Selection of the required deceleration for high-mobility wheeled vehicles with wear resistant brake systems

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Abstract. Analysis of the driving cycles of the high-mobility wheeled vehicles shows that the driver tries to provide a high speed when performing maneuvers. This results in high loads both on the powerplant and the brake system. During quick changes of the driving modes, the service brake system has to dissipate a lot of energy, which may cause overheating of its working elements and changes (deterioration) in their frictional properties or their increased wear, or even a failure of the system. The article reveals the method for selection of the required deceleration of high-mobility wheeled vehicles with wear resistant brake systems providing higher speeds and considerably lower loads on the service brake system and increased life of the frictional elements. The method is based on the generation of the driving cycle, which is close to the real-life operation conditions of the high-mobility wheeled vehicles with the use of stochastic data on the roads. Thus, driving conditions are described as realizations of random functions of such external factors as the trajectory curvature, the friction coefficient between the tire and the road, and the tire rolling resistance coefficient. The driving cycle is generated with allowance for the vehicle stability, the powerplant performance, the physiological capability of the driver to withstand accelerations, and the maximum available deceleration. By varying the vehicle deceleration we can find the dependence of the vehicle average speed on its deceleration. The data obtained during the analysis helps to select the deceleration which can be effectively provided by the wear resistant brake system without the use of the service brake system. A method for selection of the wear resistant brake system characteristics with the use of stochastic data on the driving conditions. The method allows selection of the torque – speed curves of the units of a wear resistant brake system providing required mobility of the vehicle.

1. Introduction
High-mobility vehicles drive on the roads and on the terrain with different surface conditions [1, 2]. As a rule, when a driver performs a transportation task, he tries to provide the maximum speed determined either by the braking/tractive performance of the vehicle or by the condition of the vehicle stability during maneuvering. In this case, the driving cycle in the given road conditions is determined by the said limitations on the vehicle speed rather than by the driver [3, 4].

This assumption is the base of the proposed method for selection of the required deceleration of the high-mobility wheeled vehicles. At the first stage, a driving cycle is generated for the conditions
described by long realizations of random functions of the external disturbances. Then, the vehicle deceleration is varied and the dependence of its average speed on its deceleration is defined.

The data obtained during the analysis is used for the selection of the required deceleration, which can be effectively provided by the wear resistant brake system only in order to provide high mobility of the vehicle, as well as to reduce the heat loads and increase life of the frictional elements of the service brake system [5, 6, 7].

2. Generation of the vector of external disturbances

The routes for the high-mobility vehicles are generated from the known stochastic parameters of the roads as realizations of random functions of such road conditions as road curvature, maximum friction coefficient between the tire and the road, rolling resistance coefficient, road surface longitudinal slope angle [8 – 14].

The efficient method for generation of the above-mentioned stochastic parameters of the road is the non-canonical decomposition [9 – 12].

Thus, generating random combinations of these parameters for different road types and different terrain we can form a vector of the external disturbances, which is a combination of the long realizations of the random functions of the distance travelled by the vehicle. In order that this combination of the random parameters (road curvature, maximum friction coefficient between the tire and the road, rolling resistance coefficient, road surface longitudinal slope angle) could be used as a driving route, it should meet the following conditions:

• long realizations of the random functions must be generated as a sequence of the road segments with constant road conditions;
• on each of the road segment all the stochastic parameters must be defined: road curvature, maximum friction coefficient between the tire and the road, rolling resistance coefficient, road surface longitudinal slope angle.

The route generated with this method will have discontinuity at the boundaries of the road segments for each of the stochastic parameter function. Bearing this in mind, we have made the assumption that the given speed (maximum available speed for each segment) is obtained in the middle of the segment. At the same time, between the segments the vehicle speed changes according to a linear law, i.e. the vehicle moves with a constant acceleration. To form the equations of the vehicle motion we will assume that during the transition from one road segment to another the values of the parameters describing the vehicle interaction with the road are equal to their mean values derived from the values for the two segments (figure 1).
3. Generation of the driving cycle for a high-mobility wheeled vehicle with the use of a quasi-stationary model of the vehicle motion

During generation of the driving cycle of the vehicle, first, the upper bound of the vehicle maximum speed on each road segment and the length of the route must be estimated.

