Selection of the torque – speed curves for the units of a wear-resistant brake system of high-mobility wheeled vehicles

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Abstract. With the growth of the power-to-weight ratio of the modern high-mobility wheeled vehicles, the capacity of their working brake systems is no longer sufficient for operation at the high speeds owing to high thermal loads of the brakes. In order to increase the average speed of the vehicle during the long-term downslope decelerations, as well as at the intensive use of the wheel brakes it is necessary to use additional devices – exhaust and retarder brakes. This kind of devices include the exhaust brake installed inside the internal combustion engine and the transmission hydro- and electrodynamic brakes (retarder brakes). The use of an electric machine operating in generator mode for storing the braking energy in an electric storage device is also a promising option. The combination of such additional devices is called “a wear-resistant brake system”. The research is focused on the selection of the torque – speed curves of a wear-resistant brake system providing higher average speeds of the wheeled vehicles and lower loads of the working brake system. Selection of the torque – speed curves of the units of a wear-resistant brake system is carried out by the analysis of the wheeled vehicle operation in different driving cycles which are close to its real operation cycles. These cycles are usually generated for a stationary case with the use of a quasi-stationary mathematical model of the vehicle motion. In the model, the vehicle – terrain interaction is described by means of the equations of the vehicle motion including stochastically generated roads. The analysis of the vehicle operation in different driving cycles provides stochastic data on the loads of the wear-resistant brake system as a whole, which allow selecting the torque – speed curves of its units depending on their operational features. The proposed method provides selection of the required torque – speed curves of the wear-resistant brake system units: the internal combustion engine (operating in the brake mode), the electric machine, and the retarder. Selection of the parameters of the electric machine is the first step for the solution of the next important problem – selection of the required capacity of the electric energy storage device. The results of the analysis provide selection of such combination of the characteristics for each unit of the wear-resistant brake system that will ensure the required deceleration of the vehicle.

1. Introduction

The proposed method of the selection of the torque – speed curves for the units of a wear-resistant brake system of high-mobility wheeled vehicles is based on the following assumption. 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 [1, 2]. It means that the speed of the vehicle in the given road conditions is rather determined by the driver than by the limitations imposed by the braking/tractive performance of the vehicle or by its stability during maneuvering [3, 4].

Thus, a statistically average driving cycle of the vehicle can be obtained with the use of stochastically generated routes which are realizations of random functions of external disturbances such as: the trajectory curvature, the maximum friction coefficient between the tire and the road, rolling resistance coefficient and the slope angle of the road surface [5, 6, 7, 8, 9, 10].

The obtained driving cycle of the vehicle allows selecting the required torque – speed curves of the retarders providing the given deceleration. At the same time, in order to increase the average speed of the vehicle and also decrease the loads on the working brake system it is reasonable to have the vehicle deceleration due to the operation of the wear-resistant brake system the same as the one realized at the service braking (and not higher $a^* = 1 \text{ m/s}^2$ [11, 12]).

2. Results of the quasi-stationary simulation

The required torque – speed curves of the wear-resistant brake system are selected from the driving cycles of the vehicle obtained by the quasi-stationary model of the vehicle motion and stochastically generated roads taking into account the limitations on the longitudinal and lateral accelerations determined by the tractive performance of the tire, the trajectory curvature, the vehicle power plant performance, and physiological capability of the driver to sustain accelerations (see figure 1 and figure 2) [13].

![Figure 1. A high-mobility vehicle driving cycle limited by the deceleration $a^* = 1$.](image1.png)

![Figure 2. A driving cycle of the high-mobility vehicle for the deceleration $a^* = 1$, the speed being limited by the physiological capability of the driver to sustain acceleration (0.1g).](image2.png)
Stochastically generated routes used for the generation of the presented driving cycles represent a consecutive set of segments. Each segment features its own slope, curvature, coefficient of rolling resistance and friction coefficient, these parameters remaining constant within the segment.

Thus, the dependence of the magnitude of the traction/braking force \( F_{\text{req}} \) required to maintain a given law of motion (on each \( i \)-th segment of the route) can be obtained using a quasi-stationary mathematical model of the vehicle motion and road parameters [14, 15].

\[
F_{\text{req}} = \left( m \delta + J \Delta k \right) a_i + m g \left( f_i \cos(\alpha_i) + \sin(\alpha_i) \right) + k_w \Delta k t + J \Delta k \frac{\Delta s}{\Delta t} v_{\text{tri}}^2,
\]

where \( m \) is the mass of the high-mobility vehicle; \( J \) 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 drivetrain; \( a_i \) is the acceleration of the vehicle during transition from segment \( i-1 \) to segment \( i \) of the route; \( f_i \) is the rolling resistance coefficient during transition from segment \( i-1 \) to segment \( i \) of the route; \( \alpha_i \) is the slope angle of the road during transition from segment \( i-1 \) to segment \( i \) of the route; \( k_i \) is the curvature of the trajectory during transition from segment \( i-1 \) to segment \( i \) of the route; \( \Delta k \) is the change of the curvature at transition from coordinate \( i-1 \) to coordinate \( i \) along the route; \( v_{\text{tri}} \) 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 limit longitudinal acceleration; \( g \) is the acceleration of gravity; \( k_w \) is the aerodynamic force coefficient, \( k_w = 0.5 c_x \rho_w \); \( c_x \) is the aerodynamic drag coefficient; \( \rho_w \) is the air density; \( F_{\text{front}} \) is the frontal area of the vehicle.

