Tyred wheels without braking: structural optimization

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Abstract

Tyred wheels were used in the past when labour and fixed costs were much less important than today. Modern monobloc wheels benefitted from advanced numerical tools (finite elements) and improved tooling, making them the most convenient solution.

Nevertheless, tyred wheels can still be an interesting advantage when both design and maintenance are re-considered taking into account the numerous changes in the railway vehicle design, i.e. the development of braking systems acting on dedicated surfaces (brake discs mounted on the axle, on wheel web, on parallel axles, on engine shaft, etc.)

When the thermal load due to tread braking is removed, the structural calculation of tyred wheels can be totally revisited leading to lighter, quieter and much cheaper solutions.

The paper discusses some basic considerations on the oversizing of conventional tyred wheels, showing the potential advantages. Outline of further activities are provided, including fretting analysis and wheel centre design.

Keywords: tyred wheels, railway wheels, Finite Element Analysis, structural strength, thermal loads

1 Introduction

Tyred wheels were used since the beginning of the railways era, being the direct extension of cart wheels made by spoked wooden centre and a steel ring fixed to it. The main advantage of tyred wheels is in fact the possibility to replace the worn tyre with a new one without changing the wheel centre that, when properly dimensioned and checked, has an infinite life.

This advantage was linked to the use of (expensive) cast steel spoked wheel centres, needed to reach inboard journal bearings that equipped steam locomotives. Until very recent years, and with a large number of differences, the inboard bearings solution disappeared in favour of external bearings and the spoked cast wheel disappeared as well replaced by cast or rolled/forged wheel centres.

Starting approximately 40 years ago, the use of monobloc wheels became widespread for several reasons. They can be listed as follows, in no particular order:

- centrifugal actions on the emerging high speed operations were detrimental for safety, a problem that monobloc wheels do not experience;
- braking leads to relatively frequent tyre loosening especially when the tyre is worn, with serious effects on safety;
- maintenance costs, especially labour costs, raised dramatically with the advent of automation, something that cannot be used extensively with the current design of tyred wheels.

The authors give in two companion papers some new inputs in the field of tyred wheels that did not develop in last decades. Specifically, paper [1] shows how modern numerical computing methods may lead to lighter and quieter wheel centres, while paper [2] discusses a simple modification to the geometry of the tyre / wheel centre pair mating surface that makes tyred wheels an interesting option for automated maintenance cycles.

The key to the resurrection of tyre wheels is linked to the overcoming of the previously mentioned limitations. This is possible considering that braking is performed today on specifically designed surfaces, i.e. brake discs mounted either on the axle or on wheel web, eliminating tyre heating generated by tread braking and the "tread loosening problem". Nearly all modern passenger vehicles are disc braked, and considering the widest part of the vehicles, i.e. conventional speed fleets, the three problems mentioned above can be effectively remove by conceiving a safe, lightweight and easy-to-maintain tyred wheel.

This paper discusses also a key issue of conventional tyres wheels, i.e. their high mass compared to monobloc wheels. The need to withstand both mechanical and thermal loads generated tyres of noticeable thickness that are not needed anymore when removing the braking action. It will be shown that disc-braked tyred wheels can be competitive in terms of mass with monobloc wheels, and that with a proper design of the overhaul cycle they can also be much cheaper to maintain.

2 Current regulations on tyred wheels

Wheel centres are not standardized by EN norms that were written only for monobloc wheel. The only reference to tyred wheels is in the informative Annex H of the EN 15133 [3] about vehicle wheelset maintenance where it says "*Requirements for tyred wheels are specified in the following UIC leaflets: UIC 810-1, UIC 810-2, UIC 810-3, UIC 812-1, UIC 812-4, UIC 812-5 and UIC 813*".

It is not clear whether the word "requirements" refers only to maintenance for two reasons: first, the mentioned UIC leaflets (see refs. [4] to [10]) include design considerations and are not limited to maintenance; second, no common standards or codes that may help the designer to dimension a new tyred wheel are available today.

It is interesting to highlight that UIC 812-4 [8] is quite vague. For example, the interference between the wheel centre and the tyre is defined at par. 5.1 according to the following excerpt (Figure 1):

$$c_{\text{wheel tyre bore}} = C_{\text{wheel centre}} - \frac{X \cdot C_{\text{wheel centre}}}{1000}$$

$C_{\text{wheel centre}}$	=	the value measured on the outer diameter of the wheel centre rim in mm
X	=	a factor within the range from 1.3 to 1.8 to make allowance for influences, such as wheel centre design and stiffness. (1)

(1) The "X" factor may be brought down to 1.1 for large diameter wheels.

