Pertti Virtala

RIDE-model

Definition of the RIDE-model

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Sammanfattning

Ruotsinkielinen tiivistelmä on kaikissa Liikenneviraston tutkimukset ja set -sarjan julkaisuissa. Muissa julkaisuissa se ei ole pakollinen.

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Summary

Englanninkielinen tiivistelmä on kaikissa Liikenneviraston tutkimuksia ja selvityksiä -sarjan julkaisuissa. Muissa julkaisuissa se ei ole pakollinen.

Esipuhe

Tässä dokumentissa on määritetty RIDE-tunnuslukujen laskentaperiaatteet. Dokumentti on tehty Liikenneviraston toimeksiannosta (Juho Meriläinen) ja sen on tuottanut Destia Oy (Pertti Virtala).

Helsingissä toukokuussa 2017

Liikennevirasto
Väylänpito/Kunnossapidon ohjaus ja kehittäminen

Table of Content

1 Introduction

The International Roughness Index (IRI) developed by the World Bank (1986) has been in use in the Finnish Transport Agency almost 30 years. It has been the main indicator describing the roughness (or unevenness) of a road pavement in the Asset Management Systems (RAMS) and in the Quality Assurance Standards (QS).

Numeral studies have showed that IRI is not the best index concerning riding comfort. Kim et al. [3] performed correlation analysis to determine the relation between measured accelerations and subjective ratings of four expert drivers using a psychophysical power law. Nair et al. [4] have described the experience, accomplishments, and challenges of using pavement roughness to set specifications for ride comfort on Virginia's highways. The practical experiences in the Finnish Transport Agency support those statements. The pavement management system in Transport Agency works with the definition of the amount of pavements in poor condition. Tree main indexes (rut depth, IRI, and crack index) are classified into five distinctive classes (from poor to excellent) where two worst classes belong to the category of poor condition. The main weight is typically set to rutting of main roads and to the cracking of secondary roads. Roughness has not been as important as those two. It is a sort of secondary index affecting to the type of action or treatment when the main indexes exceed threshold values. There have clearly been problems in using the roughness information of pavements in pavement management during the past years. Typical statements have been that it does not represent the riding comfort of pavements well enough and it cannot show where the bad sections locate or what the type of roughness is. The conclusion is then that a better index describing the roughness of pavements is needed. It should locate the bad sections more precisely and it should give information of what kind of roughness exists. The relation of roughness index to the riding comfort should be better too. [1].

As a result of several studies funded by the FTA the conclusion has been to develop a better index to describe the roughness of road pavement. The main question was, what were the root causes for the problem, where IRI seems not to represent the riding comfort of a traveller in a car or in a truck well enough. A root cause analysis was conducted and the following potential root causes were found [1,2]:

- IRI is calculated using a quarter car model, where only one longitudinal profile is fed to the model and the outcome is therefore missing some important dimensions. One longitudinal profile cannot have any information of how the cross fall varies in transversal direction.
- Quarter car model is based on an American car where the suspension system differs from what is in a typical European or Japanese car or a truck.
- IRI represents the vertical movement of a car chassis which is not the best source for searching factors causing discomfort. (ISO 2631 uses vertical accelerations) [2].
- The wavelength of longitudinal profile data used in calculating IRI is filtered to cover wavelengths between 0.5 and 50 m. Wavelengths shorter than 0.5 m can affect to the riding comfort as well.
- IRI is simulated using a constant speed 80 km/h. On high trafficked roads the speed of cars is usually much higher (100-120 km/h). On low trafficked roads the speed of cars is usually much lower (50-60 km/h). The speed used in simulations of IRI differs from the real driving speed.
- The suspension system of the driver's seat is not included in the quarter car simulation model.
- The reporting interval of 100 m is too long to locate poor road sections accurately. It also averages the outcome too much and important transients in roughness cannot be recognized.
- Some of the parameters of the suspension system in a car are changing over time, which can affect to the riding comfort.

The decision has been to move from a 2-DOF IRI-model to a full-car 7-DOF RIDE-model and introduce the indicators from RIDE-model to the Finnish RAMS. New indexes are based on vertical acceleration, roll acceleration, pitch acceleration and a combined vertical acceleration of vehicle sprung mass. This report describes the principles of the calculation of new indexes. A more detailed study of the new indicators is based on the results of the work done in [1].

