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# Contact differences between real wheel-rail and test benches

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## Abstract

The sliding velocity of a wheel travelling through a rail is the main responsible for wear. This phenomenon changes wheel and rail profiles influencing vehicle dynamics. For this reason, an optimized grinding schedule, which gets back the original profiles, plays a crucial role to ensure an economically reasonable rail life cycle while assuring running safety. Predictive methodologies for wear damage help to plan the maintenance schedule. These methodologies are based on known wear rates. As an on-site characterization of wear mechanism is difficult, test rigs are widely used. Twin-disc machines and scaled test-benches are the most common. A comparative study of the contact patch is presented under the equivalent conditions that are given in the wheel-rail interaction. For that purpose, the normal and tangential problems are solved for elliptical and rectangular contacts with their pertinent formulation. The contact conditions of a twin-disc machine differ with respect to real wheel-rail contact, in area shape and slip area ratio. This not occurs on the scaled bench, where the shape of the contact area and the slip area ratio remains identical. As the slip between different surfaces is the main responsible of wear damage, these observed contact differences regarding the slipping area, disrupt the wear rates when are directly applied from twin-disc machines.

**Keywords:** Contact, Tribology, Creepages, Scaled test-bench, Twin-disc, Wear

# 1 Introduction

The wear damage has a significant impact on the Life Cycle Cost (LCC) of a railway system maintenance. The sliding velocity of a wheel travelling through rail is the main responsible for wear. This phenomenon changes wheel and rail profiles influencing vehicle dynamics. For this reason, an optimized grinding schedule, which gets back the original profiles, plays a crucial role to ensure an economically reasonable rail life cycle while assuring running safety.

Predictive methodologies for wear damage help to plan the maintenance schedule. These methodologies are based on known wear rates [1,2]. As an on-site characterization of wear mechanism is difficult, test rigs are widely used. In this way, different contact conditions under controlled parameters can be characterized. Twin-disc machines and scaled test-benches are the most common.

On the one hand, twin-disc tests have some advantages as tests are carried out fast and specimens can be obtained easily. Nevertheless, the geometry of contact bodies differ from real wheel-rail contact conditions and only longitudinal creepage is considered. On the other hand, scaled test-benches allow to experimentally characterizing wheel-rail contact problem, and if necessary, considering the influence of lateral creepage and spin [3]. The disadvantage in these test-benches is to replicate the manufacturing process in a scaled wheel and rail.

A comparative study of the contact patch is presented under the equivalent conditions that are given in the wheel-rail interaction.

# 2 Methods

Generally, two steps are followed to solve the wheel-rail contact problem. These are the solution of normal and tangential problems. The contact problem can be divided into these two steps owing to the quasi identity assumption [4]. Friction forces between wheel and rail have not enough influence on the size of the contact patch and pressure distribution.

The normal problem allows calculating the shape and size of the contact area and the normal pressure at each point. This is carried out with the Hertz theory [5] under several assumptions. The relevant material parameters, as this is an elastic assessment, are Young's module $E$ and Poisson's ratio $v$. To calculate the semi-axis dimensions (a and b) for a given normal load $W$, the equivalent longitudinal and lateral curvatures are necessary along with the ellipticity parameter $ke$ and elliptic integral E as stated in [6]. The equations for Hertzian elliptical contact are:

- Equivalent elasticity module:

$$E' = \frac{2}{\frac{(1-v_1^2)}{E_1} + \frac{(1-v_2^2)}{E_2}} \tag{1}$$

- Reduced radius:

$$\frac{1}{R'} = \frac{1}{R_x} + \frac{1}{R_y} \tag{2}$$

$$\frac{1}{R_x} = \frac{1}{r_{1x}} + \frac{1}{r_{2x}} \tag{3}$$

$$\frac{1}{R_y} = \frac{1}{r_{1y}} + \frac{1}{r_{2y}} \tag{4}$$

- Contact dimensions:

$$a = \sqrt[3]{\frac{6\mathcal{E}WR}{\pi k_e^2 E'}} \tag{5}$$

$$b = \sqrt[3]{\frac{6k_e^2\mathcal{E}WR}{\pi E'}} \tag{6}$$

- Contact pressure:

