Stresses and strains in tyred wheels during tyre fitting process

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Abstract

Monobloc wheels are used today in almost all railway vehicles. Even if they give clear advantages for high-speed applications and where tread brakes are used, they are replacing tyred wheels also in urban and suburban trains.

Lengthy and expensive maintenance procedures wheels are largely responsible for dismissing tyred wheel. The theoretical advantage of using infinite life wheel centres and axles heavily contrasts with increasing labour costs required by mainly manual overhaul operations on tyre wheels.

As the authors are developing a strategy to revitalize tyred wheels in a large number of contexts, a design check of existing tyred wheels was compulsory. Unfortunately, no data were available from mainly manual calculations performed several decades ago. It was therefore necessary to do a sort of “reverse engineering” to understand how old existing tyred wheels work.

A specific step in the assembly of a tyred wheel is the shrink fit of the tyre, which has diametrically opposite requirements from the structural point of view. The paper tries to draw some general conclusions that the designer of tyred wheels should properly take into account when approaching the design of a new wheel.

The paper therefore shows the results of several simulations performed in order to analyse stressed and strains resulting from the tyre fitting process. A large number of parameters are evaluated to understand the behaviour of the current design of tyred wheels, with particular attention to the different shapes of the wheel centres. The ultimate aim of the research is to build a sound basis for a structural optimization of tyred wheels.

Keywords: tyred wheels, shrink fitting, Finite Element Analysis, structural strength.

1 Introduction

Even if tyred wheels have been the standard for all railway applications for more than 150 years, their success was obscured by the advent of monobloc wheels that have a number of advantages in terms of safety and maintenance and whose Life Cycle Cost is favourable compared to the standard manufacturing and maintenance process of tyred wheels. A survey about the maintenance procedures used for tyred wheels is described in a companion paper [1], together with a proposal to drastically simplify it.

Tyred wheels are made of three pieces that are assembled together: the wheel centre, the tyre and the retaining ring. The tyre, which is fitted on the wheel centre by a shrink fitting, present a lateral abutment (on the field side), which have to react the lateral force acting on the flange when the vehicle runs in a curve. In the opposite direction, the tyre is constrained to the wheel centre by the retaining ring that is assembled after the tyre fitting. With reference to the standard design of both monobloc wheels and tyred wheels, shown in Figure 1, some advantages related to monobloc wheels are evident.

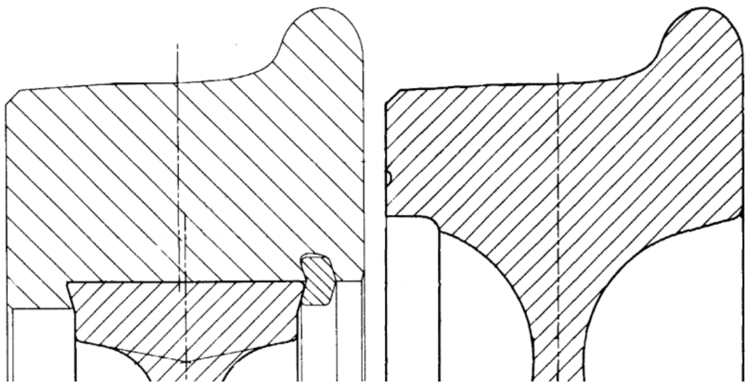

Figure 1. Drawings of a tyred (420 kg, left) and monobloc (340 kg, right) wheel with the same external diameter of 940 mm.

The ideal field of application of monobloc wheels are high-speed vehicle, as there are no negative effects of the centrifugal forces acting on tyres. Moreover, a very thick tyre is no more necessary for monobloc wheels, reducing therefore the unsprung mass.

It is interesting to highlight that the first Technical Specifications for Interoperability in Europe [2] were issued in 2002, and that the supporting EN standards ([3], [4]) described only monobloc wheels, leaving the “old” UIC standards undisturbed (a comprehensive list of applicable UIC leaflets is available in [1]). As tread brakes were responsible for a large number of wheel failures especially during drag braking, ranging from relative movements between the tyre and the wheel centre to tyre fractures, the following to monobloc wheels was linked to freight wagons.

Paper [5] discusses the effect of thermal inputs to tyred wheels, showing that tyred wheels are going to disappear from freight wagons by 2020. Moreover, in a not too far future also freight cars will be equipped with disc brakes and this completely removes the problem of thermal-induced failures starting on the wheel tread.

It is the authors' opinion that tyred wheels could continue to be an interesting alternative for conventional passenger trains (for speed up to 160 km /h) and disc-braked freight trains. Optimizing both the structural behaviour (mass reduction) and the manufacturing and maintenance process (time reduction), tyred wheels can generate large savings especially for modern vehicle architectures, such vehicles with inside bogie frame (inboard bearings wheelsets).

Considering the lack of scientific literature on the subject, the residual knowledge about tyred wheels is based on design parameters derived from the experience of a railway world that does not exist anymore. This paper shows how the design of "old fashioned" tyred wheels, although satisfactory, leaves ample margin for improvement. A number of existing wheels are simulated providing objective parameters that the designer can readily use to decide whether a new tyred wheel can compare to existing wheels in terms of mass and structural strength.

2 The tyre-wheel centre fitting model

2.1 Input data

Finite Element Analysis (FEA) is the tool used in this research to estimate stress and strains fields obtained on the wheel centre deriving from a) hub/axle shrink- or press-fitting and b) wheel centre/tyre shrink fitting.

The input loads are imposed by means of the interference values. All the interferences are calculated according to EN standards and UIC codes in force, considering their maximum and minimum values. Simulations were conducted given for four existing tyred wheels (It is worth to highlight that the *wheel 2* is designed to be equipped with brake blocks, resulting in a lower wear limit, while *wheel 4* has the highest wear limit value, which is the maintained from the original monobloc design. In fact, it is known that the wear limit for tyred wheels is always lower than the one set for monobloc wheels, as the tyre-wheel centre pressure has to be guaranteed also in worn conditions.