This estimate can be efficiently performed with a massless model of the vehicle motion [8]. The following estimates use several assumptions:

- the gearbox output torque – speed curve subject to the energy losses in the drivetrain is a constant power curve (corresponding to the maximum power of the power plant);
- increase in motion resistance at turns is negligible;
- the tire contacts the ground at one point (the turning resistance torque is neglected).

The upper bound of the vehicle maximum speed for each \(i\)-th segment of the road \(v_{lim_i}\) can be estimated as follows:

\[
\begin{align*}
    v_{lim_i} &= \min(v_{skd_i}, v_{roll_i}, v_{Nmax_i}, v_{max}), \\
    v_{Nmax_i} &= N_{max}/\left(\frac{mg}{f_i \cos(\alpha_i)} + \sin(\alpha_i)\right) + k_w F_{\text{front}} v_{Nmax_i}^2, \\
    v_{skd_i} &= \sqrt{\mu_{max} g/k_{\alpha_i}}, \\
    v_{roll_i} &= \sqrt{\frac{gB}{(2H_z k_{\alpha_i})}}, \quad k_w = c_x \rho_w/2,
\end{align*}
\]  

(1)
where $f_i$ is the rolling resistance coefficient on segment $i$ of the route; $v_{skd_i}$ is the maximum speed limited by the vehicle skidding; $v_{roll_i}$ is the maximum speed limited by the vehicle roll (the vehicle is assumed to be equipped with an ideal anti-roll bar); $v_{nmax_i}$ is the maximum speed limited by the vehicle engine power; $v_{max}$ is the maximum designed speed (the maximum speed according to specification, this speed is usually limited by the engine control system); $m$ is the mass of the high-mobility vehicle; $g$ is the acceleration of gravity; $B$ is the track width of the vehicle; $H_z$ is the height of the vehicle center of mass; $N_{max}$ is the maximum power of the vehicle engine; $k_w$ is the aerodynamic force coefficient; $F_{front}$ is the frontal area of the vehicle; $c_x$ is the aerodynamic drag coefficient; $\rho_w$ is the air density.

The dependence of the maximum speed bound $v_{lim}$ on the travelled distance for a stochastically generated route is shown in figure 2 (the vehicle moves over a surfaced road on a flat terrain).

![Figure 2](image_url)

**Figure 2.** Upper bound of the vehicle maximum speed as a function of the travelled distance.

The length of the route with the given road surface should be estimated under the following condition: the average speed of the vehicle will level off at the end of the route. After that, the simulation can be stopped since the data collected after the speed stabilization will have no effect on the result for the given road. Since the inertia of the vehicle smooths oscillations of its speed, the length of the route can be estimated with the use of a massless model (using speed upper bound as the average speed [8]).

The upper bound of the vehicle average speed $v_{lim}^{mean}$ depends on the travelled distance:

$$ v_{lim}^{seg} = \frac{v_{lim_i} + v_{lim_{i-1}}}{2}, \quad \Delta s_i = s_i - s_{i-1}, $$

$$ v_{lim}^{max} = \sum_{i=1}^{n_s} v_{lim}^{seg} \Delta s_i / s_i, \quad i = 1, 2 \ldots n_s $$

(2)

where $n_s$ is the number of the road segments constituting the route ($s_i$ is the longitudinal coordinate on the $i$-th segment, $s_{n_s}$ is the length of the route); $v_{lim}^{seg}$ is the upper bound of the maximum speed of the vehicle during transition from segment $i-1$ to segment $i$ of the route; $\Delta s_i$ is the length increment between coordinates $i-1$ and $i$ of the route (the length of the route segment).

Figure 3 shows the upper bound of the vehicle average speed as a function of the travelled distance. The example presented in figure 3 demonstrates that the required length of the route is 5 km (the discrepancy between the speed at 5 km and the final value at 10 km is less than 2%).
On the next stage, limitations on the longitudinal acceleration/deceleration of the vehicle must be imposed on the driving cycle generated with the use of the above mentioned massless model (see figure 2). These limitations are determined by the tractive performance of the engine (including inertia forces) and by a certain maximum attainable deceleration of the vehicle. At that, by varying the latter we can estimate the dependence of the vehicle speed on the said deceleration.