2 RIDE-model

2.1 Standard terminology

2.1.1 Coordinate system

Axis system is a set of three mutually orthogonal X, Y, and Z axes. In a right-handed system, Z =X *Y [3].

Coordinate system is a numbering convention used to assign a unique ordered trio of numbers to each point in a reference frame. A typical rectangular coordinate system consists of an axis system plus an origin point [3].

Vehicle axis system (X_V, Y_V, Z_V) is a right-handed orthogonal axis system fixed in the vehicle reference frame, usually in the center of gravity (CG) of vehicle. The X_V axis is primarily horizontal in the vehicle plane of symmetry and points forward. The Z_V axis is vertical and the Y_V axis is lateral. The directions should coincide with the earth-fixed axis system when the vehicle is upright and aligned with the X_V axis parallel to the X_E axis [4].

Multibody vehicle dynamics models are typically generated using right-handed axis systems and coordinate systems. The axis orientation for ISO 8855 has X forward, Z up, and Y pointing to the left-hand side of the vehicle [4].

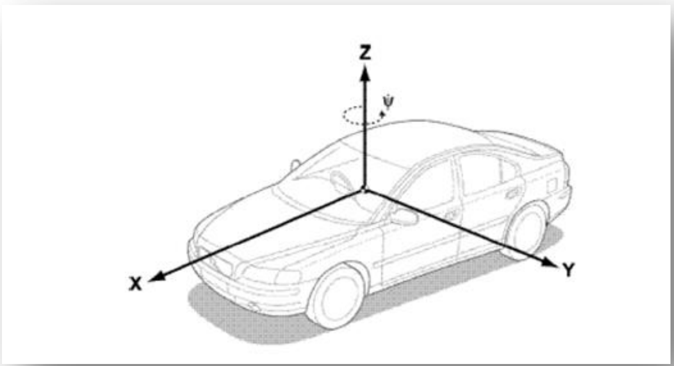

Figure 1. Vehicle body coordinate system as ISO standard [4].

Road axis system (X_R, Y_R, Z_R) — right-handed orthogonal axis system whose Z_R axis is normal to the road, at the center of tire contact, and whose X_R axis is perpendicular to the wheel spin axis (Y_W). For an uneven road, a different road axis system exists for each tire.

2.1.2 Vehicle

2.1.2.1 Size and weight

Track width (W) is the distance between the centers of tire contact on one side of the vehicle.

Wheelbase (L) is the distance between the front and rear axle. Distance of vehicle C.G. from front axle is marked with L_f and distance of vehicle C.G. from rear axle is marked with L_r .

Unsprung weight (m_u) is portion of weight supported by a tire that is considered to move with the wheel. This usually includes a portion of the weight of the suspension elements. (SAE) [3]. Each wheel has an individual unsprung weight which is marked with letters describing the front (f) or rear (r) and left (l) or right (r), as m_{ufl}, m_{ufr}, m_{url} and m_{urr}.

Sprung weight (m_s) is the weight of the vehicle chassis including driver, passengers, and loads.

Total weight of the vehicle is the sum of sprung weight and unsprung weights.

2.1.2.2 Points

C.G. (Center of gravity) is a point in the vehicle reference frame that coincides with the center of mass of the entire vehicle when the suspensions are in equilibrium and the vehicle is resting on a flat level surface.

Vertical position is Z - Z_E coordinate of the C.G.

X position is X - X_E coordinate of the C.G.

Y position is Y - Y_E coordinate of the C.G.

Pitch center is imaginary point in the $X_V Z_V$ plane through the lateral center of the vehicle reference frame at which a longitudinal force applied to the vehicle body is reacted without causing suspension jounce (front or rear). An alternate definition is that the pitch center is the intersection of the two lines shown in Figure 2. Note: this definition of pitch center does not take into account the “wind up” effects of drive train torque applied to the wheels from the vehicle body [3].

