Thermal and stress analysis of a railway wheel-rail rolling-sliding contact

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1. Introduction

The history of the locomotive-hauled trains began in the early 19th century. These two centuries, and especially the recent years, saw dynamic advances in the different fields of railway engineering.

From among the ordinary surface transportation forms, railway transportation is the safest and has the smallest energy and space demand, compared to its competitors. Thus it provides an economical solution in both freight and passenger transportation. For long distances (500-600 km), very high speed trains (250-350 km/h) are a possible choice instead of aviation [1].

Development resulted in faster travelling speeds as well as better train safety systems. In addition to tread brake systems, disc brake and electromagnetic rail brake systems also appeared. Furthermore – like in conventional road vehicles – anti skid devices, so-called WSP Systems (Wheel Slide Protection System), are applied also in railway vehicles.

Owing to these developments and others not mentioned above, rail transport has become one of the safest forms of transportation worldwide. However, these advances, beside their positive effects, also brought ever newer problems which researchers and experts had to solve.

In my Ph.D. booklet of Theses, I analysed the thermal, frictional and stress development phenomena when rolling–sliding contact occurs in a wheel–rail system, apt to cause surface damage to the wheel tread during disc braking.

1.1. The objective of my research

During the intensive wheel–braking process of railway vehicles, macroscopic sliding can occur on the wheel–rail contact patch under inappropriate adhesion conditions. Although macroscopic sliding speed is restricted, it is deliberately not eliminated by the Wheel Slide Protection system (WSP). Consequently, considerable heat is generated and enters the friction partner components in the case of a rolling–sliding contact process. This phenomenon may cause microthermal cracks on the surface of the wheel tread (Figure 1).

In my research, I analysed a new (wear–free) wheel–rail contact condition and looked for answers to the following main questions:

- What is the distribution of the contact pressure in the wheel–rail contact?
- How high is the peak temperature during rolling–sliding motion over a given time interval?
- What is the temperature distribution on and under the wheel tread/rail head during the first revolutions/pass?
– What is the effect of temperature on stress distribution? What is the thermal stress process and the distribution of thermal stress?
– What is the combined effect of contact and thermal stresses in thermal cracking phenomena?

To find the answers to these questions, I first compiled a literature overview to see the latest results in this special field.

2. Literature overview

Thermal phenomena are a very important research field of railway technology. They appear not only during the braking process (tread braking, disc braking) but also when macrosliding occurs in the wheel-rail contact. These frictional sliding contacts generate huge amounts of heat, causing thermal cracks and other defects on the sliding surfaces.

The pioneer of flash temperature investigation was Harmen Blok [3] who for the first time examined and defined – for a semi-infinite body, using a uniform square heat source – the maximum surface temperature rise at different (low and high) Peclet numbers.

Gupta et al. [4] created two- and three-dimensional finite element models to investigate the temperature rise and distribution on and under a locomotive wheel tread for different combinations of creep and adhesion. The FE model they developed can be seen in Figure 2. Combining the models they could examine the maximum temperature after one hour rolling-sliding contact (for a wheel load of 147 kN). They presented a detailed diagram which shows the results of the 3D movement and a special sequentially distributed heat source model which shows the temperature distribution on the wheel tread surface during the first five revolutions (Figure 3). Furthermore, they compared the temperature distribution under the surface at the end of the first five revolutions. The results show only a little deviation between the results
of the different two- and three-dimensional models. Finally, for the cases examined, they computed the heat flow rate partitioning factor as well.

Sábitz and Zobory [5] used FEM to investigate temperature distribution and martensite formation on the surface of the wheel tread both for tread braking and disc braking. They pointed out that “in case of block-braking-free wheel-rail contact phenomenon with the considered operation parameters, the maximum temperature value and the absolute value of the cooling rate were sufficient for the occurrence of martensite formation”. Sábitz and Kolonits [6] also used FEM to examine the temperature distribution on the wheel tread, and compared the results with those of an analytical solution.

Knothe and Liebelt [7] used Laplace transforms combined with the method of Green’s functions to analyze the contact temperature and temperature fields of components in relative sliding motion. To determine the temperature field in a wheel–rail contact, they reduced the three-dimensional problem, as an approximation, to a two dimensional one. Besides the temperatures on ideal smooth surfaces, derived from pressure distribution, they analyzed the effects of roughness and surface defects. Furthermore, the effect of constant and ellipsoidal loadings was investigated. They concluded that in the case of the examples examined, surface damages (e.g. cracks) have a higher influence on temperature rise than rough surfaces which have little impact.