In the driving mode, if the vehicle acceleration \( a_{lim} \) during transition from elementary segment \( i-1 \) to segment \( i \) of the route is higher than the limit value determined by the tractive characteristics of the road \( a_{\mu i} \) or by the engine performance \( a_{eng i} \), the driving cycle needs a correction:

\[
v_{tr i} = \begin{cases} v_{lim i}, & \text{if } 0 \leq a_{lim i} \leq \min \left( a_{\mu i}, a_{eng i} \right) \\ v_{tr i-1} + \frac{\min \left( a_{\mu i}, a_{eng i} \right) \Delta s_i}{v_{seg i}}, & \text{if } a_{lim i} > \min \left( a_{\mu i}, a_{eng i} \right) \end{cases}
\]  

(3)

where \( v_{tr i} \) is the upper bound of the maximum speed of the vehicle on segment \( i \) of the route subject to longitudinal acceleration limit; \( v_{tr i}^{seg} \) is the upper bound of the maximum speed of the vehicle during transition from segment \( i-1 \) to segment \( i \) of the route subject to longitudinal acceleration limit; \( a_{lim i}, a_{\mu i} \) and \( v_{tr i}^{seg} \) are defined as:

\[
a_{lim i} = \left( v_{tr i} - v_{tr i-1} \right) v_{tr i}^{seg} / \Delta s_i,
\]

(4)

\[
a_{\mu i} = s_{max i} g, \quad a_{max i} = \left( s_{max i} + s_{max i-1} \right) / 2,
\]

(5)

\[
v_{tr i}^{seg} = \left( v_{tr i} + v_{tr i-1} \right) / 2
\]

(6)

where \( s_{max i}^{seg} \) is the maximum friction coefficient between the tire and the road during transition from segment \( i-1 \) to segment \( i \) of the route.

The following assumptions have been made:
- the gearbox output torque – speed curve subject to the energy losses in the drivetrain is a constant power curve (corresponding to the maximum power of the power plant);
- moment of inertia of the rotating elements of the engine and the drivetrain is transferred to the wheels;
- additional resistance at turns is negligible;
- there is no slip at the tire contact with the road;
- the tire contacts the road at one point;
- kinematic radii of all the wheels are the same.

Then, the following equation can be used: [15, 16]:

\[
\begin{align*}
\alpha_{\text{eng}} - \frac{N_{\text{seg}}}{v_{\text{tr}_i}^2} - mg \left( f_i \cos(\alpha_{\text{seg}}) + \sin(\alpha_{\text{seg}}) \right) - k_W F_{\text{front}} v_{\text{tr}_i}^2 - J_2 k_i^2 \Delta k_{d_i} \frac{\Delta s_i}{\Delta s_i} v_{\text{tr}_i}^2 \\
\alpha_{\text{seg}} = \frac{\alpha_i + \alpha_{i-1}}{2},
\end{align*}
\]

(7)

where \( f_i \) is the rolling resistance coefficient during transition from segment i-1 to segment i of the route; \( \alpha_{\text{seg}} \) is the slope angle of the road during transition from segment i-1 to segment i of the route; \( k_{d_i} \) is the curvature of the trajectory during transition from segment i-1 to segment i of the route; \( \Delta k_{d_i} \) is the change of the curvature at transition from coordinate \( i-1 \) to coordinate \( i \) along the route; \( J_2 \) is the vehicle moment of inertia about its vertical axis; \( \delta \) is the mass factor taking into account the inertia of the rotating components of the driveline; \( n \) is the number of the wheels of the vehicle; \( \sum_{i=1}^{n} J_{k_i} \) is the sum of the moments of inertia of the engine, the drivetrain, and the wheels, the moments being transferred to the axes of the wheels; \( r_{k_0} \) is the tire rolling radius at zero slip (this radius can be assumed to be equal to the radius of the tire at its nominal load).

The upper bound of the maximum speed as a function of the distance is shown in figure 4.