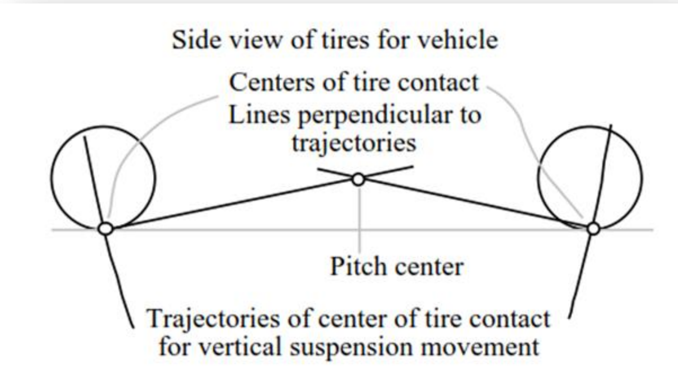

Figure 2. Pitch center [4].

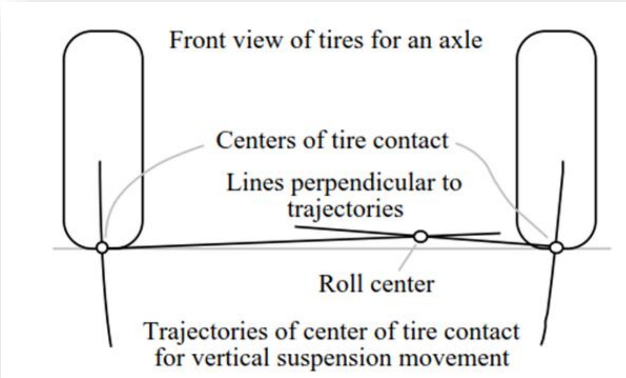

Figure 3. Roll center [4].

Roll center is an imaginary point in the Y_VZ_V plane containing the two wheel centers of an axle, at which a lateral force applied to the vehicle body is reacted without causing suspension roll angle (ISO, SAE). An alternate definition is that the roll center is the intersection of the two lines shown in Figure 3. (Note: the figure shows a non-equilibrium position of the vehicle.) [4].

2.1.2.3 Suspension

Spring is a part which combines the wheel to the chassis for each corner. *Spring stiffness (k_{fl}, k_{fr}, k_{rl}, k_{rr})* define the amount of load the spring is needed to be pressed to reach a certain compression.

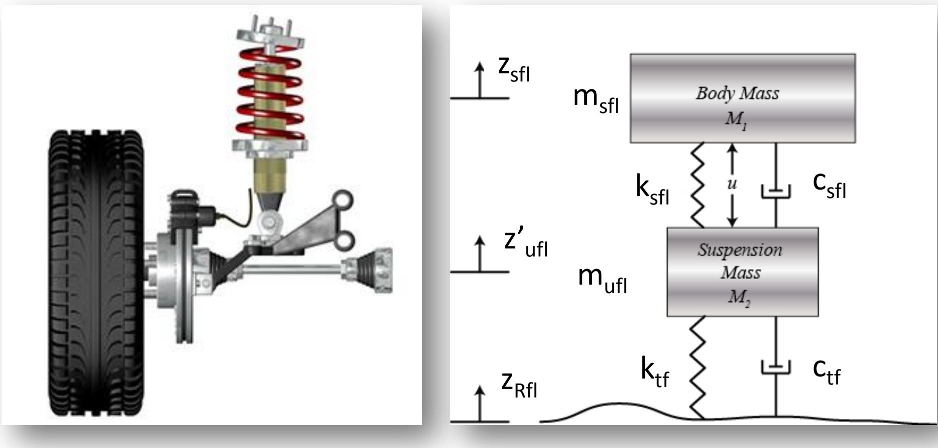

Figure 4. Suspension of a vehicles front left corner.

Damping coefficient is the ability of a shock absorber to resist movement. It is marked with a letter locating the corner in the vehicle (c_{fl}, c_{fr}, c_{rl}, c_{rr}).

A tire can have a spring parameter (k_t) and a suspension parameter (c_t).

2.1.3 Forces and moments

Damping force F_d is a compressive force applied to the vehicle body by a damper (F_{dfl}, F_{dfr}, F_{drl}, F_{drr}).

Spring force F_s is a compressive force applied to the vehicle body by a suspension spring (F_{sfl}, F_{sfr}, F_{srl}, F_{srr}).