Table 1 shows the used load and geometry parameters):

1. a D = 940 mm wheel for passenger trains (axleload 13 t/axle) with v≤200 km/h;
2. a D = 920 mm wheel for freight trains (axleload 20 t/axle) v≤120 km/h;
3. a D = 840 mm wheel for metro vehicle (axleload 12 t/axle) with v≤77 km/h;

4. a D = 920 mm wheel for passenger trains (axleload 18.5 t/axle) based on a current monobloc wheel designed to run up to 360 km/h.

It is worth to highlight that the *wheel 2* is designed to be equipped with brake blocks, resulting in a lower wear limit, while *wheel 4* has the highest wear limit value, which is the maintained from the original monobloc design. In fact, it is known that the wear limit for tyred wheels is always lower than the one set for monobloc wheels, as the tyre-wheel centre pressure has to be guaranteed also in worn conditions.

Table 1: main parameters of the simulated tyred wheels.

Wheel	**D [mm]**	**Wheel load *Pmax* [t/wheel]**	***Vmax* [km/h]**	**Hub fitting diameter D_i [mm]**	**Tyre fitting diameter D_e [mm]**	**New tyre thickness S_n [mm]**	**Worn tyre thickness S_w [mm]**	**Shape of the wheel centre**
1	940	6.5	200	190	790	75	40	S-shaped
2	920	10	120	185	790	65	40	S-shaped
3	840	6	77	160	718	61	31	Corrugated
4	920	9.25	360	220.5	790	65	30	Straight

The original wheel web shape and the thickness were maintained for *wheel 4*, resulting in a straight wheel centre that is unconventional for standard wheel centres as it does not provide enough radial elasticity of the wheel.

Wheel 3 is designed for a metro vehicle equipped with brake discs, resulting in lower external diameter D and tyre fitting diameter De. The forged wheel centre of this wheel has a complex shape that was acquired with a 3D scanner. This shape, from which the wheel is called "corrugated wheel", was quite common in the past especially for small diameter wheels and gives a great resistance of the wheel to lateral load, resulting in a lower thickness and a lower mass of the wheel centre [6].

For each wheel, four load cases are evaluated, arising from the combination of the following conditions: new/worn tyre and maximum/minimum interference. As the variation of the interference value at the mating surface of the wheel hub and the axle seat is considered not relevant to the scope of the present study, the maximum value for this parameter is used for all the performed simulations.

The following parameters were evaluated for all the cases:

i) stress state due to fitting of the axle in the hub with the maximum interference;
ii) maximum and minimum principal stress in the wheel centre;
iii) maximum and minimum principal stress in the tyre;
iv) normal load acting on the mating surface of the tyre and the wheel centre simulating a frictional contact;
v) maximum transmissible torque with tyre-wheel centre friction of 0.3;
vi) radial and axial displacements of the tyre.

According to the current maintenance procedures for tyred wheels in Italy, the interference value for the wheel centre and tyre shrink fitting is set to

$$i_e = \frac{1.3 \pm 0.1}{1000} * D_e$$

where D_e is the tyre fitting diameter in mm.

The interference value for the wheel hub and wheel seat on the fitting, was chosen accordingly to [7], considering also in this case a shrink fitting. Therefore, the value is set to

$$i_i = \frac{1.5}{1000} * D_i$$

where D_i is the hub fitting diameter in mm.

Resulting interference values are reported in Table 2. It is worth to highlight that the shrink fitting is made heating the tyre up to 220 °C, while the minimum over temperature to recover the interferences is 117 °C.

Table 2: interference values used as input of the finite element models.

	Tyre fitting		**Axle fitting**	
	i_{max} [mm]	i_{min} [mm]	i_{max} [mm]	i_{min} [mm]
Wheel 1	1.106	0.948	0.285	-
Wheel 2	1.106	0.948	0.278	-
Wheel 3	1.027	0.862	0.270	-
Wheel 4	1.106	0.948	0.330	-

Meshed models of the wheels are shown in Figure 2. Stresses and strains are evaluated for simplified 3D models by using 2D axisymmetric loads and boundary conditions, while visualization is for the whole wheel. Only for *wheel 3*, this was not possible due to the complex geometry of the corrugated wheel centre and the results were calculated for a fifth of the whole wheel and then considering a cyclic symmetry.

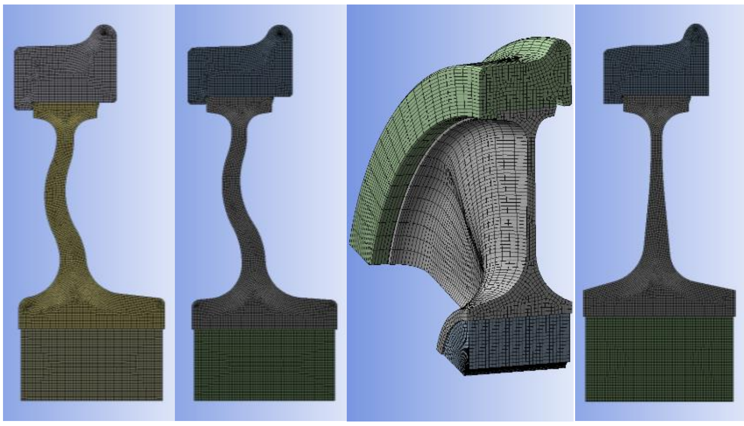

Figure 2. Left to right: meshed models of *wheels 1*, *2*, *3* and *4*.

2.2 Validation of the model

A simple validation of the model was performed by comparing the results with the behaviour of a tyred wheel shown in [8], where both circumferential and radial stresses on the external and internal surfaces of an S-shaped wheel centre were studied using strain gauges measurements during the axle fitting process and the tyre fitting process.