![Figure 4. Upper bound of the maximum speed of the vehicle subject to longitudinal acceleration limitation.](image_url)
In the braking mode, if the vehicle acceleration \( a \) during transition from elementary segment \( i \) to segment \( i \) of the route is lower than the limit value \( a_\mu \) determined by the tractive characteristics of the road or the given deceleration \( a^* \), the driving cycle needs a correction. It is important to note that in this case the segments must be taken in the order from the last one to the first one so that to arrive at the given speed of the vehicle at the end of each segment.

\[
 v_{brk_{n-s-i}} = \begin{cases} 
 v_{lim_{n-s-i}} & \text{if } 0 \geq a_{lim_{n-s-i}} \geq \max \left( -a_{\mu_{n-s-i}}, -a^* \right) \\
 \max \left( -a_{\mu_{n-s-i}}, -a^* \right) \Delta s_{n-s-i} / v_{brk_{n-s-i}} & \text{if } a_{lim_{n-s-i}} \leq \max \left( -a_{\mu_{n-s-i}}, -a^* \right) 
\end{cases}
\]  

(9)

where \( v_{brk_{n-s-i}} \) is the upper bound of the maximum speed of the vehicle on segment \( i \) of the route subject to longitudinal acceleration limit; \( v_{seg_{brk_{n-s-i}}} \) is the upper bound of the maximum speed of the vehicle during transition from segment \( n_s - i + 1 \) to segment \( n_s - i \) of the route subject to longitudinal acceleration limit; \( v_{seg_{brk_{n-s-i}}} \) is calculated by the following equation:

\[
 v_{seg_{brk_{n-s-i}}} = \left( v_{brk_{n-s-i}} + v_{brk_{n-s-i+1}} \right) / 2
\]

(10)

The speed \( v_{brk} \) as the function of the travelled distance is shown in figure 5.

![Figure 5. Upper bound of the maximum speed of the vehicle subject to longitudinal deceleration limitation.](image)

The final driving cycle of the vehicle including limitations on the longitudinal acceleration/deceleration is generated as a combination of the curves \( v_{trr} \) and \( v_{brk} \) (see figure 6).

\[
 v = \min(v_{trr}, v_{brk}).
\]

(11)

Thus, the described massless model with additional limitations on the longitudinal acceleration/deceleration provides generation of the driving cycle for a high-mobility wheeled vehicle on a given route. Taking into account the assumption that the driving cycle for a high-mobility wheeled vehicle on a given road is rather determined by the braking/tractive performance of the vehicle or by the condition of the vehicle stability during maneuvering than by the driver, we can argue that the generated driving cycle is close to the real-life operation cycle of the vehicle.
4. Estimation of the deceleration to be provided by a wear resistant brake system of a high-mobility wheeled vehicle

During selection of the deceleration to be provided by a wear resistant brake system we assume that the vehicle speed decreases only due to the operation of the retarders and the deceleration is not higher than $\alpha^*$. Then, with the use of the generated driving cycle for a high-mobility wheeled vehicle subject to the longitudinal acceleration/deceleration limitation (see figure 6) the average speed of the vehicle $v_{\text{mean}}$ in the given road conditions (for the given acceleration limit $\alpha^*$) can be calculated as:

$$v_{\text{mean}} = \sum_{i=1}^{n_s} v_i^{seg} \Delta s_i / s_{n_s}, \quad v_i^{seg} = \left( v_i + v_{i-1} \right) / 2$$

By generating driving cycles for different $\alpha^*$, we can form the vehicle average speed – deceleration limit curve. By way of example, figure 7 shows the vehicle average speed – deceleration limit curves for a high-mobility wheeled vehicle with gross weight 34 t with different power-to-weight ratios (15 kW/t and 22 kW/t) on a bituminous-concrete surface.

Figure 6. Driving cycle of the high-mobility wheeled vehicle limited by deceleration $\alpha^* = 1 \text{ m/s}^2$. 

a)
Figure 7. Average speed – deceleration limit curves for a a high-mobility wheeled vehicle: a) power-to-weight ratio 15 kW/t; b) power-to-weight ratio 22 kW/t.