Vertical tire force F_z is the Z_R component of ground resultant force. (SAE16, ISO15) (F_{zfl}, F_{zfr}, F_{zrl}, F_{zrr}).

Suspension roll moment M_{roll} is a total static roll moment applied to sprung roll angle. A positive moment causes positive vehicle roll.

Suspension pitch moment M_{pitch} is a total static pitch moment applied to sprung pitch angle. A positive moment causes positive vehicle pitch.

2.1.4 Movements and angles

Vertical displacement z_s is the Z component of displacement vector of the C.G.

Vertical velocity $\dot{z}_s$ is the Z component of velocity vector of the C.G.

Vertical acceleration $\ddot{z}_s$ is the Z component of acceleration vector of the C.G.

Pitch θ_s is an angle from X axis to X_V axis, about Y axis. (SAE, ISO) [4].

Pitch velocity $\dot{\theta}_s$ is the Y component of angular velocity vector of vehicle reference frame.

Pitch acceleration $\ddot{\theta}_s$ is the Y component of angular acceleration vector of vehicle reference frame.

Roll φ_s is the angle from X_EY_E plane to Y_V axis, about X axis (SAE, ISO) [4].

Roll velocity $\dot{\varphi}_s$ is the X component of angular velocity vector of vehicle reference frame.

Roll acceleration $\ddot{\varphi}_s$ is the X component of angular acceleration vector of vehicle reference frame.

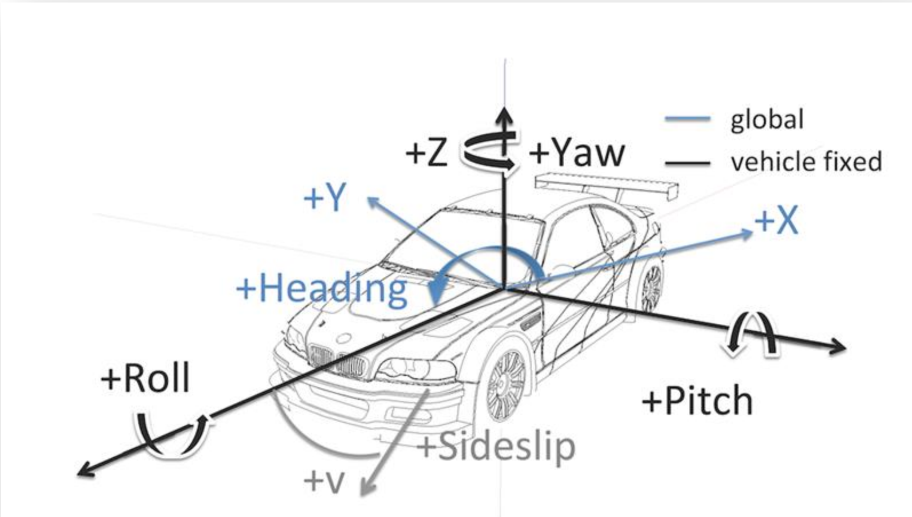

Figure 5. Roll, pitch and yaw of a vehicle.

2.2 Graphical model

The full-vehicle suspension system is represented as a linear seven degree-of-freedom (DOF) system. It consists of a single sprung mass (car body) connected to four unsprung masses (front-left, front-right, rear-left and rear-right wheels) at each corner. The sprung mass is free to bounce, pitch and roll while the unsprung masses are free only to bounce vertically with respect to the sprung mass. All other motions are neglected for this model. Hence this system has seven degrees of freedom and allows simulation of tyre load forces in all four tyres, body acceleration and vertical body displacement as well as roll and pitch motion of the car body. The suspensions between the sprung mass and unsprung masses are modelled as linear viscous dampers and linear spring elements, while the tyres are modelled as simple linear springs without damping. For simplicity, all pitch and roll angles are assumed to be small [5].

The model of a full-car suspension system is shown in Figure 6. The full-vehicle suspension model is represented as a linear seven degree of freedom system. The lateral dynamics of the vehicle are ignored. It consists of a single sprung mass m (car body) connected to four unsprung masses m1…m4 (front-left, front-right, rear-left and rear-right wheels) at each corner. The suspensions between the sprung mass and unsprung masses are modelled as linear viscous dampers and spring elements. The dampers between the body and the wheels represent sources of conventional damping such as friction between the mechanical elements. For the vehicle modelling full-car will be used as a good approximation of the entire car [5].

