a mass increase) are allowed for design variables, as this enables a higher mass decrease in other regions.

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