The value of \( a_i \) can be calculated with the following formula:

\[
a_i = \frac{(v_i^2 - v_{i-1}^2)}{2\Delta s_i}
\]

To simplify the analysis, the tractive/braking force \( F_{\text{req}} \) and the velocity \( v_i \) of the vehicle can be transferred to the required torque \( M_{\text{req}} \) and angular speed \( \omega_{\text{req}} \) of the gearbox output shaft (taking into account the assumptions of the dynamic model [15]):

\[
M_{\text{req}} = F_{\text{req}} r_{k0}/u_{mg}, \quad \omega_{\text{req}} = v_i u_{mg}/r_{k0}
\]

where \( r_{k0} \) is the wheel radius without slipping; \( u_{mg} \) is the gear ratio of the drivetrain from the gearbox output shaft to the drive wheels.

3. Selection of the characteristics of a wear-resistant brake system

Distribution of the operation cycles of the high-mobility vehicle brake system (by way of example the power-to-weight ratio was selected 15 kW/t) on the surfaced road and a flat terrain is shown in figure 3 (only the braking cycles are shown, here and further in the article the characteristics of the wear-resistant brake system are transferred to the gearbox output shaft):
Figure 3. Distribution of the operation cycles of the high-mobility vehicle brake system on a flat terrain: a) without limitations on the lateral acceleration; b) with limited lateral acceleration ($a_y = 0.1g$) (the characteristics of the brake system are transferred to the gearbox output shaft).

The histograms show that there are two regions of the brake system operation where braking by two different types of retarders is effective:

- a region of the high torques and rotation speeds of the gearbox output shaft (effective braking by the hydrodynamic retarder);
- a region of the low torques and rotation speeds of the gearbox output shaft (effective braking by the internal combustion engine (ICE) or an electric machine operating in the generator mode [16] to make the high-mobility vehicle more energy efficient.

Similar analysis of the operation on a surfaced road on an undulating terrain has shown the following (see figure 4).
The figures 3 and 4 show that in the case of an undulating terrain there are regions with high braking torque, which should be generated by the service brake system [17], [18], [19] since the retarder suitable for such operation cycles would be too bulky to be used on a high-mobility vehicle. At that, the retarder operation regions remain on the same “position” on the histogram.

Hence, further selection of the torque – speed curves of the retarders can be effectively performed from the data obtained on a flat terrain.

For this end, the cumulative distribution functions for the required power of the brake system \( N_{req} = \frac{M_{req}}{\omega_{req}} \) and the torque on the gearbox output shaft in the operation cycles in question have been obtained: one (see figure 2) taking into account lateral acceleration limits determined by the physiological capabilities of the driver (see figures 5 and 6) and one (see figure 1) without these limitation.

These results show that the effective power of the retarders with probability 95 % will not exceed \( N_{95\%} \) and the gearbox output torque – \( M_{95\%} \).

These two parameters are used for description of the “external curve” of the wear-resistant brake system as a whole.
Figure 5. Cumulative distribution functions of the high-mobility vehicle brake system parameters obtained without limitations on the lateral acceleration: a) power; b) braking torque (transferred to the gearbox output shaft).

In order to select the required power of the electric machine $N_{\text{ED}}^{\text{max}}$, used as a retarder we need to analyze the density and cumulative distribution functions of the braking power for the driving cycle with limitations on the lateral acceleration (see figure 2). Since, in this particular case according to figure 3 the electric machine is supposed to be used most often (see figure 6).

The density function has two distinctive maximums, the first of which corresponds to the region of the effective use of the electric machine. Its operation in that region will make the high-mobility vehicle more power effective by means of storing the electric power in a storage device and using it later at the acceleration.

Besides, according to figure 6c, the gearbox output shaft torque having probability 95% of not to be overrun corresponds to the same value $M_{95\%}$ as in the case without lateral acceleration limits (see figure 5b). This result means that regardless of the driving cycle the given deceleration $1 \text{ m/s}^2$ can be provided by the wear-resistant brake system torque of $M_{95\%}$ for all rotational speeds of the gearbox output shaft.
Figure 6. Density function (a) and cumulative distribution function (b) of the effective braking power and cumulative distribution function (c) of the braking torque of the high-mobility vehicle brake system with limited lateral acceleration \(a_y = 0.1 g\) (the characteristics of the brake system are transferred to the gearbox output shaft).
The selected parameters \((N_{95\%, M_{95\%}})\) of the wear-resistant brake system and the selected power of the electric machine \(N_{EM}^{\text{max}}\) allow finding the required parameters of each of the units of the brake system.

