A control algorithm for simulation of real-world operating conditions for the drivetrain of an all-wheel drive vehicle with individually driven wheels on a chassis dynamometer

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Abstract: The article presents a new control algorithm for the simulation of real-world operating conditions for the drivetrain of an all-wheel drive transportation vehicle with individually driven wheels on a chassis dynamometer. The main idea of the algorithm is that the wheel torques and angular velocities on a chassis dynamometer are synchronized with the corresponding parameters obtained on a road. For the verification of the algorithm, the authors used mathematical models of an 8 × 8 vehicle with individually driven wheels on a chassis dynamometer and on a road. Wheel torques and angular velocities obtained by the simulation of the straight-line motion of the vehicle on a road with longitudinal slope were used for the control of the driving torques in the model of the chassis dynamometer. The

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PUBLIC INTEREST STATEMENT
It is known that the main disadvantage of the full-scale vehicle testing on the chassis dynamometer is the difficulty to simulate the actual road conditions on the roller surface. The authors of the article solve this problem by using the chassis dynamometer roller control algorithms based on the mathematical simulation of the vehicle motion. This approach application would allow replacing much of the expensive road tests by chassis dynamometer testing.
simulation has proved capability of the proposed algorithm to reproduce dynamic loads in the drivetrain on a chassis dynamometer identical to the ones registered on a road.

Subjects: Automotive Design; Dynamics & Kinematics; Automotive Technology & Engineering

Keywords: chassis dynamometer control; drivetrain loads simulation; individual wheel drive vehicle dynamics simulation

1. Introduction
In recent years, Bauman Moscow State Technical University and KAMAZ PTC, the leading Russian manufacturer of the commercial vehicles, have performed a number of joint projects aimed at the development of special-purpose vehicles including KamAZ-ARCTIC 6 × 6 and 8 × 8 (“KamAZ-Arktika» проходит испытания,” 2020), wheeled vehicles for the mining industry and other vehicles. Road performance testing was an essential part of the wheeled vehicle engineering in each of the projects. It has often been demonstrated that it is very difficult and therefore costly to secure high-quality repeatable engineering data from road tests. Chassis dynamometer testing is a good alternative to road testing since it provides flexible control over tractive forces, speed modes, external load conditions and simulation of long steady-state modes, which allows carrying out all the necessary measurements, including video- and photo-recording of the experiment (Pfister et al., 2009, Galko et al., 2014, Ding-Tsair et al., 2008, Hammerschmidt, 2016, Albers et al., 2012, Gladov & Petrenko, 2010). As it is shown in (Petrushov, 2008), the advantages of the chassis dynamometer testing include capabilities to provide a wide range of experimental modes independent from environmental conditions and to automate testing procedures. At the same time, it is known that the main drawback of the vehicle testing on a chassis dynamometer is the difference between tire rolling conditions on a roller and on a road, which is difficult to overcome. The elimination of this disadvantage would make the chassis dynamometer testing the most preferable alternative to the road performance testing in terms of efficiency.

Figure 1(a) shows a principle of the simulation of the road conditions on a chassis dynamometer.

Parameters of the vehicle motion in different operation modes are recorded during road tests and used as the inputs to the chassis dynamometer actuation control system. Control of the actuation motors provides torques and kinematic parameters of the vehicle on a chassis dynamometer identical to the ones registered on a road. In order to realize this ideal situation a proper control algorithm for the actuator motors of the chassis dynamometer rollers is needed. One of the ways for its development is to use computer simulation of both the chassis dynamometer testing and the road testing. Figure 1(b) shows a block diagram of this virtual testing, in which a real vehicle on a road and on a chassis dynamometer is replaced by corresponding mathematical models. The virtual testing also needs a mathematical description of the driver inputs and a mathematical model of the chassis dynamometer control system. The control system comprises a computational unit for the generation of the control inputs as a function of the vehicle motion parameters. It must be noted that the inputs on the dynamometer should be identical to the driver inputs on the road.

2. Mathematical model of the chassis dynamometer test
The model was developed for an 8 × 8 vehicle with individually driven wheels. The dynamical model of the vehicle on a chassis dynamometer is shown in Figure 2.

Even reference numbers are used for the right side wheels of the vehicle, odd ones—for the left side wheels (not shown in Figure 2).

One of the model assumptions is that the device retaining the vehicle on the rollers is a spring—damper system. The force holding the vehicle on the chassis dynamometer acts only along the
Figure 1. Simulation of the road conditions on a chassis dynamometer.

(a) chassis dynamometer testing; (b) virtual testing.

Figure 2. Dynamical model of the vehicle on a chassis dynamometer.