The diagrams showing the trend of these stresses in function of the distance from the wheel/axle interface are shown in Figure 3. From these diagrams, it is possible to conclude that the stresses due to the tyre fitting process are much higher than those due to the axle fitting process, reaching values that can be greater than the yield point, resulting in plastic deformations.

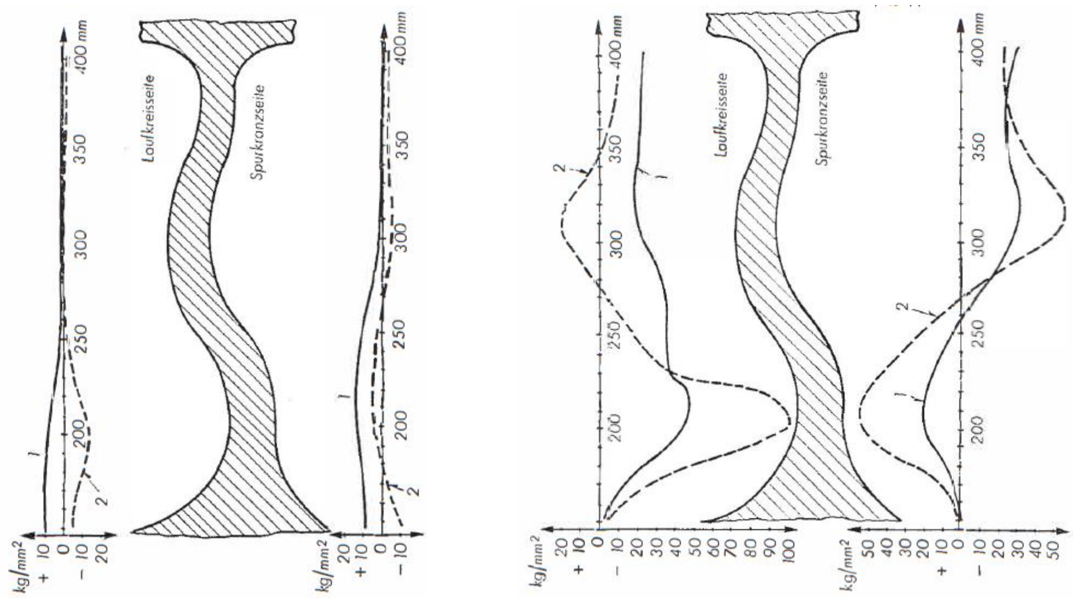

Figure 3. Left: circumferential (1) and radial (2) stresses during the fitting of the wheel on the axle. Right: circumferential (1) and radial (2) stresses during the fitting of the tyre on the wheel.

According to [9], steels used to manufacture wheel centres are characterized by high fracture elongation and quite low yield stress. For example, the steel of the wheel centre shown in [8] had a yield stress corresponding to about 200 MPa. The tensile strength curve of this material is shown in Figure 4 (left). In the same figure, this limit is compared with the von Mises equivalent stress resulting from the previous circumferential and radial stresses.

Starting from these experimental evidences, a simulation with a linear elastic material model and maximum interferences was performed on *wheel 2,* whose wheel centre has strength properties similar the ones shown in Figure 4. The results are shown in Figure 5.

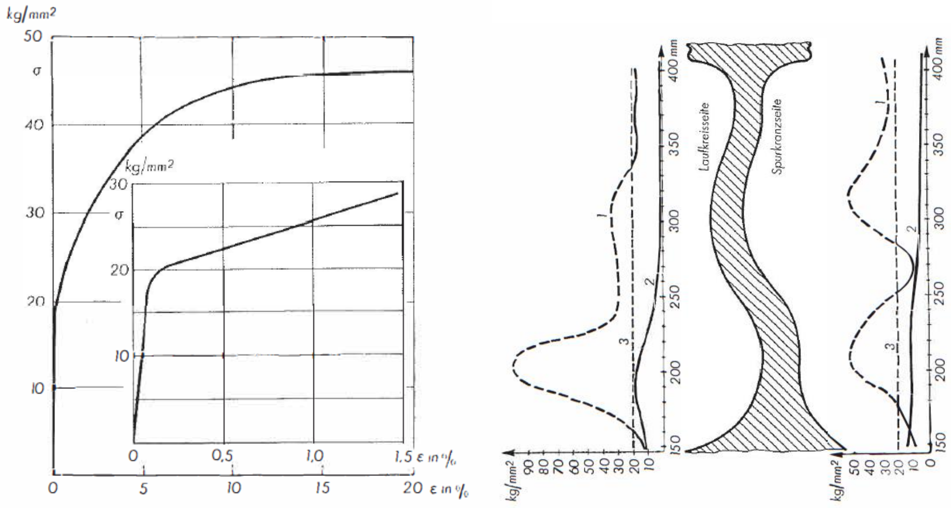

Figure 4. Left: tensile strength curve for the wheel centre steel, with a focus on the yield point. Right: comparison of the von Mises equivalent stresses with the yield stress (3), for the axle + tyre fitting (1) and for the axle fitting only (2) [8].