$$p_0 = \frac{3W}{2\pi ab} \tag{7}$$

$$p_{avg} = \frac{W}{\pi ab} \tag{8}$$

$$p(x,y) = p_0 \sqrt[2]{1 - \frac{x^2}{a^2} + \frac{y^2}{b^2}} \tag{9}$$

For twin-disc specimens, flat surfaces at the lateral direction are used. This shape change leads to a restatement of Hertz formulas, where lateral dimension $2b$ takes the value of discs' width. This rectangular area has a semi-ellipse prismatic pressure distribution that can be calculated according to the following formulas:

- Contact dimensions:

$$a = \sqrt[2]{\frac{8(W/2b)R'}{\pi E'}} \tag{10}$$

- Contact pressure:

$$p_0 = \frac{2(W/2b)}{\pi a} \tag{11}$$

$$p_{avg} = \frac{W/2b}{2a} \tag{12}$$

$$p(y) = p_0 \sqrt[2]{1 - \frac{y^2}{b^2}} \tag{13}$$

Then, the tangential problem is used to calculating the shear stresses and the relative local slips. Kalker's contact theory [7,8] is used for this assessment. The creepage coefficients are depending on Poisson's ratio and the ratio of the contact axes. These coefficients are tabulated. The compliant parameters ($L_1$, $L_2$ and $L_3$) depend on the material, geometric parameters and Kalker coefficients.

## 3 Results

Three different assessments have been carried out under equivalent contact conditions:

1. Wheel/rail contact.
2. Scaled test-bench.
3. Twin-disc contact.

From the resolution of the normal problem different shapes are obtained, an elliptical one for the wheel-rail and scaled-bench cases and another rectangular for the twin-disc. The normal pressure distribution has a semi-ellipsoid shape for elliptical contacts while for a rectangular area the pressure distribution has semi-elliptical prism shape. The shear stresses are dependent on creepage values, getting a scaled shape regarding normal pressure for saturation conditions. While for elliptical contacts the maximum values are punctual, in the case of rectangular contact the maximum value remain along the wide of discs, in the centre line of the contact area.

The figures below (Figure. 1, Figure. 2 and Figure. 3) show the results obtained for a longitudinal creepage of 0.2%, without considering lateral creepage and spin values. The used friction coefficient at saturation $\mu o$ is 0.3 for all cases. Table. 1 shows the complete results.