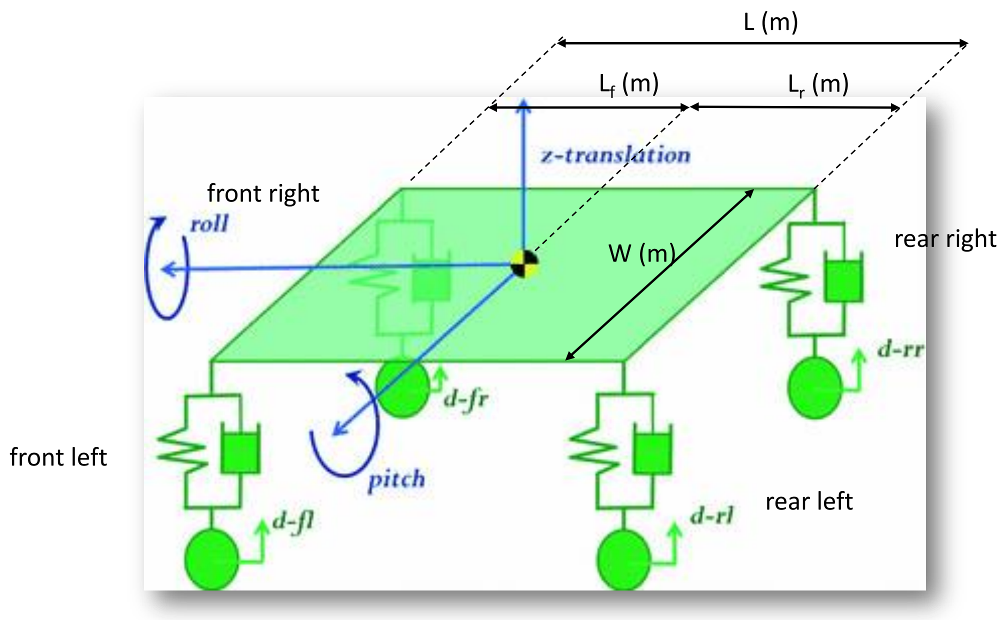

Figure 6. Graphical vehicle model.

There are some assumptions made in this model. The vehicle aerodynamic effect is neglected and the road is assumed to be level except for road disturbance. That means that the geometry of the road is neglected. The vehicle is also assumed to be rigid where the load transfer from one point to another is one hundred percent effective. Parameters of the vehicle are also assumed to be constant throughout the simulation process such as tire stiffness, spring stiffness, and damper coefficient.

2.3 Motion equations

At time t, the vertical profile of the road (road roughness) corresponding to the front right wheel is denoted by $z_{Rfr}(t)$, the road roughness corresponding to the front left wheel is denoted by $z_{Rfl}(t)$, the road roughness corresponding to the rear left wheel is denoted by $z_{Rrl}(t)$, finally the road roughness corresponding to the rear right wheel is denoted by $z_{Rrr}(t)$. In fact, since a straight road is considered here, the road roughnesses $z_{Rfl}(t)$ and $z_{Rfr}(t)$ are encountered by the front wheels (right and, respectively, left wheels) at time t, while the right and left rear wheels encounter the same road roughnesses after $\Delta t \cong L/v$ (so at t +Δt), where L is the wheelbase and v the constant car speed. Thus: $z_{Rfl}(t) = z_{Rrl}(t + \Delta t)$ and $z_{Rfr}(t) = z_{Rrr}(t + \Delta t)$.

The vertical dynamics equations of the 7 DOF model are obtained using the Newton-Euler formulation, as provided widely in the literature. There are 3 scalar dynamic equations of the sprung mass: one in terms of vertical forces (z direction), one in terms of moments with respect to the longitudinal x axis and the third one in terms of moments with respect to the transversal y axis.