Figure 1. Tyre-wheel centre interference as defined by [8]

At point 5.4 "Tyre fitting" the standard says "*The wheel tyre must be heated, preferably, to between 200 °C to 250 °C, but not above 300 °C, using an approved method*".

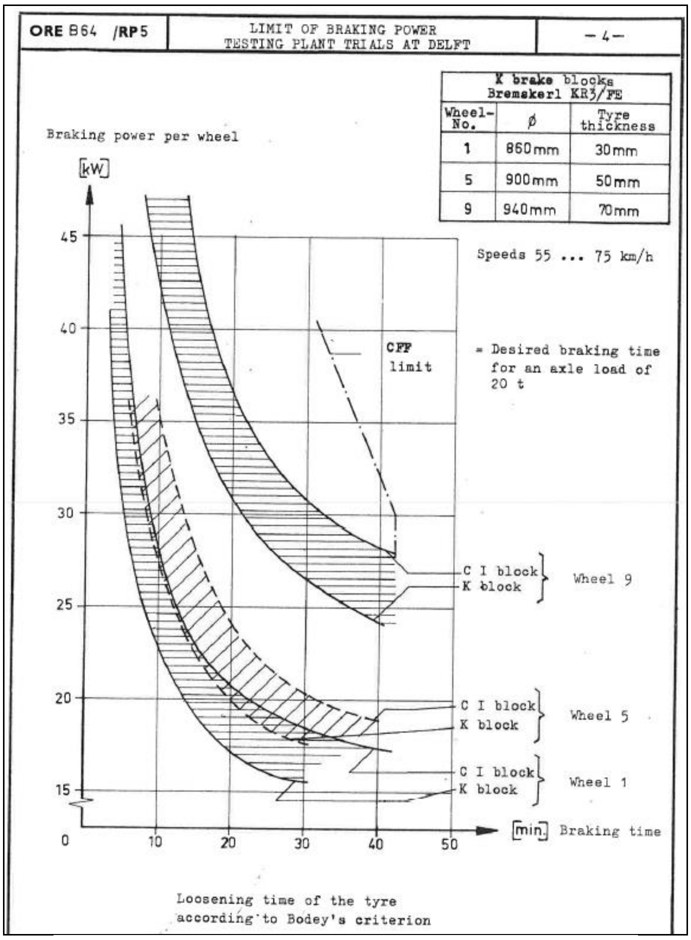
It is evident that the proper definition of the interference, that needs to be sufficient in all service conditions (new, fully worn, cold tyre, hot tyre) is left to the single administration. Structural parameters such as "*centre design and stiffness*" may vary considerably. As a result, there may be a difference of more than 60% between the minimum interference and the maximum intereference!

Again, although in theory there was the possibility to adapt the interference to the actual design of the wheel, the interference was kept constant (as an example, the Italian State Railways FS used in the past a value of ΔD=1.2÷1.4 D/1000) *regardless of the braking applied and of the axleload*. This proved to be sufficient, in most of the case, to avoid tyre spinning.

3 Limitations of tyred wheels during braking and their end

The *Office de Recherche et d'Essais* was a branch of the *Unione Internationale des Chemins de Fer* that investigated many aspects of vehicles and tracks starting in the 1950's.

The *Question B64 Composition brake block* investigated the *limits of energy dissipation during braking with tyred wheels* in its report B 64/RP 5/E [11]. Figure 2 shows the unsurpassable limits intrinsic in the tyred wheels technology when applied to tread braked wheelsets. In the case of fully worn wheels (tyre thickness = 30 mm) with an interference of 1.43 ‰, tyre loosening always happens in less than ten minutes also for moderate braking power (around 30 kW).



Wheel No.	Shrinkage allowance o/oo	Tyre thickness mm	Tread Diameter mm
1	1.43	30	860
5	1.62	50	900
9	1.83	70	940

Figure 2. Loosening time of the tyre according to Bodey's criterion ([11], [13]).

It is interesting to observe that while one of the first available studies on the field dates back to 1959 [13], Prof. Sachs in his famous book [12] issued in 1973 did not even mention the fundamental work made by ORE and published in 1968 [11].

