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Empirical tire temperature- and rolling resistance model

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Executive Summary

Tire rolling resistance (RR) is one of the main energy sinks for electric passenger vehicles, having a large impact on driving range and lifecycle electricity consumption. Accurate predictions of range and consumption in non-ideal conditions require the ability to model tire RR in a broad range of temperatures and for any given drivecycle. In this paper we describe a recently developed empirical tire RR model which is based on the formulation of Schuring et al. [1] combined with a simple lumped thermal model for the tire temperature. The model is fitted to measurement data obtained at the ISO 28580-compliant tire lab at Volvo Cars, and simulation results are compared to complete-vehicle drivecycle measurements performed in one of our chassis dynamometers. The simulated temperature and RR are found to be in good agreement with measurements.

Keywords: Electric Vehicles, Modelling & Simulation, Digital twin design tools, Energy management, Chassis systems for EVs

1 Introduction

Tire rolling resistance (RR) is one of the dominating energy sinks for an electric vehicle (EV), impacting driving range and lifecycle electricity consumption. Range and energy consumption in non-ideal conditions such as driving in cold climate have a large impact on overall customer satisfaction and are gaining increased attention both from rating agencies and legislators [2, 3]. Both RR and aerodynamic drag have a significant temperature-dependence and tend to grow with lower temperatures, exacerbating the range loss already occurring due to active heating for e.g. cabin comfort. Being able to accurately predict total vehicle energy consumption and range in these non-ideal conditions is of high importance to guide product development and to provide relevant information to the driver. Here, tire RR is one of the most important aspects to consider, highlighting the need for general and accurate tire RR models.

2 Model and method

2.1 Tire RR and temperature model

Schuring et al. [1] found that the speed- and temperature dependence of the RR force of passenger car tires can to good approximation be written in the form

$$F_{\mathrm{RR}} = (A_0 + A_1 v)e^{-(B_0+B_1 v)T_{\mathrm{tire}}} \quad (1)$$

where $A_0, \ldots, B_1$ are constants (for a given tire, load, and capped inflation pressure) and where v and T_{tire} denote vehicle speed and tire temperature respectively. Note our differing sign convention as compared to Schuring. In this approximation it is assumed that the tire temperature at any instant can be represented by a single value T_{tire}, which we in this work take to be the average surface temperature along the inside tire wall profile. We fit the constants in Equation 1 to measurement data at different speeds using tires equipped with internal infrared temperature sensors.

To model the tire temperature, we formulate a simple lumped thermal capacity model as follows (see Figure 1 for an illustration):

$$\frac{dT_{\mathrm{tire}}}{dt} = \frac{\dot{Q}_{\mathrm{IHG}} - \dot{Q}_{\mathrm{ambient}} - \dot{Q}_{\mathrm{road}} - \dot{Q}_{\mathrm{gas}}}{C_{p,\mathrm{tire}} m_{\mathrm{tire}}}. \quad (2)$$

Here, the internal heat generation is given by the RR force and vehicle speed v

$$\dot{Q}_{\mathrm{IHG}} = F_{\mathrm{RR}} v \quad (3)$$

We assume that the heat transfer to ambient air is given by a linear function of v and that it is proportional to the convective area of the tire as well as to the temperature difference w.r.t ambient air:

$$\dot{Q}_{\mathrm{ambient}} = (c + fv) \cdot \left(A_{\mathrm{tire}} - A_{\mathrm{contact\ patch}}\right) \cdot (T_{\mathrm{tire}} - T_{\mathrm{amb}}). \quad (4)$$

The heat transfer to road is assumed to be of the form

$$\dot{Q}_{\mathrm{road}} = k \cdot A_{\mathrm{contact\ patch}} \cdot (T_{\mathrm{tire}} - T_{\mathrm{road}}) \quad (5)$$

and the heat transfer to inflation gas is assumed to be

$$\dot{Q}_{\mathrm{gas}} = H_{\mathrm{gas}} \cdot \left(T_{\mathrm{tire}} - T_{\mathrm{gas}}\right). \quad (6)$$

The heat transfer coefficient from tire to inflation gas H_{gas} as well as the heat capacity $C_{p,\mathrm{tire}}$ is taken from literature [5], and we assume that the heat transfer from the tire to the rim is negligible since their contact area is small.