Since the selection of the engine is based on the required driving performance of the vehicle (obtained by calculation), in the braking mode engine characteristics can be considered to be known and determined by its design parameters: displacement, diameter of the intake/exhaust valves, etc. On the design stage it is reasonable to calculate a combined torque – speed curve of the ICE and its gearbox in the braking mode of the constant power curve. So, by way of example, we will assume that in the braking mode engine power transferred to the gearbox output shaft makes 40% of its tractive power \(N_{ICE}^{\text{maxT}} = 6 \text{ kW/t}\) that corresponds to the actuation of a "mountain brake" [16] (the tractive power-to-weight ratio is 15 kW/t).

When an electric machine is used in the drivetrain of a high-mobility vehicle, it is reasonable to install it upstream the gearbox. This solution would provide an extended operating range of the electric machine and operation of the vehicle at a low speed with additional electric drive. Thereby, we will select the vehicle drivetrain layout shown in figure 7. At that, the combined torque – speed curve of the electric machine and the gearbox can be presented as the constant power curve corresponding to the value \(N_{ED}^{\text{max}}\).

Thus, the required torque – speed curve \(M_{RET}(\omega_{gb})\) of the hydrodynamic retarder transferred to the gearbox output shaft is determined by the difference (the curves of all the units are transferred to the gearbox output shaft as shown in figure 8):

\[
M_{RET}(\omega_{gb}) = M_{95\%} - M_{ICE}(\omega_{gb}) - M_{ED}(\omega_{gb}),
\]

where \(M_{ICE}(\omega_{gb})\) is the dependence of the ICE braking torque (transferred to the gearbox output shaft) from the rotational speed of the gearbox output shaft; \(M_{ED}(\omega_{gb})\) is the dependence of the electric machine braking torque (transferred to the gearbox output shaft) from the rotational speed of the gearbox output shaft.
Figure 8. Required torque – speed curve of the hydrodynamic retarder: a) with an electric machine in the drivetrain; b) without an electric machine in the drivetrain.

The next step is the calculation of the hydrodynamic brake external torque – speed curve based on the required one $M_{\mathrm{RET}}(\omega_{gb})$.

The torque – speed curve of the filled vane-type hydrodynamic retarder is a parabola passing through the point $M_{\mathrm{RET}}(0) = 0$ [21]. Therefore the setting of the hydrodynamic retarder must be designed so that the said parabola would touch the required torque – speed curve of the retarder (see figure 9).

The operation under the torque – speed curve of the filled hydraulic retarder is provided by controlling the volume of the working fluid in the vane section of the retarder, i.e. limit of the torque – speed curve on the right is determined not by the retarder engineering but by its control system operation.
For example, in the case of a long duration braking by the retarder the limit is determined by the maximum heat power that can be dissipated by the cooling system radiator. If this limit is exceeded the coolant and hence the engine will be overheated. At that, in order to get the maximum efficiency of the retarder the heat power dissipated by the radiator must be no less than:

\[ N_{RAD} = N_{95\%} - N_{ICE}^{\text{max RT}} - N_{ED}^{\text{max}}. \] (5)

Since in general case the cooling system of the high-mobility vehicle engine features high thermal inertia, much higher brake power can be utilized at braking (higher than \( N_{RAD} \)), but for a short period of time. In this case, the limit for the torque – speed curve of the retarder on the right will be the maximum heat power \( N_{RET}^{\text{max}} \) at which the coolant is not boiling when passing through the retarder (6). Boiling of the coolant should be avoided since it would lead to a considerable decrease in the working fluid density and a lower effective braking torque.

\[ N_{RET}^{\text{max}} = c_f \rho_f Q_f \Delta t, \] (6)

where \( c_f \) is the heat capacity of the working fluid at the critical temperature; \( \rho_f \) is the density of the working fluid at the maximum temperature; \( Q_f \) is maximum flow rate of the working fluid in the cooling system; \( \Delta t \) is the maximum temperature difference (\( \Delta t \approx 110 \text{°C} - 90 \text{°C} = 20 \text{°C} \)).

Thus, the reasonable operational power limit of the hydrodynamic retarder is \( N_{RET}^{\text{max}} \). At the same time, the load on the hydro-retarder must be lowered in case the temperature of the coolant reaches the critical value (~110°C).

Bearing all this in mind the external torque – speed curve of the retarder can be selected in the form shown in figure 9.

![Figure 9](image)

(a)

(b)

**Figure 9.** External torque – speed curve of the hydrodynamic retarder: a) with an electric machine in the drivetrain; b) without an electric machine in the drivetrain.
4. Conclusion

The proposed method provides selection of the required torque – speed curves of the units of the wear-resistant brake system of a high-mobility wheeled vehicle: an internal combustion engine (working in brake mode), an electric machine and a retarder the combination of which will ensure the required deceleration of the vehicle.

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