As an inevitable consequence, tyred wheels did not benefit from the modern calculation and testing techniques. They were left to spare parts and for some "poor" markets such as "*Asia and parts of Africa*" [14].

In 1994, the *International Organization for Standardization* in its standard ISO 1005-1 on tyres [14] said "*at present, tyres are preferably used for repairs while new wheels are mainly manufactured as solid wheels… this will therefore be the last edition of ISO 1005-1 and it was decided not to include an alignment of tyre grades gives in this part of ISO 1005 with the grades of solid wheels in ISO 1005-6*".

The UIC Code 510-2 [15] can be considered the "gravestone" on the tyred wheels application. "*With effect from 1 July 2000*" (therefore before the publication of EN standards on wheelsets), at point 1.3.1 it states "*From 1.1.89, new wagons are to be equipped with solid wheels*". After that date no literature was further published on the subject of tyred wheels.

After the Viareggio accident in 2009, due to a broken axle, the European Commission established a Task Force (TF, named "Freight Wagon Maintenance") under the leadership of the European Railway Agency, in order to examine the all the technical subjects about axle fatigue, define monitoring, evaluate the role of standards and *assist the Sector and NSAs to establish sound evidence and advice on the causes of the broken axles problem*. In document [16] it is reported that "*It has to be noted that only one of these companies (counting for about 5% of all axles in this FTA) still uses tyred wheels. The sector itself agreed to stop tyred wheels operation by 2020 at the latest*". Therefore, in no more than a couple of years tyred wheels will disappear forever from freight wagons.

4 Thermal behaviour of monobloc wheels

Tread-braked monobloc wheels are used today mainly for freight wagons as FEM calculations allow to properly take into account a number of non-linear parameters such as material properties, specific heat and heat exchange convection and conduction coefficients. This approach led to wheels with different web shapes, intended to provide a sufficient elasticity (without noticeable plasticization) during the particularly demanding heating / cooling cycle that the wheel must withstand during the qualification phase.

In recent years many models including the thermal effect given by tread braking, supported by experimental evidences, were developed and validated through measuring campaigns conducted on *monobloc* wheels. Some test results can be found in [18], while [17] describes a thermal model of the wheel and the bearings (that can be affected by heat transmitted through the wheel web). Among the most interesting results in combined structural / thermal calculations it is necessary to mention the papers by Vernersson ([19][20][21]) and Teimourimanesh ([22][23]) belonging to the school led by Prof. Lundén. Their result range from heavy haul to metros, with models validated on brake benches and with field trials. The interested reader is referred to this material that *will not be used* in this research as the only type of braking considered here is *disc braking* that introduces negligible thermal input in the wheel centre also in case of web-mounted disc brakes.

5 Structural re-design of tyred wheels

5.1 Introduction

Modern calculation tools were not available during the development of tyred wheels. Structural requirements were obtained primarily by experience on similar cases, and the number of documented case studies in the existing literature is extremely limited.

As tyred wheel went out of fashion, no one investigated the possibility of conducting an optimization of the tyred wheel considering that braking is more and more applied to web-mounted or axle-mounted brake disc, freeing therefore the tyre and the entire wheel from any braking-related coupled thermo-elastic problem. High maintenance costs and the absence of standards (discussed in detail in [2]) completed the job, leading to the disappearance of tyred wheels.

Allowable (minimum) tyre thickness and fitting interference must satisfy the following boundary conditions (see also [1]):

- compressive stresses in the wheel centre descending from the shrink fitting of the tyre have to remain “reasonably” within elastic limits of the material;
- traction stresses in the fully worn tyre have to remain compatible with its strength without fracturing it;
- pressure at the tyre-wheel interface has to remain sufficient to transmit the full torque also in fully worn conditions of the tyre;
- overall mass of the tyred wheel should be reduced as much as possible to be competitive with monobloc wheels.

A fully elastic analysis is discussed in the following to determine the minimum mass of a disc-braked tyred wheel.

5.2 Literature analysis

The problem of tyre loosening and possible tyre/wheel centre relative rotation has always been central. One of the milestones on the study of slip of thin tyres fitted on wheels is the paper by Anscombe and Johnson [24], although only normal load (no traction or braking) is considered, and the results are particularly interesting for bearing rings.

One of the few (if not the only one) contribution to the purely elastic problem when tangential forces are considered is the paper by Reina *et. al.* [25], which defines the minimum friction coefficient between the tyre and the wheel centre to avoid spinning when no thermal inputs are involved. The use of the paper is nevertheless limited for a number of reasons.