In this work we employ the simplifying assumption that the temperature of ambient air and that of the road are equal, $T_{\mathrm{amb}} = T_{\mathrm{road}}$. This means that Equations 4 and 5 can be combined into a speed-independent $(\dot{Q}_c)$ and a speed-dependent $\left(\dot{Q}_v\right)$ heat transfer to the ambient road+air:

$$\frac{dT_{\mathrm{tire}}}{dt} = \frac{\dot{Q}_{\mathrm{IHG}} - \dot{Q}_c - \dot{Q}_v - \dot{Q}_{\mathrm{gas}}}{C_{p,\mathrm{tire}} m_{\mathrm{tire}}}. \quad (7)$$

Thus, under this simplifying assumption, the thermal model has a convective parameter f and a single lumped (convective + conductive) parameter z, which we identify from Equations 4 and 5 as

$$z = \left(c \cdot \left(A_{\mathrm{tire}} - A_{\mathrm{contact\ patch}}\right) + k \cdot A_{\mathrm{contact\ patch}}\right). \quad (8)$$

Considered together, Equations 1 and 7 define a coupled RR and thermal model for the tire. After having found the necessary fitting parameters as outlined in the following subsection, we solve the model numerically for tire temperature and RR force using a stepwise explicit-Euler method.

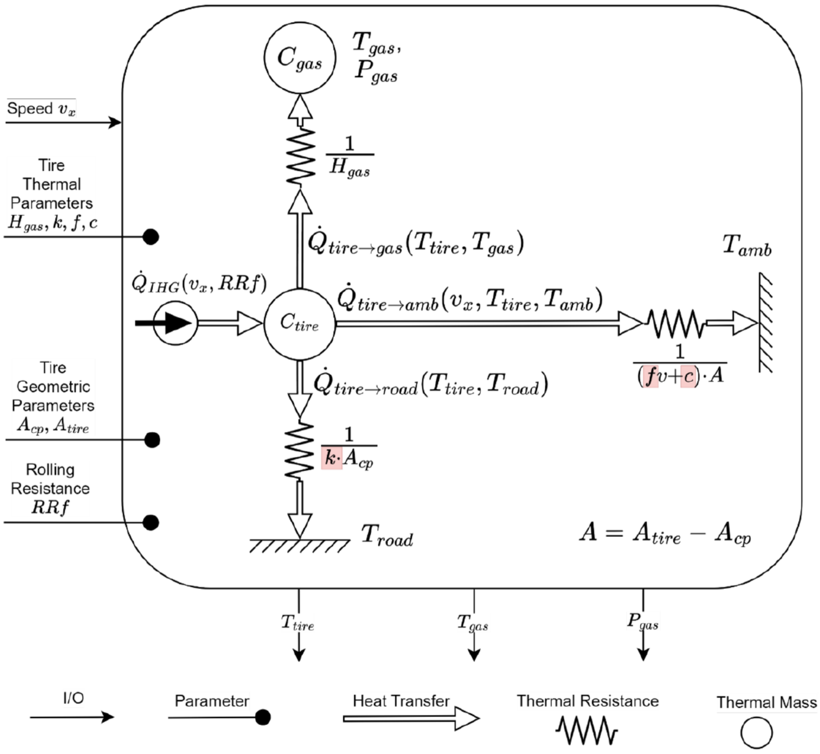

Figure 1: Thermal model for the tire temperature T_{tire} with the unknown heat transfer coefficients highlighted in red. From [4].

2.2 Tire data and parameter fitting

Timeseries data of RR force and tire temperature are measured in an ISO 28580 compliant [6] rig setup owned by Volvo Cars. Data is recorded starting from the initial ambient-soaked state at 23 degrees C, throughout the warmup phase until steady state. Using at least two different constant speeds this allows us to extract the four unknown parameters in Equation 1 from an exponential fit for force versus temperature.

To fit the thermal model, we discretize Equation 7 as follows,

$$\frac{dT}{dt} = \frac{\dot{Q}_{\text{IHG}} - \dot{Q}_c - \dot{Q}_v - \dot{Q}_{\text{gas}}}{C_p m} \;\rightarrow\; T_{i+1} = T_i + \Delta t \frac{\dot{Q}_{\text{IHG},i} - \dot{Q}_{c,i} - \dot{Q}_{v,i} - \dot{Q}_{\text{gas},i}}{C_p m} \tag{9}$$

where i denotes a discrete timestep. By substituting the measured force and temperature in Equation 9 we obtain a linear system of equations which we solve numerically for the two unknown heat transfer coefficients.

The fitted tire parameters are considered as proprietary Volvo Cars data and therefore cannot be published explicitly; some general observations for the specific tires we studied can however be noted (with vehicle speed in units of m/s):

- The A_0 coefficient is a factor ~70 to ~100 larger than A_1
- The B_0 coefficient is a factor ~2 000 to ~10 000 larger than B_1 – note however that the uncertainties in the fitted values for B_1 are quite large
- Regarding the heat transfer coefficients, the parameter z (governing the speed-independent heat transfer) is larger than the parameter f (governing the speed-dependent heat transfer) by a factor ranging from ~4.5 for the smallest tire (OD 670.6 mm) up to ~27 for the largest tire (OD 764.6 mm).