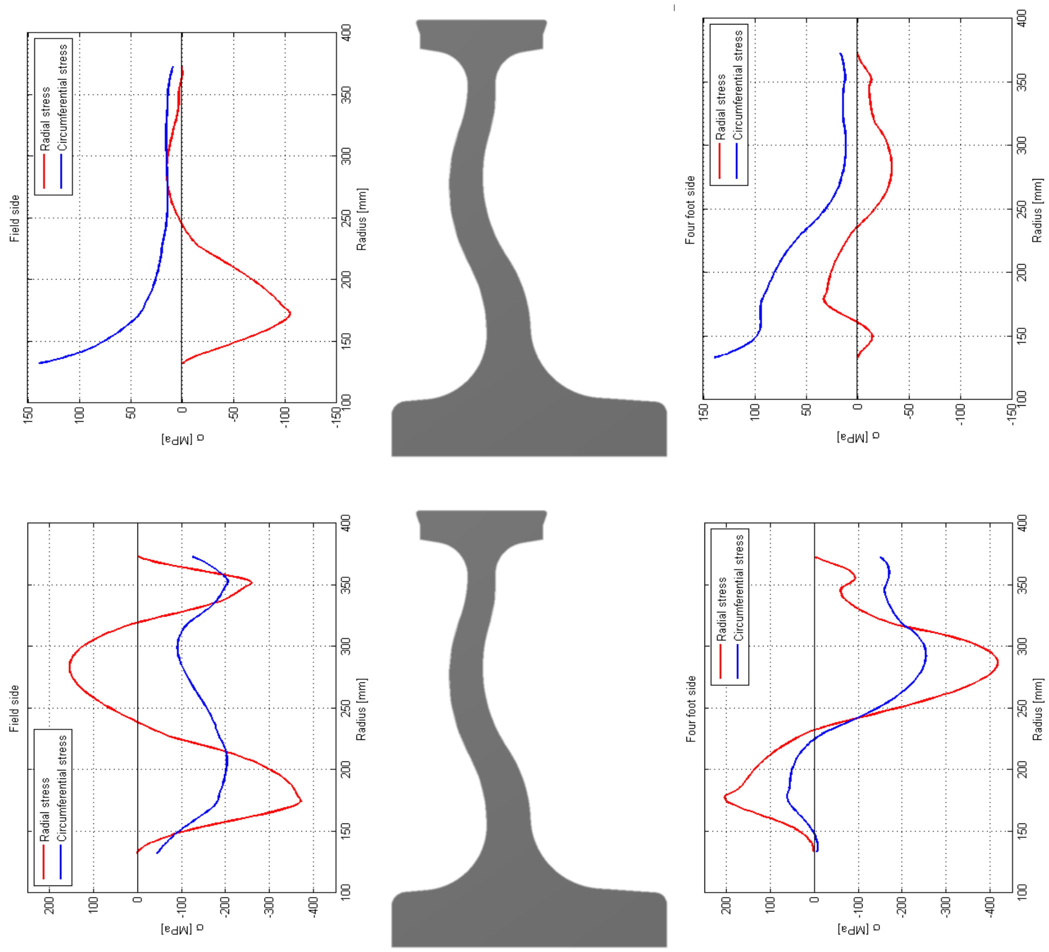

Figure 5. Radial and circumferential stresses in the case of axle fitting only (top) and in the case of tyre fitting only (bottom).

Even if the simulated maximum values are slightly lower (the interference values used in [8] are higher than the ones used in the simulations and small geometry differences may influence the results), the trend of the stresses are almost equivalent. In addition, von Mises stresses were calculated with the aim to compare the model to the results previously described in Figure 4.

It is evident that the elastic limit is exceeded. A purely linear model of the material behaviour is not sufficient and an ideal elasto-plastic material model with a yield stress set to 235 MPa was used. Resulting von Mises stresses are plotted in Figure 6 for both material models.

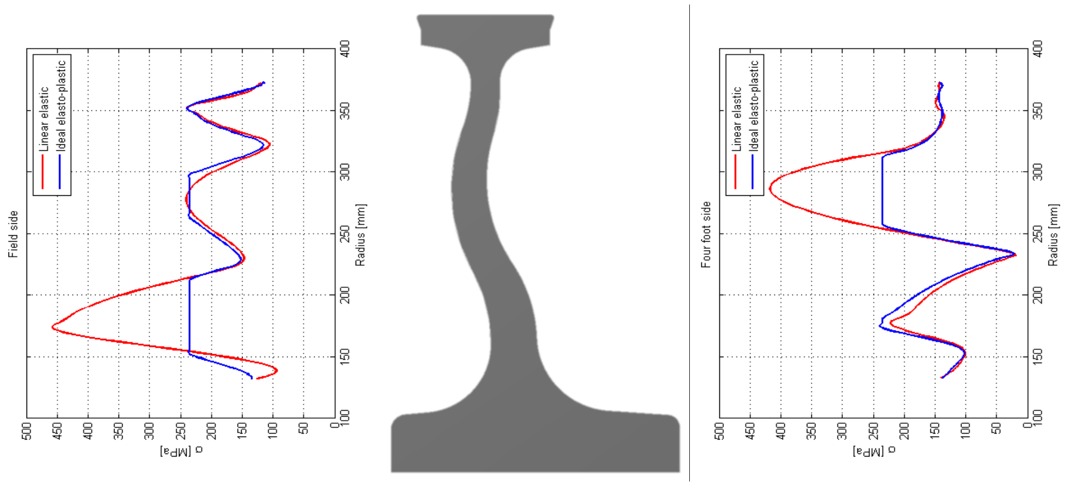

Figure 6. von Mises stresses for a linear elastic material model and an ideal elasto-plastic material model.

The resulting equivalent plastic strains in the zones where the yield point is exceeded are shown in Figure 7. Their maximum values are relatively small (2 ‰), but they may be sufficient to modify permanently the shape of the wheel centre, with possible influence on the wheelset tolerances, which have to be guaranteed for the correct wheel and wheelset assemblies.