Figure 7 shows that the vehicle deceleration higher than 1 by the units of the secondary brake system (retarders) is not efficient, since it doesn’t increase the average speed neither for the vehicles already in operation (with power-to-weight ratio 15 kW/t) nor for the advanced vehicles (with power-to-weight ratio 22 kW/t).

The upper bound of the vehicle average speed did not change with the change of the power-to-weight ratio because in this case the vehicle maximum speed is limited not by the maximum power of the engine, but by the vehicle specification (the control system limits the maximum speed by the value 105 km/h).

The driving cycle of the vehicle with the power-to-weight ratio 15 kW/t for the selected deceleration limit 1 is shown in figure 6.

This deceleration correlates with the data from [17] (deceleration at service braking does not exceed 0.8 – 1.7g) and corresponds to the boundary of the comfort zone of the longitudinal accelerations [18] (the limit longitudinal acceleration for a comfortable ride is 0,1g, table 1).

| Limit values for the root mean square of the accelerations acting on the people onboard a car [14]. |
|---------------------------------------------------------------|
| Limit values for the root mean square of the accelerations for | Vertical accelerations | Longitudinal accelerations | Lateral accelerations |
| comfort boundary | 0,1g | 0,06g | 0,05g |
| comfortable ride | 0,25g | 0,1g | 0,07g |
| short term accelerations | 0,4g | 0,2g | 0,1g |

The driving cycle shown in figure 6 was generated without limitation on the lateral acceleration acting on the driver (lateral accelerations when the vehicle moves with the roll limit speed are $\mu_{\text{max}}g$)

People are not able to withstand driving with these accelerations for a long period of time. Therefore, the driving cycle of a high-mobility vehicle should account for the lateral acceleration limit value $a_y = 0,1g$ [18], which corresponds to the limit value for the short term accelerations (see table 1).

To this end, the upper bound of the vehicle maximum speed for each $i$-th segment of the road must be calculated as:

$$v_{lim_i} = \min\left(v_{a_y}, v_{skd_i}, v_{roll_i}, v_{Nmax_i}, v_{max}\right), \quad v_{a_y} = \sqrt{a_y/k_{d_i}}, \quad (13)$$
where $v_{a_y}$ is the vehicle maximum speed subject to lateral acceleration limitation.

Then, the vehicle speed $v_{\text{mean}}$ as the function of the deceleration limited by the value of the lateral acceleration can be presented by the curve shown in figure 8.

![Figure 8](image)

**Figure 8.** Average speed – deceleration limit curves for a high-mobility wheeled vehicle: a) power-to-weight ratio 15 kW/t; b) power-to-weight ratio 22 kW/t.

The driving cycle of the vehicle with the power-to-weight ratio 15 kW/t for the selected deceleration limit $1 \text{ m/s}^2$ limited by the value of the lateral acceleration $a_y = 0.1g$ is shown in figure 9.

The curves of figures 7 and 8 show that the required deceleration of the high-mobility wheeled vehicle (with limitation of the lateral acceleration) provided by the retarders is $1 \, m/s^2$, which is the same value as in the case without limitation of the lateral acceleration.

Thereby, we can conclude that regardless of the engine power and accelerations acting on the driver the required deceleration which must be provided by the units of the wear resistant brake system of the high-mobility wheeled vehicle does not exceed $1 \, m/s^2$.

It is easy to see that the lateral acceleration limit $0.1 \, g$ corresponds to the roll limit acceleration on an ice-covered road $\mu_{s_{\text{max}}} = 0.1$ [19]. It means that the results shown in figures 8 and 9 can be regarded as the results obtained for driving on ice, and the deceleration which must be provided by the wear resistant brake system does not depend on the tractive characteristics of the tire and is also not higher than $1 \, m/s^2$.

5. Conclusion
The analysis has shown that the deceleration which must be provided by the units of the wear resistant brake system of a high-mobility wheeled vehicle for obtaining the efficient average speed does not depend on the engine power and driving conditions, and is not higher than $1 \, m/s^2$.

The described method for the selection of the required deceleration of the high-mobility wheeled vehicles at the design stage provides development of the specifications and torque – speed curves for the units of the wear resistant brake system.

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