3 Results

The coupled RR- and temperature model described in Section 2 is applied to a simulation of a rear-wheel drive Volvo EX40 electric vehicle with the same tires used in the fitting procedure, running the SAE J1634 multicycle test (MCT) procedure [7] in a chassis dynamometer lab owned by Volvo Cars. We apply the ISO 28580 drum diameter correction to the calculated force in each timestep to account for the slightly smaller roller diameter in the dynamometer compared to the tire rig. In Figure 2 we compare the simulated and measured tire temperatures which are found to be within +/- 1 degree C for the entirety of the test.

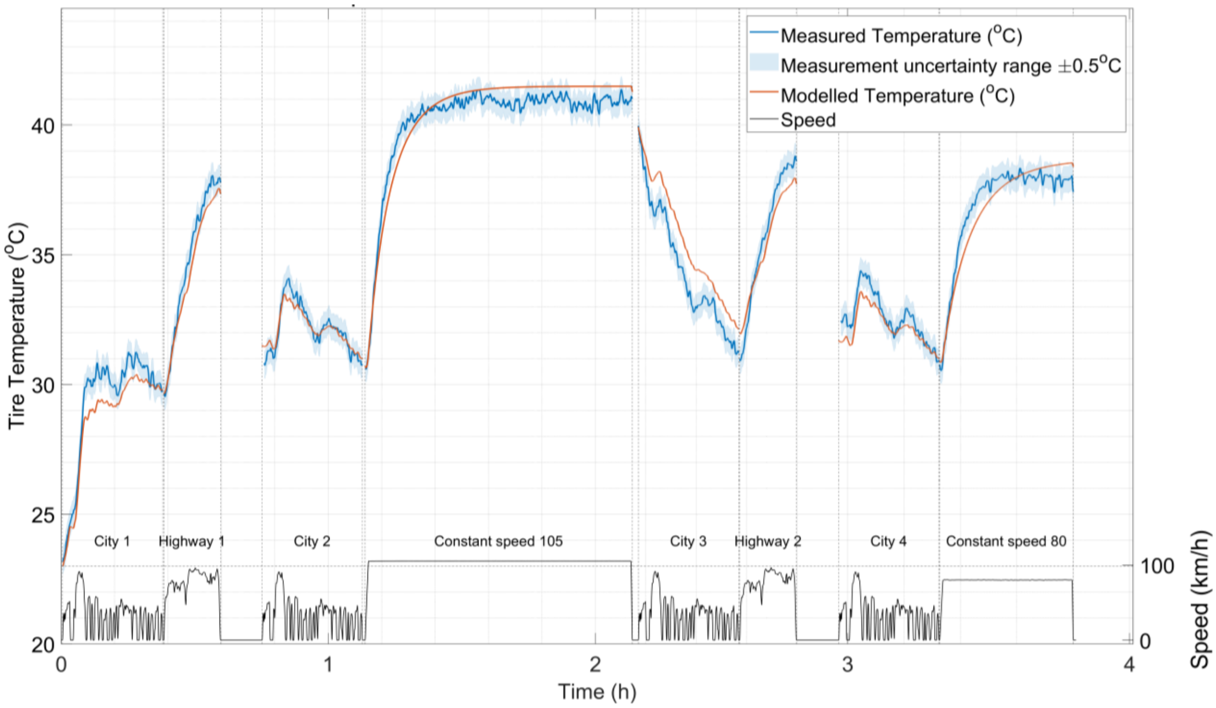

Figure 2: Tire temperature simulation results in red are compared to experimental data from a complete-vehicle test on a chassis dynamometer. The agreement is generally within 1 degree C. From [4].

To assess the RR part of the model we configure the dynamometer such that both rollers (front and rear) rotate in sync, although it is only the rear roller which is driven by the vehicle. In this way, the front roller needs to overcome the front axle RR which is dominated by the tire RR, and has a small contribution from wheel bearings and brake drag. In Figure 3, the corresponding rolling power loss for the two highway sections of the MCT is compared to our simulated rolling power loss P_{sim} for the front axle wheels, where we have used our simulated RR force and added estimations of the brake drag and wheel bearing contributions:

$$P_{\mathrm{sim}} = \left(F_{\mathrm{RR}} + F_{\mathrm{brake}} + F_{\mathrm{bearing}}\right)v \tag{10}$$

The brake and bearing forces are approximated to be constants and their values are estimated based on proprietary Volvo Cars data. The simulated power is found to be in good agreement with the observed data, as can be seen in Figure 3.