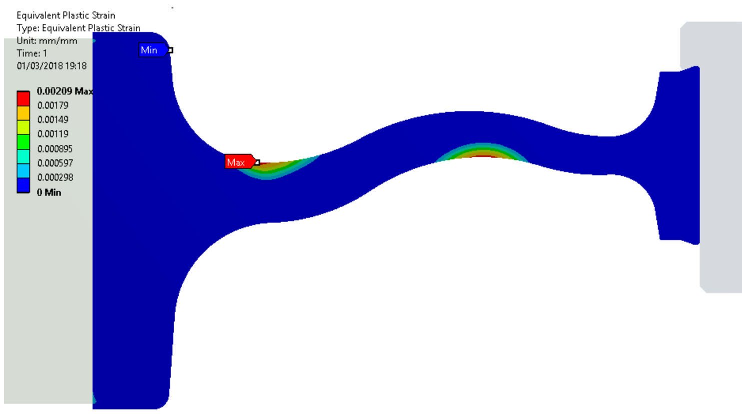

Figure 7. Equivalent plastic strain resulting from the tyre fitting process, considering an ideal elasto-plastic material model for *wheel 2*.

3 Results of the analysis

3.1 Wheel centre radial stiffness as an indicator

In order to compare the behaviour of the different tyred wheels, the radial stiffness of the wheel centre was chosen as a representative parameter. It mainly depends on the shape of the wheel centre; the ability of the wheel centre to withstand the fitting of the tyre and to maintain the pressure at the tyre/wheel centre interface as the tyre wears is linked to this parameter. As the tyre thickness decreases due to wear and reprofiling, the wheel centre has to be elastic enough to guarantee a minimum pressure at the interface. The value of the radial stiffness was calculated applying a pressure of 1 MPa at the tyre/wheel centre interface and evaluating the radial displacement of the central node of the interface surface (Table 3).

Table 3. Radial stiffness of the wheel centres.

Wheel	**Radial Stiffness [MPa/mm]**
1	108.23
2	81.70
3	56.18
4	119.05

The higher value of radial stiffness is obviously related to the straight web of *wheel 4*, while the others wheels have a more elastic wheel centre given by the S-shaped or corrugated web. For the corrugated wheel (*wheel 3*), the radial stiffness is calculated taking into account the mean value of the radial deformation around the circumference length, as shown in Figure 9Figure 8. As the radial stiffness of the straight web *wheel 4* is considered the highest practically reachable value, it was taken as a reference value. Radial stiffness shown in the following for the others wheels are therefore normalized to this reference value.

Even if it was shown above that permanent deformations may occur during the fitting process, for the sake of simplicity the results are presented considering a linear elastic material model. Results can therefore be evaluated regardless the different material properties (i.e. yield strength), but only considering the shape and the geometry parameters of the wheels.

3.2 Hub fitting process

The first step of the analysis is the evaluation of the hub fitting process, considering the maximum interference between the axle and the wheel centre and evaluating the pressure at their interface (Figure 9). The mean value of the pressure was used to calculate the normal force acting at the wheel/axle interface, the tangential force with a friction coefficient of μ=0.3 and the resulting maximum transmissible torque.

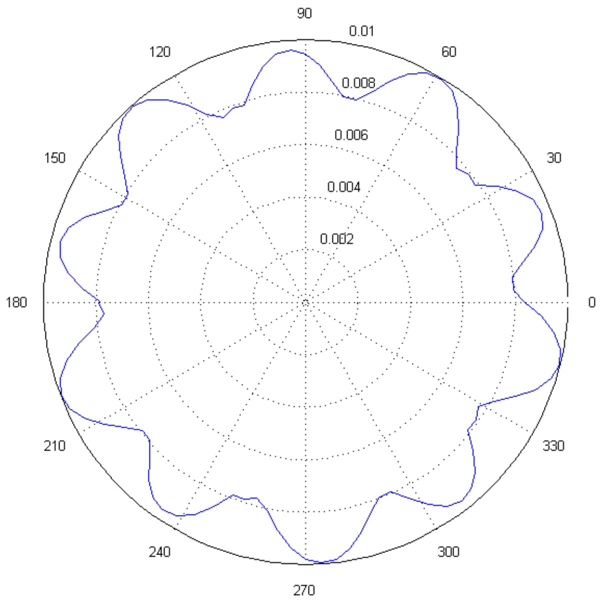

Figure 8. Radial deformation of a corrugated wheel centre.