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x-axis. Displacement of the vertical axis of the \( i \)th wheel along the roller vertical axis is neglected and compensated by the \( i \)th rolling resistance torque.

Equations of motion for the “vehicle—rollers” system can be written as follows:

\[
\begin{align*}
& m \cdot a_x = \sum_{i=1}^{8} R_{xki} - P_{br}, \\
& J_{ki} \cdot \epsilon_{ki} = T_{kri} - T_{fki} - R_{xki} \cdot r_i, \\
& J_{bi} \cdot \epsilon_{bi} = R_{xbi} \cdot r_{bi} - T_{fbi} + T_{bi},
\end{align*}
\]

(1)

here \( i = 1\ldots8 \); \( m \) is the vehicle mass; \( a_x \) is the longitudinal acceleration of the vehicle center of mass on the chassis dynamometer (its value is near zero); \( R_{xki} \) is the longitudinal force on the \( i \)th wheel from the \( i \)th roller; \( R_{xbi} \) is the longitudinal force of the \( i \)th roller from the \( i \)th wheel; \( P_x \) is the drawbar force; \( \epsilon_{ki} \) is the angular acceleration of the \( i \)th wheel; \( \epsilon_{bi} \) is the angular acceleration of the \( i \)th roller; \( J_{ki}, J_{bi} \) are the moments of inertia of the \( i \)th wheel and \( i \)th roller, respectively; \( T_{kri} \) is the driving torque on the \( i \)th wheel; \( T_{fki} \) is the rolling resistance torque of the \( i \)th wheel on the \( i \)th roller; \( T_{fbi} \) is the rolling resistance reaction torque on the \( i \)th roller from the \( i \)th wheel; \( T_{bi} \) is the braking (driving) torque generated by the actuator of the \( i \)th roller; \( r_i, r_{bi} \) are the distances from the contact point to the axis of the \( i \)th wheel and \( i \)th roller, respectively.

There are different layouts of chassis dynamometers known in the community. In the state-of-the-art chassis dynamometers, the rollers are usually coupled with electric drive motors (see Figure 3) providing both braking and driving torques.

Let us assume that the driving torque \( T_b \) generated by the actuating motor can be both negative (braking torque) and positive (driving torque).

Now we shall consider how to calculate the terms of system of Equations (1).

Wheel longitudinal force from the roller depends on the normal force and friction coefficient:

\[
R_{xki} = R_{xbi} = \mu_{xi} \cdot R_{zi}
\]

(2)

here \( \mu_{xi} \) is the actual coefficient of friction between the \( i \)th wheel and \( i \)th roller; \( R_{zi} \) is the normal force on the \( i \)th wheel.
Relationship between the friction coefficient and longitudinal slip can be approximated by different formulae (Petrushov, 2008; Rozhdestvenskiy & Mashkov, 1982). Authors of (Petrushov, 2008) have proposed the following relationship between the friction coefficient and longitudinal slip:

$$\mu_x = \mu_{x_{\text{max}}} \cdot \left(1 - e^{-\frac{S}{S_0}}\right) \cdot \left(1 + e^{-\frac{S}{S_1}}\right)$$  \hspace{1cm} (3)

Here $\mu_{x_{\text{max}}}$ is the friction coefficient at full slip; $S$ is the longitudinal slip; $S_0$ and $S_1$ are the shape parameters of the $\mu(S)$ curve.

Longitudinal slip coefficient for the $i$th wheel on a roller is defined as

$$S_i = \frac{\omega_{ki} \cdot r_i - \omega_{bi} \cdot r_b}{\omega_{ki} \cdot r_i},$$  \hspace{1cm} (4)

Here $r_i$ is the tire rolling radius of the $i$th wheel on the $i$th roller at zero slip. Conditions of the interaction of each wheel with respective roller are assumed to be identical. The surface of the roller is essentially rigid; therefore, we can assume that all the radii $r_b$ are equal to the value given in the chassis dynamometer specification.

Rolling resistance torque $T_{f_{ki}}$ of the $i$th wheel is mostly determined by the internal energy losses and can be approximated by the following expression (Gnadler et al., 1995):

$$T_{f_{ki}} = f \cdot R_{zj} \cdot r_i^-$$  \hspace{1cm} (5)

Here $f$ is the rolling resistance coefficient for the wheel on the roller.