First, the friction tyre-wheel centre friction coefficient is a parameter that is outside the control of railway engineers, as steel-on-steel coupling has a "conventional" friction coefficient of μ=0.3 that cannot be easily modified.

Then, the authors mix the wheel-rail contact problem with the tyre-wheel centre spinning problem. As reported in the large existing literature for the first case, no one is particularly scared by plasticization at the wheel-rail contact, which is something that frequently happens and, under controlled conditions, is beneficial leading to stress-hardening of the steel. The conclusions reached in the paper can therefore be easily refuted by using, e.g., equations from [26] or [27].

Last but not least, the half-space formulation is too rough for an effective transverse wheel-tyre coupling size of around 90 mm compared to the contact patch semi axes of the elliptical patch, in the order of 5 to 10 mm. Although interesting, the paper is useless in case someone needs to perform reliable numerical estimation from the equations described there.

Ref. [15] states the operating (fully worn) limits for tyre thickness. Passenger coaches have a limit of 35 mm, but they can use tyred wheels only if their speed is $v \leq 160$ km/h. Freight wagons tyre maximum wear depends on homologation speed, as follows:

- $v > 120$ km/h: tyred wheels are not permitted;
- $v = 120$ km/h: 35 mm
- $v = 100$ km/h: 30 mm
- $v < 100$ km/h: 25 mm

Although vehicles running in under S or SS conditions set out in UIC leaflet 432, it looks clear that a minimum thickness of 25 is still compatible with normal operations of a low-speed vehicle, including emergency and drag tread braking. As usual with step-wise limits, a question arises about the real differences in the behaviour of a vehicle homologated for 100 km/h or, let us say, 95 km/h maximum speed.

Once again, it is useful to underline that UIC leaflets are all written keeping in mind *tread braking*, while the present paper discusses only *disc braked vehicles* without any thermal input. The only value that must be preserved from old UIC leaflets is the *tread wear* that cannot be greater than 40 mm to avoid coupling problems of adjacent vehicles.

5.3 Dimensioning of the tyre-centre pair

As said above, new tyre conditions are the worst for stresses in the wheel centre (but the safest against spinning) while fully worn tyre conditions are the worst for both tyre stresses and the risk of spinning.

In order to compare the purely mechanical behaviour of different vehicles when equipped with tyred wheels, a selection among the vehicles circulating in Italy was made according to the following:

- a light DMU (11 t/axle) with wheel arrangement 1A'-A'1 (two diesel motors, one on each driving wheelset) equipped with tyred wheels [28];
- a heavy diesel locomotive (20 t/axle) with wheel arrangement B'B' (one diesel motor with fully hydraulic transmission with cardan shafts and gearboxes) equipped with monobloc wheels [29];
- an average mass electric locomotive (17.2 t/axle) with wheel arrangement B'B'B' with one DC electric motor per bogie and chopper converter, equipped with tyred wheels [30];
- a heavy electric locomotive (21 t/axle) with wheel arrangement B_0'-B_0', AC three-phase converter with asynchronous motors, equipped with monobloc wheels [31].

The reason for this choice is linked to two considerations:

- on coupled axles locomotives, chatter phenomena (a.k.a. *broutage*) may lead temporarily to extra loads on the transmissions. In this case it is prudent to adopt a *conventional wheel-rail friction coefficient* of μ=0.6;
- electric locomotive may suffer from short circuit at the motors, of limited duration but significantly higher torque. It is usual in this case to adopt a *conventional wheel-rail friction coefficient* of μ=0.8.

About starting torque, a limiting value of μ=0.33 was adopted according to the classical equation by Curtius & Kniffler [32]. The calculation of the pressure at the tyre-wheel centre interface was based only on tangential (circumferential or traction) forces as with the design proposed in [2] the coupling surface does not need to bear lateral loads by friction as the *dovetail coupling* ensures the absence of any lateral motion by means of *positive interaction of tapered surfaces*.

A summary of vehicle properties and the *equivalent wheel-rail friction coefficient* is shown in Table 1.

When considering the force that can be transmitted at the wheel-rail contact, the vertical load acting on the wheel cannot be neglected. Let us consider a sufficiently short portion of the tyre of thickness t to be considered as a sort of "third body" inserted between the tyre and the rail (Figure 3).