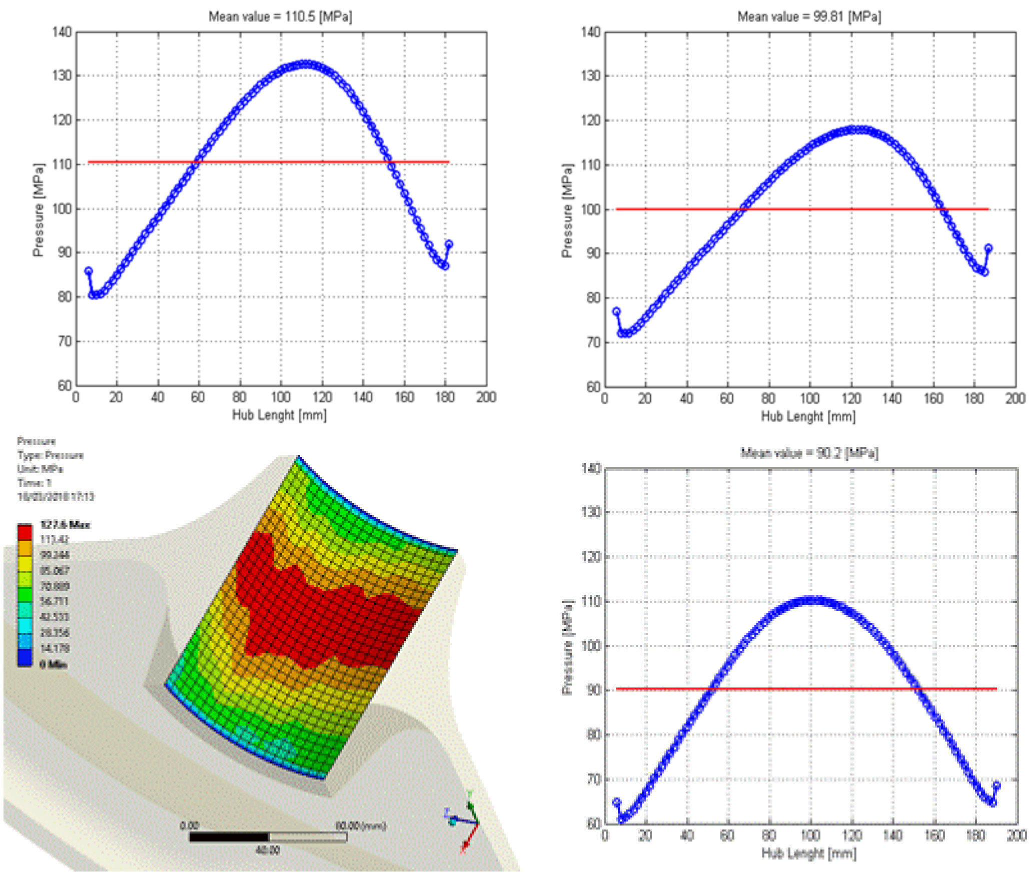

Figure 9. Pressure at the wheel/axle interface with the maximum interference for the axisymmetric wheels. Left to right, top to bottom: *wheel 1*, *wheel 2*, *wheel 3* (non-axisymmetric) and *wheel 4*.

All these values are in line with the standard practice of wheel fitting. In fact, as described in [7], the wheel has to withstand an axial force of *4*Di* [kN] for 30 seconds, where *Di* is the fitting diameter in mm, to guarantee the absence of slipping in the fit. In all the studied cases, the tangential force resulting from the fitting is higher than the request value, with a minimum safety coefficient of 3.1 (Table 4). These values confirm the validity of the model and supply the reference values for the transmissible torque at the tyre-wheel centre interface.

Table 4: results of the hub fitting process for the maximum interference value.

Wheel	**Mean Pressure [MPa]**	**Tangential Force [kN]**	**Minimum Tangential Force (*4*Di*) [kN]**	**Safety coefficient**	**Maximum transmissible torque [MNm]**
1	99.8	3240	760	4.3	0.31
2	110.5	3390	740	4.6	0.31
3	88.9	1997	640	3.1	0.16
4	90.2	3450	880	3.9	0.39

3.3 Tyre fitting process

The second step of the analysis consists in adding the shrink fitting of tyre to the previously analysed axle fitting. In this case, both maximum and minimum interference are evaluated. In the following paragraphs, the main parameters are described separately.

3.3.1 Pressure at the tyre-wheel centre interface

The pressure distribution at the wheel centre/tyre interface is the parameter that guarantees the consistency of the tyred wheel assembly even in worn tyre conditions and with the minimum interference (Figure 1Figure 11). These pressure values are strongly depending on the radial stiffness of the wheel centre.

Plotting pressure values against the radial stiffness of the wheel centres it is possible to see that the mean value increases with the radial stiffness, but at the same time the pressure reduction due to a lower interference or a worn wheel is greater for a wheel centre with a high radial stiffness. This behaviour, clearly visible in Figure 11 where all the simulated conditions of both tyre fitting pressure and maximum transmissible torque are shown are plotted, is correlated to the inability of the straight wheel centre of *wheel 4* to deform radially and to recover this deformation when the constraint offered by the tyre is reduced due to the wear process.

Reduction at the wheel centre-tyre interface is reported in Table 5. Even if the straight wheel centres has a tyre pressure reduction of about 50% from the initial value, its minimum value is not zero and therefore a certain constraining action of the tyre on the wheel centre is always guaranteed. In general, the maximum transmissible torque values are always greater than the ones obtained for the wheel/axle interface.

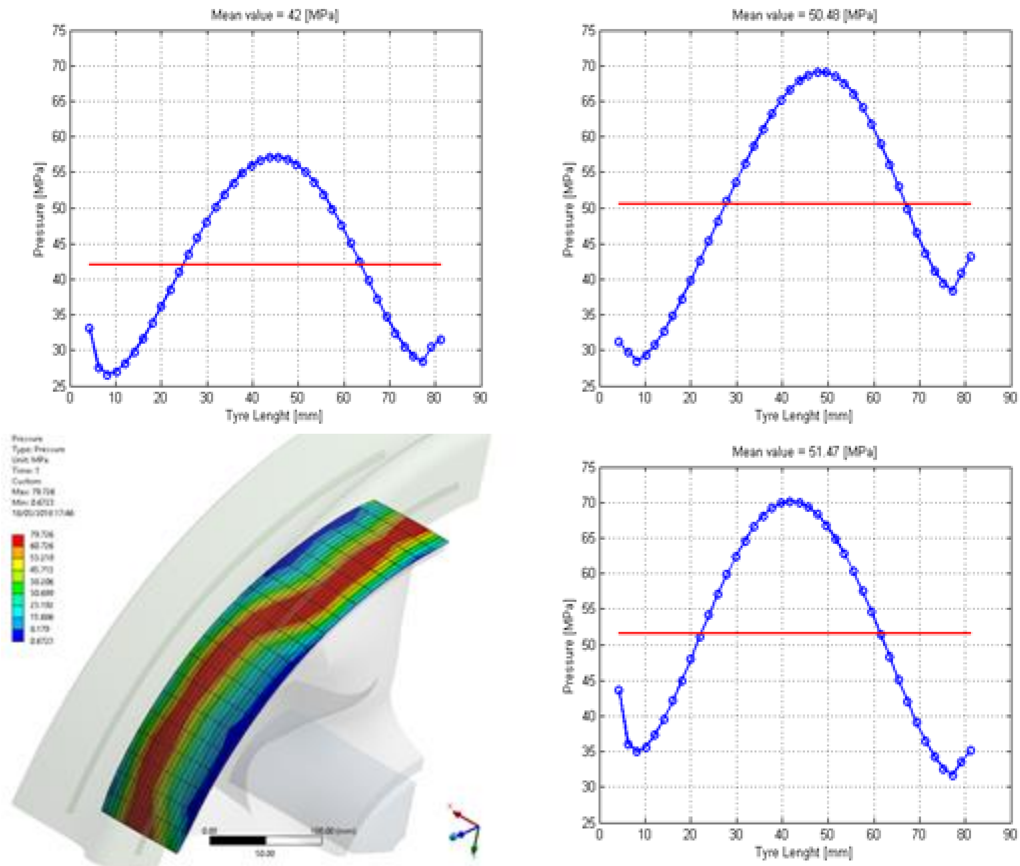

Figure 10. Pressure at the wheel/tyre interface with the maximum interference and new tyre for the axisymmetric wheels. Left to right, top to bottom: *wheel 1*, *wheel 2*, *wheel 3* (non-axisymmetric) and *wheel 4*.