The dynamic of the sprung mass is defined by:

$$m_s\ddot{z}_s = k_{sf}\left(z_{Rfl} - z_{sfl}\right) + k_{sf}\left(z_{Rfr} - z_{sfr}\right) + k_{sr}(z_{Rrl} - z_{srl}) + k_{sr}(z_{Rrr} - z_{srr}) + c_{sf}\left(\dot{z}_{ufl} - \dot{z}_{sfl}\right) + c_{sf}\left(\dot{z}_{ufr} - \dot{z}_{sfr}\right) + c_{sr}(\dot{z}_{url} - \dot{z}_{srl}) + c_{sr}(\dot{z}_{urr} - \dot{z}_{srr}) - F_{zfl} - F_{zfr} - F_{zrl} - F_{zrr} + m_s g$$

Where,

m_s = the mass of the vehicle chassis (kg) without wheels and axels
$\ddot{z}_s$ = the body vertical acceleration (m/s^2)
z_R = road profile for each wheel (m) (front left, front right, rear left and rear right)
z_s= vertical displacement (m) of the sprung mass for each corner
$\dot{z}_s$ = vertical displacement velocity of the sprung mass for each corner (m/s) (front left, front right, rear left and rear right)
k_s = spring coefficient (N/m) for the spring in each corner (front left, front right, rear left and rear right)
c_s = damping coefficient (Ns/m) for the damper in each corner (front left, front right, rear left and rear right)
$\dot{z}_u$ = vertical displacement velocity (m/s) for the unsprung mass in each wheel (front left, front right, rear left and rear right)

The roll effect of the vehicle is given by:

$$J_x\ddot{\varphi} = +\left\{\left[k_{sf}\left(z_{ufl} - z_{sfl}\right) + c_{sf}\left(\dot{z}_{ufl} - \dot{z}_{sfl}\right) + k_{sr}(z_{url} - z_{srl}) + c_{sr}(\dot{z}_{url} - \dot{z}_{srl})\right] - \left[k_{sf}\left(z_{ufr} - z_{sfr}\right) + c_{sf}\left(\dot{z}_{ufr} - \dot{z}_{sfr}\right) + k_{sr}(z_{urr} - z_{srr}) + c_{sr}(\dot{z}_{urr} - \dot{z}_{srr})\right] - \left(F_{zfl} + F_{zrl}\right) + \left(F_{zfr} + F_{zrr}\right)\right\} {}^{W}\!/_{2}$$

Where,

J_X = the moment of inertia about x-axis (kgm^2)
$\ddot{\varphi}$ = the roll acceleration (rad/s^2)
W/2 = the length of the vehicle from the center of gravity to the right end or to the left end of the vehicle (m).

The pitch effect of the vehicle is given by:

$$J_y\ddot{\theta} = -\left[k_{sf}\left(z_{ufl} - z_{sfl}\right) + c_{sf}\left(\dot{z}_{ufl} - \dot{z}_{sfl}\right) + k_{sf}\left(z_{ufr} - z_{sfr}\right) + c_{sf}\left(\dot{z}_{ufr} - \dot{z}_{sfr}\right)\right]L_f + [k_{sr}(z_{url} - z_{srl}) + c_{sr}(\dot{z}_{url} - \dot{z}_{srl}) + k_{sr}(z_{urr} - z_{srr}) + c_{sr}(\dot{z}_{urr} - \dot{z}_{srr})]L_r + \left(F_{zfl} + F_{zfr}\right)L_f - (F_{zrl} + F_{zrr})L_r$$

Where,

J_Y = the moment of inertia about y-axis (kgm^2)
$\ddot{\theta}$ = the pitch acceleration (rad/s^2)
L_f = the length of vehicle from the center of gravity to the front axel (m)
L_r = the length of vehicle from the center of gravity to the rear axel (m)