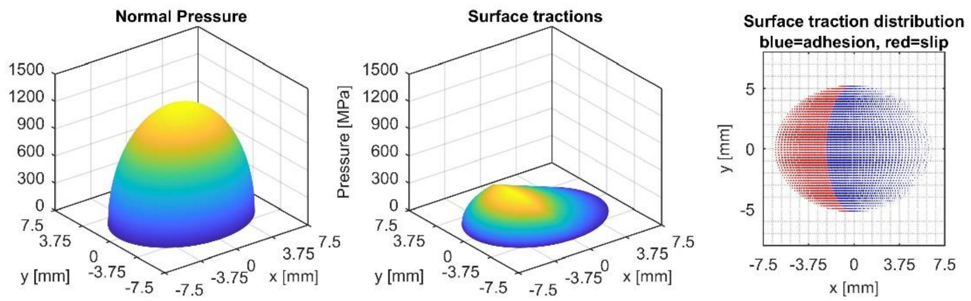


Figure. 1. Wheel-rail contact

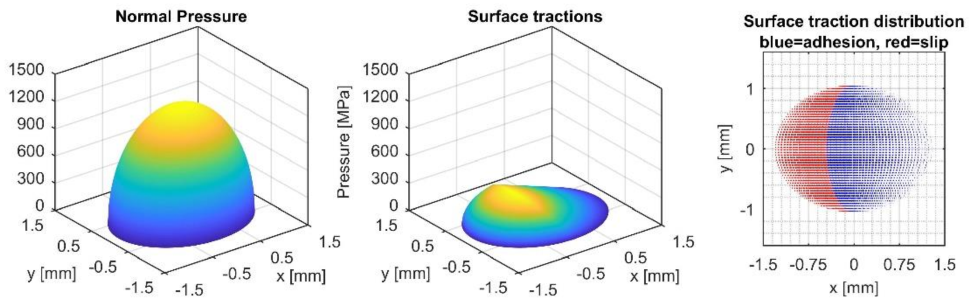


Figure. 2. Scaled test-bench contact

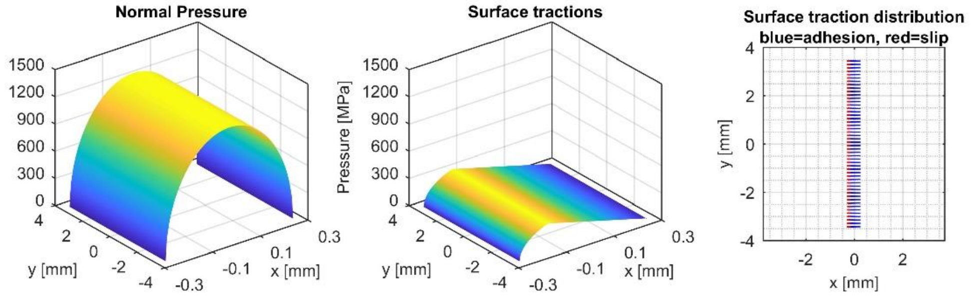

Figure. 3. Twin-disc contact

For the scaled-bench, dimensional results are related to scaling factor (1/5) achieving the same average and maximum pressure values and $A_{slip}/A_{total}$ ratio. In the case of twin-disc machine, only the maximum pressure $p_0$ is equal with a different $A_{slip}/A_{total}$ ratio. To obtain the same ratios, creepages must be enough high to saturate the friction coefficient where the slip and total areas will be equal.

| Contact | Wheel/Rail | Scaled-bench | Twin-disc |
|---|---|---|---|
| $r_{1x}$ wheel (mm) | 425.0 | 170.0 | 21.5 |
| $r_{2x}$ rail (mm) | inf | 170.0 | 30.0 |
| $r_{1y}$ wheel (mm) | inf | inf | Inf |
| $r_{2y}$ rail (mm) | 300.0 | 60.0 | inf |
| $E$ (GPa) | 213 | 213 | 213 |
| $v$ (-) | 0.29 | 0.29 | 0.29 |
| $W$ (kN) | 90.000 | 3.600 | 3.648 |
| Shape | Ellipse | Ellipse | Rectangle |
| $a$ (mm) | 6.574 | 1.315 | 0.267 |
| $b$ (mm) | 5.267 | 1.053 | 3.5 |
| $A_{total}$ (mm²) | 108.775 | 4.3510 | 3.743 |
| $p_{avg}$ (GPa) | 0.827 | 0.827 | 0.975 |
| $p_0$ (GPa) | 1.24 | 1.24 | 1.24 |
| $\mu_o$ (-) | 0.30 | 0.30 | 0.30 |
| $\xi$ (-) | 0.2% | 0.2% | 0.2% |
| $A_{slip}/A_{total}$ (-) | 0.34 | 0.34 | 0.22 |

Table. 1: Contact type differences.

## 4 Conclusions and Contributions

In the present work, three different types of contact have been assessed. Firstly, a real wheel and rail contact has been analysed and then, the contact from both scaled test-bench and a twin-disc machine. For that purpose, the normal and tangential problems are solved for elliptical and rectangular contacts with their pertinent formulation.

Once the real case is solved, the comparisons with the scaled-bench and the twin-disc machine show different conclusions. The contact conditions of a twin-disc machine differ with respect to real wheel-rail contact, in area shape and slip area ratio. This not occurs on the scaled bench, where the shape of the contact area (with the scaled dimensions) and the slip area ratio remains identical real wheel and rail contact conditions.

As the slip between different surfaces is the main responsible of wear damage, these observed contact differences regarding the slipping area, disrupt the wear rates when are directly applied from twin-disc machines. The wear rates per contact area only can be applied directly with saturated friction coefficients, which are obtained for high values of creepages that unusually take place at real wheel-rail contact. Therefore, newer approaches are needed considering the differences between tested conditions and the assessed real case.

## Acknowledgements

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