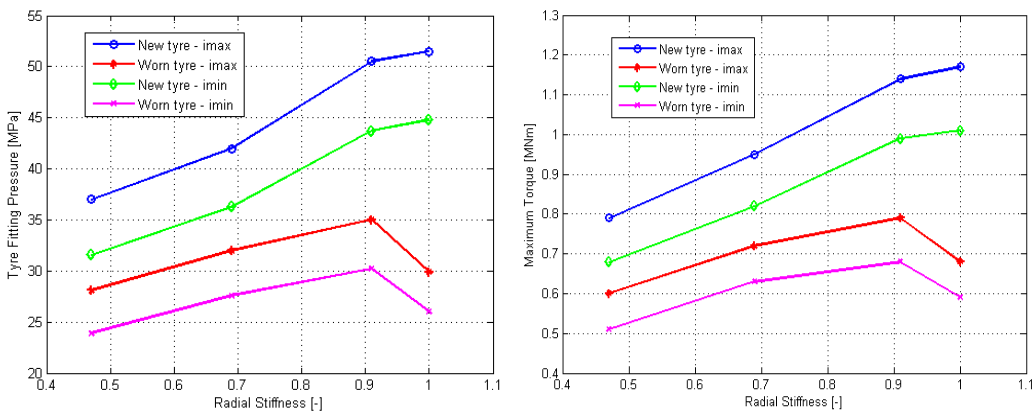

Figure 11. Tyre fitting pressure (on the left) and maximum transmissible torque (on the right) against the adimensionalized radial stiffness of the wheel centre.

Table 5. Tyre pressure reduction considering a maximum tyre wear of 40 mm

Wheel	Tyre pressure reduction
1	35 %
2	38 %
3	32 %
4	48 %

A high pressure reduction at the tyre interface may be incompatible with tread braking in which the tyre expands due to temperature increases. In literature, recent studies of the thermal behaviour of wheels during the tread braking can be found only for monobloc wheels [10]. As the goal of the research is focused on disc-braked wheelsets, this topic will not be expanded further.

3.3.2 Tyre stresses and lateral displacement

During the tyre fitting on the wheel centre positive (i.e. traction) circumferential stresses are present in the tyre. These stresses increase with the reduction of the thickness of the tyre and with the increase of the radial stiffness of the wheel centre (Figure 12).

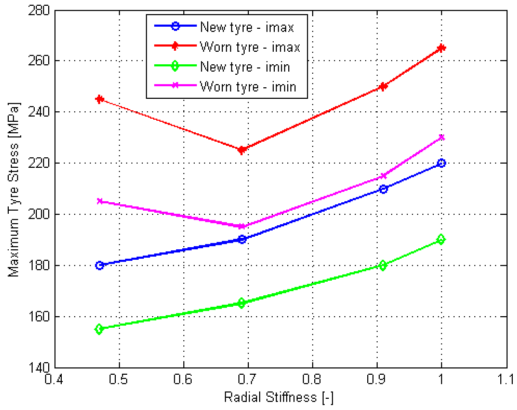

Figure 12. Tyre circumferential stresses vs. radial stiffness in new and worn conditions.

As shown in Figure 13, the maximum stress always occurs at the field side abutment, which have the minimum cross section, representing the structurally weakest point of the tyre, obviously neglecting rolling contact fatigue damages on the wheel surface. However, when the thickness of the tyre become slimmer all the tyre resistance section decrease and the circumferential stress become more uniformly distributed.

Another important parameter about the tyre is the lateral displacement. In fact, if the wheel centre has not a sufficient axial (lateral) stiffness a lateral displacement of the tyre may occur during the fitting. Even if the maintenance procedures of tyred wheels are discussed in [1], it can be said that is a common practice to finish the tyre profile only after the fitting in order to prevent possible uncontrolled variations of the nominal dimensions that have to be respected before putting the wheelset in service. However, the models do not show particularly high lateral displacements, as shown in Figure 14, where the maximum value is 0.45 mm for *wheel 2*, while for the other wheels the value is always about zero. Thus, lateral displacement is not depending on radial stiffness or on interference and wear conditions. These values of lateral displacement seems to be manageable without machining the wheel after the tyre fitting; it is necessary, however, to take into consideration that permanent deformations may occur as aforementioned.