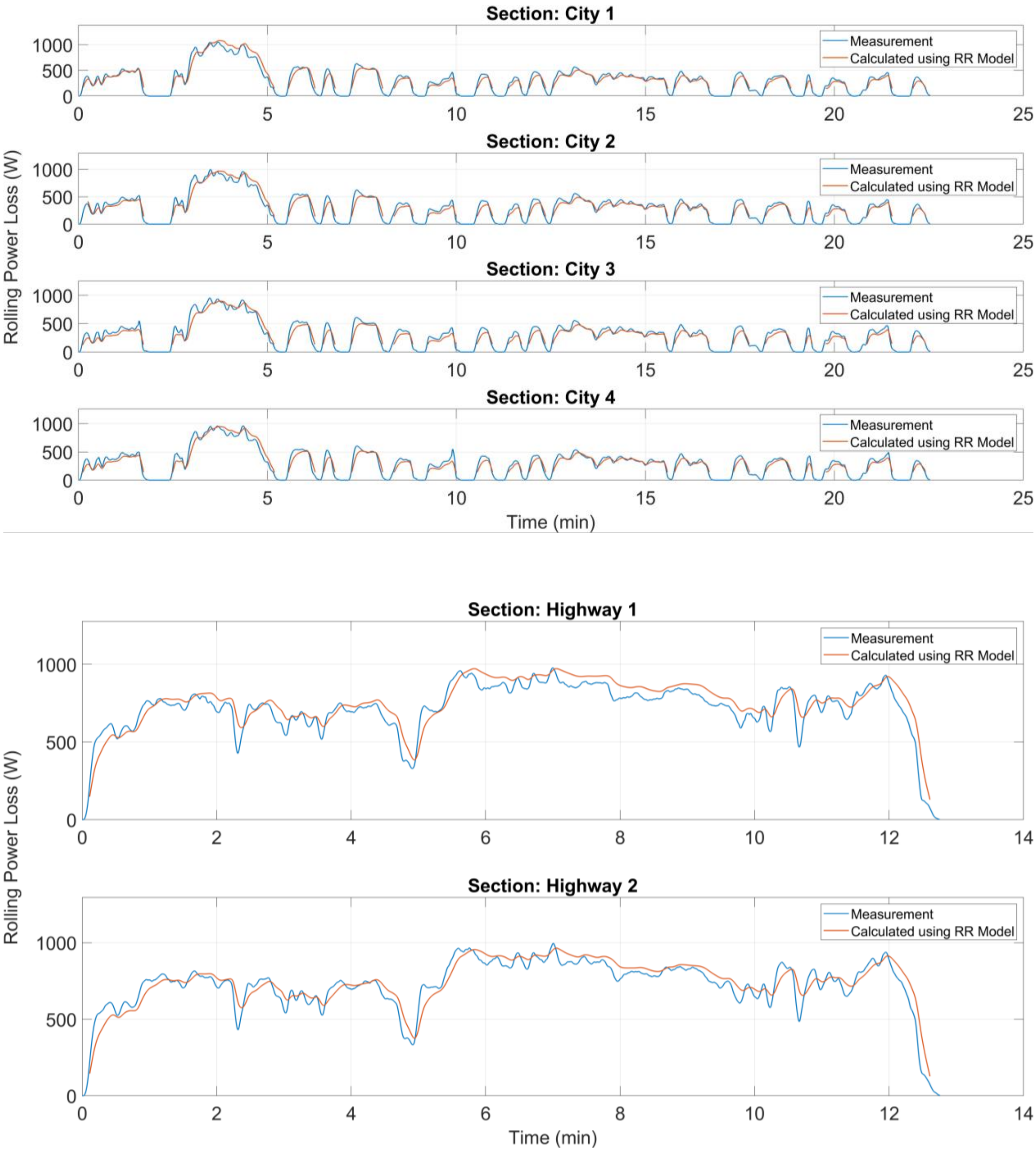

Figure 3: Rolling power loss simulation results in red are compared to experimental data from complete-vehicle tests on a chassis dynamometer. Top: four “City” sections from the MCT test. Bottom: two “Highway” sections from the MCT test. Taken from [4].

4 Conclusion and Outlook

We have demonstrated that by using our comparatively simple model, it is possible to reproduce observed tire temperature and rolling resistance under dynamic conditions to good accuracy. Our model requires 7 fitting parameters (which can be reduced to 6 under the approximation of a single ambient temperature for both air and road) which are taken from measurements.

Future work might investigate whether these parameters could be obtained virtually from more detailed tire simulations – e.g. constitutive models or FEM models. This would enable using our simple and less computationally demanding RR model in complete-vehicle simulations at earlier stages in the product development process.

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Presenter Biography

Johan Lindquist Holmberg is Technical Expert at the Energy Efficiency Centre at Volvo Cars, focusing on the energy consumption of electric vehicles. He obtained his Ph.D. in physics from Heidelberg University in 2015.