Let us assume that the rolling resistance torque transmitted to the roller from the wheel is $T_{f_{bi}}$.

$$T_{f_{bi}} = T_{f_{ki}}$$  \hspace{1cm} (6)

Normal forces $R_j$ can be found from the following system of equations under the assumption that the vehicle is symmetrical about the vertical plane containing its longitudinal axis:

$$\begin{align*}
\sum_{j=1}^{4} R_{ij} &= m \cdot g; \\
\sum_{j=1}^{4} R_{ij} \cdot x_{ij} + m \cdot a_x \cdot H_c + P_{kr} \cdot H_{kr} + \sum_{i=1}^{8} T_{f_{ri}} &= 0; \\
R_{21} \cdot (x_{k2} - x_{k3}) + R_{22} \cdot (x_{k3} - x_{k1}) + R_{23} \cdot (x_{k1} - x_{k2}) &= 0; \\
R_{21} \cdot (x_{k2} - x_{k4}) + R_{22} \cdot (x_{k4} - x_{k3}) + R_{24} \cdot (x_{k1} - x_{k2}) &= 0.
\end{align*}$$  \hspace{1cm} (7)

Here $j = 1-4$ is the index of the vehicle axle; $R_{ij}$ is the normal force on the $j$th axle; $H_c$ is the height of the center of mass of the vehicle; $H_{kr}$ is the height of the drawbar force $P_{kr}$ application point, $x_{ki}$ is the coordinate of the $j$th axle along the $x$-axis from the vehicle center of mass.

Additional equations for the solution of system (7) are derived under the assumption that the endpoints of the vectors of the normal forces belong to the same plane. It will be true if we make the following assumptions: the suspension of each wheel is independent and is equivalent to a spring without lateral resistance, vehicle body dynamics is neglected.

The driving torques on the wheels are assumed to be distributed proportional to the relative normal loads on each wheel (Smirnov, 1990).

The driving (actuating) torque on an $i$th wheel is calculated under the condition that the wheel is driven by an electric motor with a hyperbolic torque/speed curve and limited maximum values of torque and speed.
Driving torque $T_{kri}$ is defined by

$$T_{kri} = K_{1i} \frac{N_{d,\text{max}}}{\sum_{i=1}^{n} \omega_{di}} \cdot h_G,$$

(8)

here $N_{d,\text{max}}$ is the maximum power of the $i$th electric motor; $h_G$ is the level of use of the total power of all the electric motors (a control input similar to the accelerator pedal position), its values are within the range $[0, 1]$; $n$ is the number of the vehicle wheels, $\omega_{di}$ is the angular velocity of the $i$th electric motor output shaft; $K_{1i}$ is the correction coefficient.

The correction coefficient is defined by

$$K_{1i} = \frac{R_{zi}}{\frac{\sum R_{zi}}{n}},$$

(9)

Drawbar force $P_{kr}$ retaining the vehicle on the roller can be defined by the spring and damper parameters of the holding device:

$$P_{kr} = c \cdot x + b \cdot V,$$

(10)

here $x$ is the longitudinal displacement of the vehicle center of mass; $c$ is the longitudinal stiffness of the holding device; $b$ is the longitudinal damping of the holding device.

Let us assume that the torque on the roller depends on the chassis dynamometer parameters and control parameters and can be calculated by

$$T_{bi} = \frac{T_{b,\text{max}}}{C_{1}} \cdot h_{bi},$$

(11)

here $T_{b,\text{max}}$ is the maximum torque on the roller (determined by the dynamometer specification); $h_{bi}$ is the maximum torque level of the $i$th roller $[-1 \ldots 1]$.

The mathematical model of the vehicle on a chassis dynamometer was implemented in Matlab/Simulink (Gorelov et al., 2012). The general block diagram of the model shown in Figure 4 includes two subsystems: VEHICLE—subsystem with the equations of the vehicle motion, CHASSIS DYNAMOMETER—subsystem with eight equations of the roller motion, which are shown in general form in the last equation of system (1). Angular velocities $\omega_{bi}$ from the CHASSIS DYNAMOMETER subsystem are transmitted to the VEHICLE subsystem for calculation of the slips $S_i$. Thus, the first and the second subsystems are linked by the values of the longitudinal forces on the rollers ($R_{xi} = R_{xki} = R_{xbi}$). These forces are calculated in the VEHICLE subsystem and used in the CHASSIS DYNAMOMETER subsystem (see Figure 4).
3. Mathematical model of the road test

Road tests were simulated with the use of the vehicle straight-line motion model described in detail in (Smirnov, 1990). The model of the vehicle motion is shown in Figure 5.

The vehicle in the model is regarded as a rigid body moving along a smooth rigid road with longitudinal slope. Its system of equations of motion enables calculation of the current values of the accelerations by the values of the forces acting on the vehicle and wheel torques:

\[
\begin{align*}
\dot{m} \cdot \alpha &= \sum_{i=1}^{8} R_{xki} - m \cdot g \cdot \sin \alpha - P_w; \\
J_{ki} \cdot \epsilon_{ki} &= T_{rki} - R_{xki} \cdot r_{di} - T_{fki},
\end{align*}
\]  

(12)

here \( \alpha \) is the road slope angle.

