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A model for investigating the influence of road surface texture and tyre tread pattern on rolling resistance

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ABSTRACT

The reduction of rolling resistance is essential for a more environmentally friendly road transportation sector. Both tyre and road design can be utilised to reduce rolling resistance.

In both cases a reliable simulation tool is needed which is able to quantify the influence of design parameters on the rolling resistance of a tyre rolling on a specific road surface. In this work a previously developed tyre/road interaction model is extended to account for different tread patterns and for losses due to small-scale tread deformation. Calculated contact forces and tyre vibrations for tyre/road interaction under steady-state rolling are used to predict rolling losses in the tyre. Rolling resistance is calculated for a series of different tyre/road combinations. Results are compared with rolling resistance measurements. The agreement between simulations and measurements is generally very good. It is found that both the tyre structure and small-scale tread deformations contribute to the rolling losses. The small-scale contribution depends mainly on the road roughness profile. The mean profile depth of the road surface is identified to correlate very well with the rolling resistance. Additional calculations are performed for non-traditional rubberised road surfaces, however, with mixed results. This possibly indicates the existence of additional loss mechanisms for these surfaces.

1. Introduction

In 2006 23% of the CO_2 emissions in the European Union were due to the fuel consumption of the road transportation sector. For cars powered by classical combustion engines, only about 10% to 40% of the chemical energy stored in the fuel is available as mechanical energy from the engine. The other part is lost in the form of waste heat because of engine inefficiency [1]. The available energy is consumed by aerodynamic drag, rolling resistance and acceleration. Depending on driving conditions, hysteretic losses in the tyres, i.e. rolling resistance, eventually account for 5% to 30% of the fuel consumption of a typical passenger car [1]. These figures are even higher for trucks and other heavy vehicles, ranging from 15% to 40%. Accordingly, there is a large potential to reduce a vehicle's overall fuel consumption by decreasing energy losses due to rolling resistance. A 10% to 20% reduction in rolling resistance is for example believed to reduce the fuel consumption within the EU by up to 2.5% for passenger cars and 3.6% for heavy vehicles [2]. Additionally, it has also been shown that the release of local pollutants such as CO and NO_x can be reduced by lower rolling resistances [1].

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Hybrid and electrically powered cars have a greatly improved power train efficiency when compared to vehicles with combustion engines. This implies that for this new type of vehicles the influence of the rolling resistance on the overall energy efficiency is even bigger. To conclude, a reduction of rolling resistance is essential for a more environmentally friendly road transportation sector.

Both tyre and road design can be utilised to reduce rolling resistance. In both cases a reliable model is needed which is able to quantify the influence of design parameters on the resulting rolling resistance. Obviously, it is desirable to be able to use the same model for road and tyre design purposes.

Numerous examples of calculations of rolling resistance are available in the literature (see [3] for an overview). Besides some approaches partly based on measured data [4,5], most of them follow the same conceptual approach of using a steady-state rolling tyre model to calculate rolling resistance either as a drag force or based on energy dissipation.

A stationary tension band on an elastic foundation was used by Stutts and Soedel [6] to model the tyre; rolling resistance was then calculated from the deflection in the contact zone. In [7] hysteretic energy dissipation is determined from a model of an elastic ring supported by a viscoelastic foundation. Several other analytical methods for computing rolling resistance, including a flexible ring model, are compared in [8]. All these tyre models, however, are geometrically over-simplified, limiting their usefulness for tyre design purposes.

Finite element modelling (FEM) of the tyre is another very common approach, often combining mechanical and thermal models. Typically, the heat generation rate is calculated from a viscoelastic model of a free rolling tyre. Thermal simulations give the rolling resistance from the hysteresis. Models following this approach are e.g. [9–12]. In a more recent work this approach was used by Cho et al. [13] to investigate the influence of different tread patterns on rolling resistance. A slightly different technique was used by Ghosh et al. [14], who calculated the dissipated energy directly out of the strain energy density distribution in a steady-state rolling FE model. Another method which directly computes the energy loss during the mechanical FEM process was introduced as the *directional incremental hysteretic model* in [15]. Recently, Ali et al. [16] calculated the rolling resistance of a heavy truck tyre rolling on different non-smooth surfaces from reaction forces obtained by FE simulations. As the road surface profiles were artificially created no comparison to measurements could be made. The advantage of FEM approaches is that it is usually possible to use commercial software, thus reducing the implementation effort. The models are also rather geometrically detailed, allowing the extraction of information on the distribution of dissipation inside the tyre, thus aiding in the design process.

A general problem with all mentioned methods is the contact implementation. The used tyre/road interaction models are usually limited to the large scale deformations caused by static loading of a tyre rolling under steady-state conditions on a smooth road. Even in the few cases where non-smooth surfaces are considered the contact resolution is very coarse: in [16], for example, contact is calculated based on a tyre circumference divided into 60 segments. For the considered heavy truck tyre this corresponds to a contact patch size of 6 cm or more. Accordingly, the small scale variations of road roughness and the complex interrelation between tyre deformation and contact forces, which are both essential for the correct modelling of tyre vibrations, are typically ignored in FE simulations. Recently, Boere et al. [17] proposed a two-step process to overcome this limitation. It superposes a dynamic local tread/road interaction on the large scale deformations obtained from FEM. Rolling resistance is calculated by means of input power into the system. Comparison of results to measurements showed that while rolling resistance was generally overestimated, correct tendencies for different road surfaces were obtained. A limitation of the implementation is that only 2D road roughness data, i.e. without lateral variation, is considered, and that there is no interaction between the tread/road and large-scale-deformation stages.

Another approach has been used by Fraggstedt [18]. He calculated the vibrations and rolling losses of a rolling tyre with a waveguide finite element model (WFEM). Frequency and wave order distributions were shown together with individual element contributions to the overall dissipation. However, the deviations between simulated and measured rolling resistances were slightly higher than for some of the other mentioned methods. Furthermore, only a few different road surfaces were considered and a detailed analysis of the results was missing.

An extension of the approaches suggested by Fraggstedt and Boere et al. is used in the work presented in this paper. Rolling resistance can either be defined as the drag force F_R acting in opposite direction to the rolling of the tyre, or, as is done in ISO 18164:2005 [19], as the dissipated energy E_{diss} over the distance of rolling L. Normalising with the total averaged radial contact force F_c, the rolling resistance coefficient C_r for steady-state rolling at speed V is obtained as

$$C_r = \frac{F_R}{F_c} = \frac{E_{\text{diss}}}{F_c L} = \frac{P_{\text{diss}}}{F_c V}, \tag{1}$$

where P_{diss} is the dissipated power. Due to conservation of energy, and under the assumption that frictional losses can be ignored for steady-state rolling, the power input into the tyre through the contact, P_{in}, is identical to the lost power, i.e. P_{diss}. For a discrete description of the lateral contact position the time-average input power for steady-state rolling is given as

$$P_{\text{in}} = \frac{1}{T} \sum_i \int_0^{2\pi} \int_0^T F_i(t,\theta) \frac{\partial v_i(t,\theta)}{\partial t} \, \mathrm{d}t \, \mathrm{d}\theta. \tag{2}$$

Herein, F_i and $\partial v_i / \partial t$ denote the radial contact force and radial tyre velocity for the lateral contact track i, T is the evaluation time, and θ is the circumferential angle.

This means that the rolling resistance can be expressed in terms of the time-varying contact forces and tyre deformations as given by a dynamic tyre/road interaction model typically used for tyre/road noise calculations. The use of such a model