Peng et al. in [8], [9] used a 3D non-linear finite element model to evaluate the thermal stresses, allowing for “rail chill” effects (a hot wheel passing over a cold rail) in the case of a tread braked wheel. They examined two different braking processes (stop and drag braking). The analyses consisted of three different stages. During the first stage, thermal stresses were evaluated considering the rail chill effect
under different braking processes. In the second stage, semi-analytical computation was used to determine the stress intensity factor of thermal cracks. During computations in the third stage, the Frost-Dugdale approach was used to model thermal fatigue crack growth. As a result of the analyses, they concluded that rail chill had an influence of approximately 10% on the maximum temperature for both braking processes. It was also concluded that rail chill caused significant difference in fatigue life, computed both for stop and drag braking.

Similarly to Peng et al., Moyar and Stone in the early 90ies [10] examined the mechanism of thermal defects of the wheel considering the rail chill effect using critical plane fatigue initiation theory (this theory uses the variation of shear and normal stress components on fixed planes during critical plane fatigue analysis).

Sábitz and Zobory [11] used FEM to analyze the temperature and thermal stress distribution in the case of a tread braked wheel. They analyzed not only the temperature distribution during a complete passenger route between Budapest and Hegyeshalom (188 km) but also the tangential stress distribution on the tread.

In addition to contact and temperature studies, a number of investigations are available in print and online that use numerical and experimental methods to deal with the mechanism of wheel-rail defects.

Handa et al. [12] used an experimental test equipment to examine how tread thermal cracks occur and propagate. The analysis involved not only the wheel–rail contact but also the tread breaking process. Figure 4 illustrates the distribution of tread thermal cracks on and under the running surface of the test wheel. As a result of the measurement, they could make the following statements. Tread thermal cracks appeared on the surface of the tread less than 100 times of braking cycles and less than 106 times of rolling contact under a contact load of 60 kN. These cracks are generated and propagated in the inner region of the wheel–rail contact width (Figure 4), however, the contact width with the brake block was wider. This means that both cyclic thermal load and rolling contact are necessary for thermal crack generation. “Accordingly, tread thermal cracks can be defined as an idiosyncratic phenomenon affected by interactions of low cycle thermal fatigue and rolling contact fatigue”. Finally, they detected the tensile residual stress in the range of 100 to 500 MPa for depths of up to 500 μm.

Handa and Morimoto [2] used the same test apparatus as in [12]. The aim of the analysis was to examine the dominating factor of thermal crack development and the possible countermeasures. Residual stress distribution was also examined by numerical analysis based on the temperature conditions computed and predicted by FEM and measurement. Figure 5 shows the locations of the tread thermal cracks that were highlighted with penetrant testing. As a result of their investigation they could make the following important statements. The alternating loading caused by tread braking and the additional tangential force imposed by the rolling contact induced tread thermal cracks. “The main cause of tread thermal cracking is considered to be residual..."
stress and wheel-rail tangential force”. Last but not least the Hertzian contact pressure has a very small influence on thermal cracking.

3. Tools and methods used during the research

During my research to model and investigate the theoretical background of thermal wheel surface damage development – after the detailed analysis of the available methods – the FEM method was used. Taking into consideration that the problem under examination is so complicated, the problem has been divided into several partial tasks with several partial FE models. The analysis of the contact and the frictional state of a wheel–rail connection was studied, and the heat and the thermal stress generation between them were examined separately. To validate the new FE method, analytical methods described above were used. In the case of contact analysis, the results are compared with the Hertz theory. The thermal computation validation is based on Tian & Kennedy’s analytical solution. Some of the results were compared with experimental data and measurement results from the literature.

The schematic structure of my segmented computation method and validation processes can be seen in Figure 6. During the research, the ANSYS Workbench 14.5 software was used.
In the course of the investigations, a 14° piece of a railway wheel of D=920 mm diameter with simplified geometry was examined. During the computation, the flange of the wheel rim and the conical shape of the tread were neglected. Since the geometries thus established are totally symmetrical to the Y–Z plane, half models were produced from the pieces extracted for computations.

The segment of the analyses was subdivided into sub-segments. This was intended to produce a sufficiently fine mesh. For the sake of higher computation accuracy, 20-node hexahedron elements were used for each segment. The element size was 5-10 mm for the base body; and 1 mm for the segments marked (C); in the case of segments (B) and (A), element size was reduced to 0.4 mm in tangential direction,
and in segment (A) was broken down to further 5 elements in the direction of depth, consequently the element height was 0.1 mm here. Thus the FE mesh consisted of 37,108 elements and 178,311 nodes. The sub-segments are shown in Figure 7.