Table 1: vehicles considered for wheel centre-tyre interface pressure calculation

Vehicle type	Tyred wheel geometric properties					Axleload	Wheel load	Friction	Broutage	Electric overload	Equivalent friction
	D_{new} [mm]	D_{worn} [mm]	Fit geometric properties			Q [t/axle]	N [N]	$\mu_{wheel\text{-}rail}$	$\mu_{broutage}$	μ_{ovload}	μ_{eq}
			w_{fit} [mm]	d_{fit} [mm]	A_{fit} [mm^2]						
ALn668	920	840	90	780	220540	11	53955	0.33	0	0	0.33
G1000	1000	920	90	860	243159	20	98100	0.33	0.27	0	0.6
E652	1040	960	90	900	254469	17.16	84170	0.33	0.27	0.2	0.8
E483	1250	1170	90	1110	313845	21	103005	0.33	0	0.47	0.8

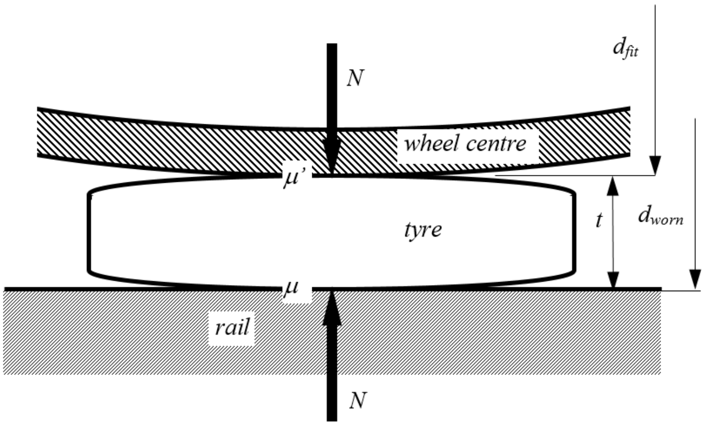


Figure 3. Torque transmissible at the tyre-rail and at the wheel centre-tyre contacts

Resembling the tyre to a "bar of soap" to better understand slipping conditions, as aforementioned the torque T to be considered as an input to the problem is the one *conventionally* calculated at the wheel-rail contact ("dry foot and wet floor") when μ=0.8.

To transmit any given torque at the wheel centre-tyre contact ("dry floor and wet foot") purely by the fitting, a pressure p'_{fit} acting on the surface A is needed according to

$$T' = \mu' p'_{fit} A \frac{d_{fit}}{2}, \tag{1}$$

while a pressure p'_{fitN} according to

$$T' = \mu' \left(N + p'_{fitN} A \right) \frac{d_{fit}}{2} \tag{2}$$

is needed when a vertical load is present. By equating eqns. (1) and (2) it results that

$$p'_{fitN} = p'_{fit} - \frac{N}{A} \tag{3}$$

It is obviously concluded that the presence of an external radial load is beneficial in terms of transmissible torque acting a further load beyond the existing preload. A direct comparison from this point of view is shown in Table 2.

Table 2: pressure requested at the wheel centre-tyre contact to transmit the exceptional torque without and with the forces arising from the wheel load.

	WITHOUT VERTICAL LOAD				WITH VERTICAL LOAD				
Vehicle type	Max torque	Tyre-centre tangential force, friction coefficient and required pressure			Friction coefficients and ratio			Tangential force to be transmitted	Required pressure
	T [Nm]	F [N]	μ'	p_{min} [MPa]	μ	μ'	μ/μ'	N*(μ–μ')	p_{min} [MPa]
ALn668	**7478**	19175	0.3	0.290	0.33	0.3	1.10	1619	0.045
G1000	**27076**	62967	0.3	0.863	0.6	0.3	2.00	29430	0.460
E652	**32321**	71825	0.3	0.941	0.8	0.3	2.67	42085	0.610
E483	**48206**	86858	0.3	0.923	0.8	0.3	2.67	51503	0.594

It can be seen that in the worst case, i.e. the E652 locomotive subjected to both *broutage* and short circuit of electric motors with small diameter wheels, the pressure needed at the wheel centre-tyre contact to transmit the highest torque is around 0.95 MPa, and that this value decreases at 0.61 MPa when the beneficial effect of the vertical load is considered.