Acceleration at unsprung mass for each wheel is given by:

$$m_{ufl}\ddot{z}_{ufl} = k_{tf}\left(z_{Rfl} - z_{ufl}\right) + c_{tf}\left(\dot{z}_{Rfl} - \dot{z}_{ufl}\right) + k_{sf}\left(z_{sfl} - z_{ufl}\right) + c_{sf}\left(\dot{z}_{sfl} - \dot{z}_{ufl}\right) + F_{zfl} + m_{ufl}g$$

$$m_{ufr}\ddot{z}_{ufr} = k_{tf}\left(z_{Rfr} - z_{ufr}\right) + c_{tf}\left(\dot{z}_{Rfr} - \dot{z}_{ufr}\right) + k_{sf}\left(z_{sfr} - z_{ufr}\right) + c_{sf}\left(\dot{z}_{sfr} - \dot{z}_{ufr}\right) + F_{zfr} + m_{ufr}g$$

$$m_{url}\ddot{z}_{url} = k_{tr}(z_{Rrl} - z_{url}) + c_{tr}(\dot{z}_{Rrl} - \dot{z}_{url}) + k_{sr}(z_{srl} - z_{url}) + c_{sr}\left(\dot{z}_{sfr} - \dot{z}_{url}\right) + F_{zrl} + m_{url}g$$

$$m_{urr}\ddot{z}_{urr} = k_{tr}(z_{Rrr} - z_{urr}) + c_{tr}(\dot{z}_{Rrr} - \dot{z}_{urr}) + k_{sr}(z_{srr} - z_{urr}) + c_{sr}(\dot{z}_{srr} - \dot{z}_{urr}) + F_{zrr} + m_{urr}g$$

2.4 Road

2.4.1 Raw data

The raw data is a data consisting of the longitudinal profile data of a 3.2 m wide lane with 100 mm intervals. Currently the raw data is measured with the GE manufactured (Greenwood Engineering Ltd) point laser-based high speed monitoring vehicle. The measurement beam is locating in front of the vehicle and it has at a minimum of 17 point-lasers which measure the vertical distance of the road surface from the beam. The beam construction has usually more sensors in both wheel paths than in between. If the wheel path is measured with five sensors then the longitudinal profile is calculated as a mean of information collected with those five sensors. Longitudinal profile for left wheel path is obtained from sensors measuring the left wheel path and for the right wheel path respectively from sensors measuring that. The reference point of raw data is the data of the left wheel path at the location of 0 meters. All other vertical data is measured proportional to that.

The raw data is filtered as is usually filtered when calculating the IRI.

Road is given to the dynamic simulation as a profile for the left wheel path and for the right wheel path in 10 cm intervals. It is given as:

$$z_{Rfl}(t) = mean\left(p_3(t), p_4(t), p_5(t), p_6(t), p_7(t)\right)$$

$$z_{Rfr}(t) = mean\left(p_{11}(t), p_{12}(t), p_{13}(t), p_{14}(t), p_{15}(t)\right)$$

$$z_{Rrl}(t) = mean\left(p_3(t-\Delta t), p_4(t-\Delta t), p_5(t-\Delta t), p_6(t-\Delta t), p_7(t-\Delta t)\right)$$

$$z_{Rrr}(t) = mean\left(p_{11}(t-\Delta t), p_{12}(t-\Delta t), p_{13}(t-\Delta t), p_{14}(t-\Delta t), p_{15}(t-\Delta t)\right)$$

Where,

$p_i(t)$ = individual parallel profiles in one wheel path covering a 50 cm width (m) where i is the sensor number in a point laser profilograph with 17 sensors in transversal direction starting from the left side
t = time (s) starting from 0 and ending to the t_{max} representing the time travelled to the end of a road section using speed v (m/s)
Δt = time interval (s) defined by the axel base L and speed v (m/s)
$z_R(t)$ = road profile (m) for each wheel (front left, front right, rear left and rear right) at the moment of t (s)

This principle is designed to work with the current measurement technologies. However it is possible to produce the longitudinal profiles for each wheel using other technologies and calculation methods.

Generally the rear wheels get the same profile than front wheels but the profile information flow is delayed with time interval Δt.

2.4.2 Additional filtering

Many low-volume roads are hilly or curvy meaning that the gradient in vertical direction or the curvature in lateral direction is varying quite a lot. Some parts of those roads cannot be driven with a car using constant speed representing the speed limit. That means that the simulation cannot either be driven with that speed. To avoid expanding simulations covering unlimited amount of different speed limits the longitudinal profile is necessary to be re-filtered. Re-filtering should be made according to the following principles:

```
%GEOMETRIA FILTTEROINTI
% Timo Eskola 7.3.2016 use filtering only if bUseProfileFilter>0 e.g. 1
% from ui
if bUseProfileFilter>0;
    p_v(:,2)=filtfilt(b,a,p_v(:,2));
    p_o(:,2)=filtfilt(b,a,p_o(:,2));
    p_tv(:,2)=filtfilt(b,a,p_tv(:,2));
    p_to(:,2)=filtfilt(b,a,p_to(:,2));
end
%END OF GEOMETRIA FILTTEROINTI
```

where

p_v() represents the left road profile and the p_o() represents the right road profile
a=xxxxxx
b=xxxxx