![Figure 7 The structure of the FE mesh](image)

In the course of the computations, three sample lines were specified, along which results were sampled at four different instants of time at each revolution for both the wheel and the rail. Figure 8 shows the location of sample lines on the wheel (on the rail, the sample lines are at the same location), and Figure 9 illustrates the positions of the heat sources at the instant of each sample (for 5 revolutions) on the plan view of the analysed zone.

In the first part of the analysis, the contact normal stress distribution of the contact between wheel and rail was examined and was validated with an analytical Hertz computation. In addition to the contact normal stress, the von Mises stress distribution on and under the contact surfaces was also analyzed. According to the results, the maximum of the von Mises stress and the principal shear stress occurs under the wheel tread at 2–4 mm depth (under the studied conditions). The second part of the analysis described the investigation of the initial stick–slip phenomenon. The results show that the initial sticking zone within the contact zone decreases after just ~0.0015 mm tangential displacement of the wheel, and full sliding occurs at higher tangential displacement.

Related publications: [14]; [15]; [16]; [17]; [18]; [19]; [20].
Figure 8 Location of sample lines used for the evaluation of results on the test model:
St according to the direction of sliding, Sd in direction of depth,
Sp in direction across the tread

Figure 9 Positions of the moving heat source/contact pressure at the moments of sample,
with the corresponding position and time date for the 1st revolutions (the highlighted s coordinates indicate the centre of the current position of the moving heat source and the pressure)
3. Theses

**Thesis 1:** A transient thermal FE algorithm was used to analyze the thermal state of the rolling-sliding wheel–rail connection during disc braking, using moving and distributed heat source models. Considering the intensive disc braking of high-speed trains, I determined, in the higher (15%) slip range and assuming the presence of a Wheel Slide Protection System (WSP):

   a) The high local temperature peak and its longer „tail” generated already during moderate sliding friction conditions ($\mu=0.15$). This temperature peak slightly increases with each revolution, mainly on the wheel side. Already after a few revolutions heat conduction causes a heat increase also directly under the surface in depth direction and is „accumulated” over time. Temperature decreases to approximately one third at about 30 mm from the contact zone.

   b) The temperature peak and the base temperature were determined during my computations of the complete disc braking process, and it was pointed out that after the very first phase, the initial quickly rising contact temperature reversed to a decreasing tendency, caused by the changing heat partition and the effect of „rail chill” (Figure 10).

*Related publications:* [21]; [22]; [23].

![Figure 10](image-url)  
*Figure 10* Surface temperature on the wheel’s and rail’s thread under the braking process (the vertical dashed line represents the beginning of the slipping braking period, the dotted line represents the first 5 revolutions)
Thesis 2: I used transient thermal-elastic-plastic FE models and solutions to examine the effect of heat generated in a rolling-sliding wheel–rail contact. I determined that:

a) During one cycle of the heat flux, at first, high compressive stresses are generated at the surface of the wheel in tangential direction, then they decrease, and at the next instant, increasing tensile stress appears as the heat flux leaves the region.

b) Below the surface (at a depth of about 50-100 μm) the representing thermal stresses remain compressive stresses, while their magnitudes decrease and move in deeper layers as the heat flux moves away from the region studied. During subsequent revolutions, subsurface stress maxima will be slightly higher due to the „accumulation” of thermal stresses as obtained from the elastic-plastic computations. During the cooling down phase, residual stresses are generated, mainly on the surface and in near-surface layers (Figure 11).

Related publications: [22]; [23]; [24].

Figure 11 Tangential stress component (σ_t) distribution on and under the wheel tread in the intersection of the Sd-St-Sp sample lines as a function of time until the first quarter revolution.
**Thesis 3:** I studied the joint effect of contact and thermal stresses using transient thermal-elastic-plastic FE models. I determined that:

a) The coupled stress state caused plastic behaviour in deeper layers compared to the “pure” thermal stress computations. The former stress maxima migrated from the surface to deeper layers (0.2-0.3 mm under the surface (Figure 12)).

b) On the surface, alternating compressive and tensile stresses followed each other in accordance with the warming and cooling of the tread.

c) During the unloading phase I determined the residual stresses and residual strains which characterize the near room temperature state.

*Related publications: [20]; [25].*

![Figure 12 Tangential stress component distribution on and under the wheel tread in the intersection of the Sd-St-Sp sample lines as a function of time until the first quarter revolution](image)
5. Bibliography