In the limit case where the transmissible torque is the same, i.e. *T'*=*T*, this implies

$$\frac{\mu}{\mu'} = \frac{\left(N + p_0 A\right) d_{fit}}{N d_{worn}} \tag{4}$$

from which the needed pressure p can be found as a function of friction coefficients and normal load as

$$p = \frac{N}{A}\left(\frac{\mu}{\mu'}\frac{d_{worn}}{d_{fit}} - 1\right) \tag{5}$$

If $\frac{\mu}{\mu'} = 1$ the pressure becomes very small

$$p = \frac{N}{A}\frac{d_{worn} - d_{fit}}{d_{fit}} \simeq 0 \qquad (6=$$

In a companion paper [1], the authors analysed several existing tyred wheels. In the case of maximum tread wear (final thickness = 30 mm) and with the minimum fit interference, the pressure was 23.9 MPa, a value that is 25 times the one strictly needed when there is no thermal input! It can be seen that radial stiffness of the wheel centre plays a very important role as all the parameters are strongly influenced by it. About the nominal transmissible torque, the minimum value is T_{min}=0.51 MNm = 510000 Nm, i.e. more than ten times the maximum torque that a heavy electric locomotive can output under short circuit failure conditions.

It appears evident that traditional tyre coupling was designed to provide sufficient contact pressure (and therefore sufficient transmissible torque) during drag braking. This led to high (compressive and/or bending) stresses in the wheel centre and high (traction) stresses in the tyre. It looks that a redesign may overcome the limitations intrinsic in the traditional dimensioning, leading to lighter and safer tyred wheel.

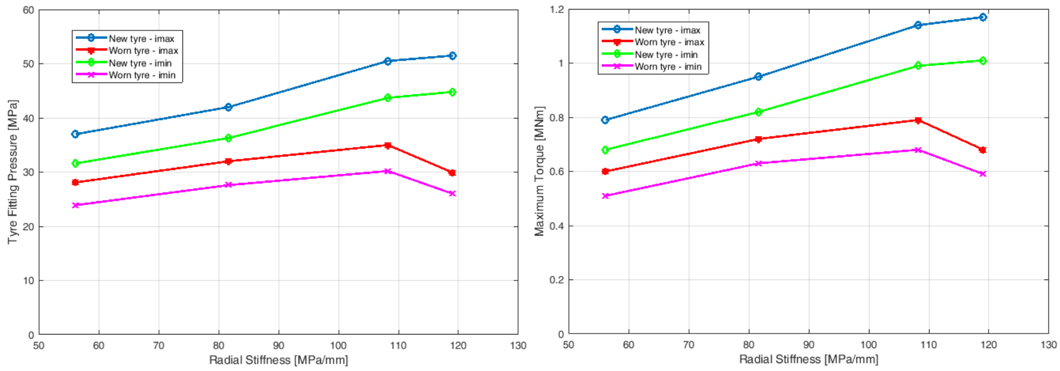


Figure 4. Pressure at the wheel centre-tyre interface (left) and maximum transmissible torque with μ=0.3 for wheel centres of different radial stiffness (new and worn tyres, maximum and minimum fit interference, adapted from [1]).

5.4 Limitations of the current analysis

The analysis shown above has interesting fallouts on the design of disc-braked tyred wheels. The comparison alone of the tyre-wheel centre pressure p_0 is extremely significant as traditional designs result in minimum values of around p_0=20 MPa, while the worst of the analysed cases here leads to less than p_0=1 MPa.

The design of tyres, maximum allowable wear, wheel centre radial elasticity and so on needs to be rethought completely.

Regardless this apparently brilliant results, it should not be forgotten that hardly predictable phenomena, such as fretting, frequently happen on insufficiently pressed hub-shaft couplings such us bearing rings and shafts/hubs. What shown above is a simple evaluation based on equilibrium equation of mechanics and may have a limited relationship with service.

6 Conclusions and further developments

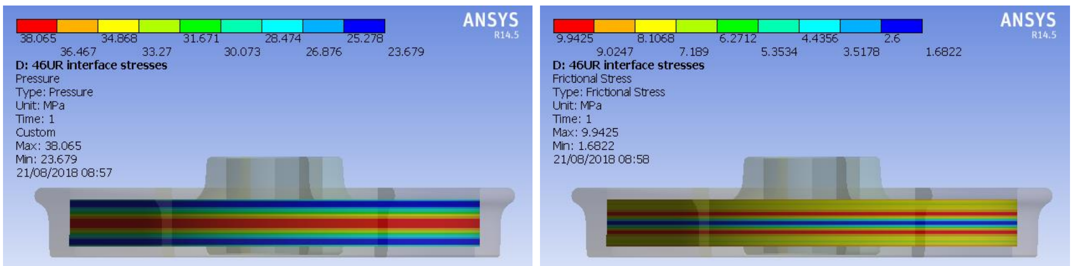
A thorough analysis conducted by using full three-dimensional linear elastic models of tyred wheels has shown that once the thermal input is removed, i.e. braking is applied on discs instead on treads, the conventional tyred wheel has performances that largely exceed the actual needs.