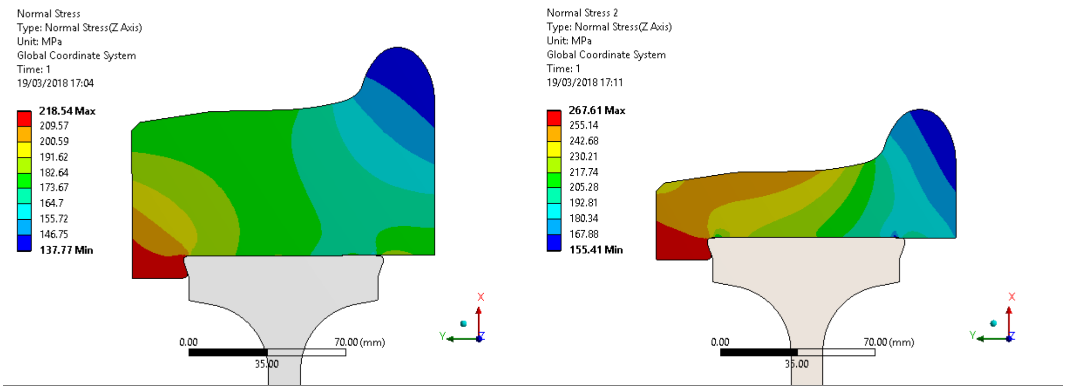

Figure 13. Circumferential stresses of the tyre of *wheel 4:* new tyre with maximum interference (left), worn tyre with maximum interference (right).

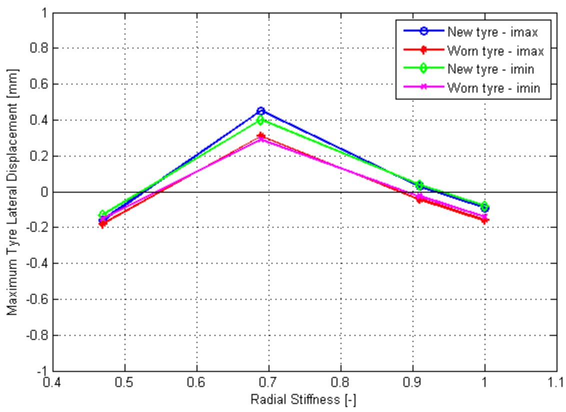

Figure 14. Lateral displacement of the tyre after the shrink fitting in all the simulated conditions.

3.3.3 Wheel centre stresses

As the tyre is mainly subjected to traction stresses, it is common to think about the wheel centre as subjected to compressive stresses. However, in a standard S-shaped wheel centre both radial and circumferential stresses occur, as seen above. Radial stresses are dominant and even if the maximum values are related to compressive radial stresses, also traction radial stresses can reach relevant values. These high stresses are generated by the bending of the curved transitions of the elastic wheel centres and they can be eliminated for a straight wheel centre, which is subjected only to compressive stresses. Maximum and minimum principal stresses are shown in Figure 15. All stresses in the wheel centre decrease with the reduction of tyre thickness and of the interference values, as the stiffness of the fitted tyre becomes lower.

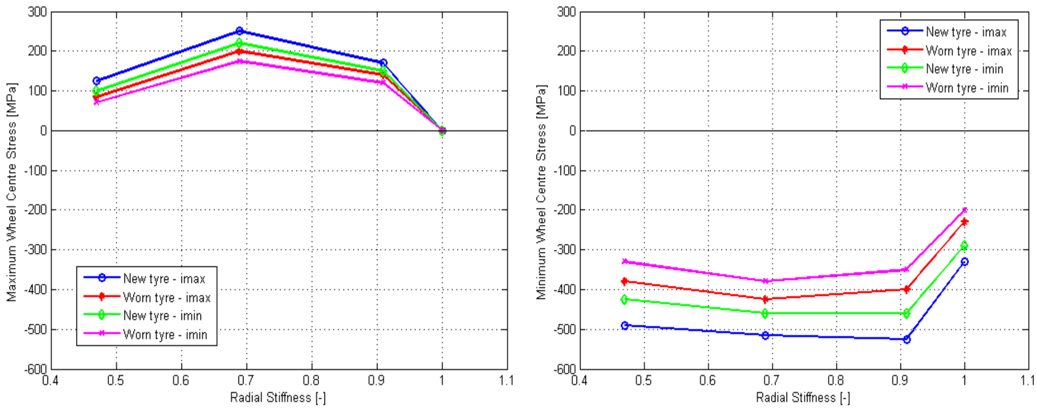

Figure 15. Maximum and minimum principal stresses in the wheel centre for all the simulated conditions.

4 Conclusions

Enhancements in the design of tyred wheels are very limited as it is based only on the old knowledge suggesyed by the operator experience. Studies about this topic are not present in the literature and the industrial progress is negligible as well. All the effort are concentrated on monobloc wheel, forgetting that in certain conditions tyred wheel can be an important alternative to save time and money during maintenance.

This research evaluated tyred wheels behaviour during the fitting operations of the centre wheel on the axle, and of the tyre on the centre wheel. Three existing tyred wheels were analysed using Finite Element Analysis, adding a fourth hypothetic tyred wheel with a straight web to expand the range of the shapes the wheel centres. All the simulations were performed without thermal loading and considering as input only the interferences at the mating interfaces. Stresses and strains in various operating conditions were evaluated both considering linear-elastic and ideal elasto-plastic material models.

A series of parameters were evaluated to identify possible weak points of this kind of wheels and with the final aim to understand possible developments for a structural optimization.

Radial stiffness of the wheel centre was found to be the most important parameter to maintain an optimal value of the pressure at the tyre/wheel centre interface for both new and worn tyres. Typical S-shape of old wheels was a way to limit radial stiffness but involved bending (and therefore traction stresses) in the wheel web. On the other hand, a radially very rigid straight wheel web is subjected only to smaller compressive stresses, generating however high circumferential stresses in the tyre in worn conditions.

Asymmetry in the S- or C-shaped wheel centre inevitably results in lateral displacements of the tyre after cooling. Moreover, as resulting permanent strains in the low yield stress steel used for wheel web were common while searching the highest tyre-wheel centre pressure to avoid tyre spinning, tyre fitting may results in hardly controllable lateral displacements of the tyre. This was in the pas fully recovered by machining the tyres after mounting, a practice that is so expensive that led to the practical withdrawal of tyred wheels.

The set up model developed proved to supply results in line with the existing literature and is at the basis of the development of a new philosophy of the use of tyred wheels of which it is the first and fundamental step.

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