Wheel slip on a road in the driving mode is calculated by

\[
S_i = \frac{|V_x - \omega_i \cdot r_{ki}|}{\max(V_x, \omega_i \cdot r_{ki})}
\]

(13)

here \( r_{ki} \) is the rolling radius of the \( i \)th wheel at zero slip.

When the vehicle moves on the road, normal forces \( R_{zi} \) can be found from the following system of equations:

\[
\begin{align*}
\sum_{j=1}^{4} R_{zj} &= m \cdot g \cdot \cos \alpha; \\
\sum_{j=1}^{4} R_{zj} \cdot x_{kj} &= P_w \cdot H_w + m \cdot a_x \cdot H_c + m \cdot g \cdot \sin \alpha \cdot H_c + \sum_{i=1}^{8} T_{fki} = 0; \\
R_{z1} \cdot (x_{k1} - x_{k2}) + R_{z2} \cdot (x_{k3} - x_{k1}) + R_{z3} \cdot (x_{k1} - x_{k2}) &= 0; \\
R_{z1} \cdot (x_{k2} - x_{k3}) + R_{z2} \cdot (x_{k4} - x_{k2}) + R_{z4} \cdot (x_{k1} - x_{k2}) &= 0.
\end{align*}
\]

(14)

\( j = 1 \ldots 4 \) is the index of the vehicle axle; \( H_w \) is the height of the air drag force application point.

4. Control algorithm for simulation of road conditions on a chassis dynamometer

Rolling resistance coefficients \( f \) and longitudinal friction coefficients \( \mu_{x\text{max}} \) are different for the vehicle on a road and on the rollers, i.e. rolling conditions on the chassis dynamometer and on the road are different. The control algorithm of the chassis dynamometer should provide rolling conditions on the rollers identical to the ones on a road. During continuous (or discrete) simulation of the vehicle motion on a road, the model monitors the wheel torques which are calculated in the same manner as it is done on the chassis dynamometer,

Figure 5. Dynamical model of the vehicle on a road.
i.e. by Equation (8). Therefore, when the torques $T_{kri}$ on the chassis dynamometer are synchronized with the driving torques $T_{di}$ on the road and the power consumption of all the electric motors is same, agreement between the angular velocities of each $i$th wheel on the chassis dynamometer and on the road can be provided. Agreement between the driving torques and between the angular velocities mean that the drivetrain loads obtained on the road are fully reproduced on the chassis dynamometer.

Figure 6 shows the virtual testing model diagram. The mathematical model of the vehicle on a chassis dynamometer is implemented as the CHASSIS DYNAMOMETER TEST subsystem. The model of the vehicle on a road is included in the general model as the ROAD TEST subsystem, which solves the vehicle equations of motion with parameters identical to the ones on the chassis dynamometer, but for the conditions different from the ones on the rollers.

During simulation, torques $T_{kri}$ and $T_{di}$ are fed from the ROAD TEST and CHASSIS DYNAMOMETER TEST subsystems to the CHASSIS DYNAMOMETER CONTROL SYSTEM MODEL subsystem (see Figure 6). Right-hand side of the third equation of system (1) contains torques $T_{bi}$ on the rollers for each actuating motor. These torques can be controlled to provide agreement between the driving torques on the road and on the chassis dynamometer.

In order to provide agreement between the torques $T_{kri}$ and $T_{di}$ the model varies the control inputs $h_{bi}$ within the range $[-1, 1]$, which changes the magnitude and direction of the torques applied to the rollers, as a function of the normalized difference $\Delta_i(T_{di}/T_{kri})$:

$$\Delta_i(T_{di}/T_{kri}) = \frac{T_{di} - T_{kri}}{T_{di}}$$

This algorithm shows good results for a single wheel as it was demonstrated in (Kotiev et al. 2014).

In the model, control input for the accelerator pedal of the vehicle on the chassis dynamometer is fully synchronized with the corresponding input during the road test. During real chassis dynamometer testing, this function is supposed to be implemented by an automated system.

5. Verification of the developed control algorithm

Below there is an example of the implementation of the proposed control principle during virtual testing of an all-wheel drive four-axle vehicle. Model parameters used in the simulation are given in Table 1.

The $\mu_x(S)$ characteristics used during simulation for the vehicle wheels on the rollers and on the road are shown in Figure 7.
Initial speed on both the chassis dynamometer and on the road is $V_0 = 1$ km/h. Slope angle $\alpha = 10^\circ$. 