Tyre-wheel centre pressures are 25 times bigger and transmissible torque with fully worn tyres are over 10 times bigger than those needed for purely “cold” traction also in exceptional (failure) cases. This was checked by comparing existing vehicles in worst operating conditions to with “old” vehicles designed according to the UIC leaflets.

A research is in progress in order to evaluate the compatibility of thinner (and lighter) tyres with the existing set of pressures and stresses in the tyre. The phenomenon of fretting, a.k.a. “contact rust”, is a surface deterioration linked to the simultaneous presence of low normal pressure σ, high tangential (surface) stresses τ and the presence of an oxidizing agent (atmosphere oxygen). The key element in the origin of the phenomenon is the presence is some areas, typically the lower loaded ones, where $\sigma/\tau<\mu_{tw}$, i.e. local sliding is possible where the tyre – wheel centre friction coefficient μ_{tw} is exceeded.

Figure 5 shows normal ($p=\sigma$) and frictional stresses (τ) deriving from tyre fitting, highlighting the presence of “bands” linked to the continuously varying radial stiffness of the wheel centre and of the tyre, obviously considered here at the maximum allowable wear. This proves that the founding hypothesis of [25] that any vertical cross-section of the wheel is equally representative of the actual situation is totally wrong.

The same figure also shows normal pressure and frictional stresses during the application of the vertical load on the wheel and of the maximum torque corresponding to the global spinning/sliding of the wheel when the wheel- rail friction coefficient μ_{wt} is exceeded.



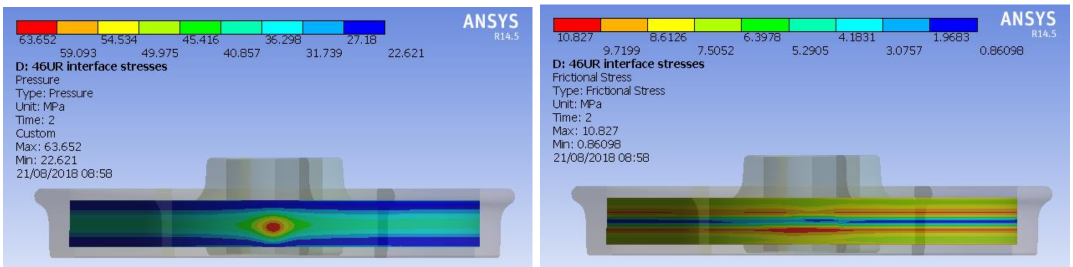


Figure 5. Top: pressure at the wheel centre-tyre interface (left) and frictional stresses (right) for μ_{tw} =0.3 for a freight wagon wheel with the tyred fitted at maximum interference (1.106 mm). Bottom: the same with vertical load (100 kN) and maximum torque (14795 Nm).

The analysis of tyre-wheel centre contact may give considerable insight in the lowly-investigated interface phenomena. Preliminary results show that substantial reductions in the fitting interference and/or the tyre thickness may be obtained without giving rise to either tyre spinning or fretting phenomena.

Another step in the research in the improvement in the overall performances of the wheel centre-tyre pair. Wheel centres can be made lighter when the mounting pressure is reduced, and fully worn tyres may be thinner for the same reason. The ultimate goal is to produce a tyred wheel that is as heavy as (or as light as) the competing monobloc wheel.

The authors have already performed a re-design of the wheel centre based on different manufacturing approaches and using materials that are widely used in other sectors but that are not common (yet) in the railway industry. An abstract has been submitted to the next International Wheelset Congress [33] where a casted wheel centre made of Austempered Ductile Iron (ADI) is described. It is 50 kg lighter than the conventional forged and rolled wheel centre; combining this research with the studies about the residual thickness of the tyre that can be made smaller than today (30 mm), the overall mass should reduce consistently.

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