2.5 Initial position of the vehicle

Motion equations are describing the dynamic behavior of the vehicle during the simulation. The initial position of the vehicle must be calculated as initial values in z-direction.

```
% ------------------------- Initial position ------------------------
            massa=(m_k(i)+m_to(i)+m_tv(i)+m_eo(i)+m_ev(i));

% Wheel loads according to the C.o.G
            F_rr_stat(i)=m_k(i)*g*L_f(i)/L(i)/2;
            F_rl_stat(i)=m_k(i)*g*L_f(i)/L(i)/2;
            F_fr_stat(i)=m_k(i)*g*L_r(i)/L(i)/2;
            F_fl_stat(i)=m_k(i)*g*L_r(i)/L(i)/2;

% Initial displacement due to gravity
            as_afl(i)=-(m_fl(i)*g+F_fl_stat(i))/k_rfl(i);%+p_l(1,2);
            as_afr(i)=-(m_fr(i)*g+F_fr_stat(i))/k_rfr(i);%+p_r(1,2);
            as_arl(i)=-(m_rl(i)*g+F_rl_stat(i))/k_rrl(i);%+p_l(1,2);
            as_arr(i)=-(m_rr(i)*g+F_rr_stat(i))/k_rrr(i);%+p_r(1,2);

% initial displacement of front right due to gravity
            as_fr(i)=-F_fr_stat(i)/k_fr(i)+as_afr(i);

% initial displacement of front left due to gravity
            as_fl(i)=-F_fl_stat(i)/k_fl(i)+as_afl(i);

% initial displacement of rear right due to gravity
            as_rr(i)=-F_rr_stat(i)/k_rr(i)+as_arr(i);

% initial displacement of rear left due to gravity
            as_rl(i)=-F_rl_stat(i)/k_rl(i)+as_arl(i);

% Initial state of the rear center
            z_kr(i)=sin(asin((as_rr(i)-as_rl(i))/L_r(i)))*L_r(i)/2+as_rr(i);

% Initial state of the front center
            z_kf(i)=sin(asin((as_fr(i)-as_fl(i))/L_f(i)))*L_f(i)/2+as_fl(i);

% Initial Pitch angle
            as_pitch(i)=asin((-z_kr(i)+z_kf(i))/L(i));

% Initial displacement of C.o.G
            as_k(i)=z_kr(i)+sin(as_pitch(i))*L_r(i);

            z_l(i)=sin(asin((as_fl(i)-as_rl(i))/L(i)))*L_r(i)+as_rl(i);
            z_r(i)=sin(asin((as_fr(i)-as_rr(i))/L(i)))*L_r(i)+as_rr(i);

            s(i)=l_t(i)*sin(atan((L_f(i)-L_r(i))/(2*L(i))));
            as_roll(i)=asin((z_r(i)-z_l(i))/(L_r(i)));%+2*s(i)));
            %Alkuheilahduskulman pitäisi olla nolla
```

3 Lähteet

Tekstissä olevan lähdeviittauksen tarkoituksena on viitata julkaisun lopussa olevaan kirjallisuusluetteloon, jossa esitetään tarkat bibliografiset tiedot siteeratuista julkaisuista. Lähdeviitteen on ohjattava lukija vaivattomasti lähdeluettelon oikeaan kohtaan.

Suositeltava ja informatiivinen viittaustapa on nimi-vuosi-järjestelmä eli Harvardin järjestelmä.

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6.

4 Liitteet

Liitteet ovat varsinaista kirjoitusta täydentäviä itsenäisiä kokonaisuuksia, jotka sijoitetaan julkaisun tekstiosan jälkeen. Tavallisimpia liitteitä ovat taulukot, kuvat ja kartat, kyselylomakkeet yms.

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