Table 1. Model parameters used in simulation

| Parameter                                      | Value   |
|-----------------------------------------------|---------|
| Acceleration of gravity, m/s$^2$              | 9.81    |
| Vehicle mass, kg                              | 60,000  |
| Wheel moment of inertia, kg·m$^2$             | 83      |
| Roller moment of inertia, kg·m$^2$            | 300     |
| Distance from the contact point to the wheel axis, m | 0.725   |
| Distance from the contact point to the roll axis, m | 0.6     |
| Rolling resistance coefficient on the chassis dynamometer, $f_{road}$ | 0.016   |
| Rolling resistance coefficient on the road, $f_{road}$ | 0.1     |
| Maximum torque on the roller, T$_{max}$, N·m | 10,000  |
| Electric motor maximum output torque, T$_{max}$, kN·m | 39      |
| Electric motor maximum power, $N_{max}$, kW   | 60      |
| Longitudinal stiffness of the holding device, c, N/m | 5e7     |
| Longitudinal damping of the holding device, b, N·s/m | 2.5e7   |
| Height of the vehicle center of mass, H$_c$, m | 1.925   |
| Height of the drawbar force application point on chassis dynamometer, H$_{kr}$, m | 0.88    |
| Coordinate of the 1st axle, x$_{k1}$, m       | 4.245   |
| Coordinate of the 4th axle, x$_{k4}$, m       | -1.905  |
| Sliding friction coefficient on the rollers, $\mu_{max_b}$ | 0.4     |
| Sliding friction coefficient on the road, $\mu_{max_r}$ | 0.7     |
| Shape parameter S0 of the $\mu_S(S)$ curve   | 0.04    |
| Shape parameter S1 of the $\mu_S(S)$ curve   | 0.15    |

Figure 7. $\mu_S(S)$ Dependencies for the wheel on the chassis dynamometer and on the road.
Chassis dynamometer control (torque variation) begins after 2 s from the motion start, i.e. before that moment the vehicle motion on the rollers is independent from the road conditions; control input $h_G$ on the road ramps up from 0.1 to 1 during the first second of motion and then remains constant. Control input $h_G$ of the chassis dynamometer is synchronized with the control input $h_G$ on the road. Total simulation time is $t = 10$ s.

Simulation results are shown in Figures 8-15. At the given conditions on the road with longitudinal slope angle $\alpha = 10^\circ$, the vehicle obtained maximum possible speed 2.8 m/s at 10 s (Figure 8).

Figure 9 shows that during the first 2 s the torque $T_d$ applied to the wheel of the first axle on the road and the torque $T_d$ applied to the same wheel on the roller are generated independently, i.e. the roller torque control input $h_b = 0$ (see Figure 10).
At time $= 2$ s, the roller driving torque control was initiated (see Figure 10) in order to equalize the driving torques on the road and on the roller as it is shown in Figure 9.

The time history of the wheel angular velocities (Figure 11) shows that, as during the first 2 s there was no resistance torque on roller 1 and the driving torque was generated as a function of redistribution of the normal forces, which are lower for a vehicle driving upslope than for a vehicle on the chassis dynamometer, angular velocity for the wheel on the roller was growing slower than the speed of the same wheel on the road (Figure 11).

Then, due to the control of the torque $T_y$ on roller 1 the driving torques became equal (see Figure 9). Since the electric motor torque is a function of its angular velocity and the equal power consumption is provided during the test on the road and on the roller, we get equal angular velocities starting from time $= 2$ s.
Maximum drawbar force was $10,6 \times 10^4$ N (see Figure 12).

Figures 13–15 show that after the start of the roller driving torque control (at 2 s), the driving torques (see Figure 13) and angular velocities (see Figure 15) for a wheel of the 4th axle on the road and on the chassis dynamometer became equal.

This effect was provided due to generation of the control input according to the command signal (Figure 14).

At the same time, during the first 2 s the wheel angular velocity on the roller was increasing faster than during the upslope motion on the road (see Figure 15).
This can be explained by the fact that during the upslope motion there was an additional load on the rear axle of the vehicle in comparison with the horizontally positioned vehicle on the chassis dynamometer.

6. Conclusions
The proposed control algorithm provides synchronization of the wheel torques and angular velocities on a chassis dynamometer with the corresponding parameters obtained on a road.

Simulation results have proved capability of the proposed control algorithm to simulate road conditions on a chassis dynamometer for the drivetrain of an 8 × 8 vehicle with individually driven wheels on a smooth rigid road with longitudinal slope, which parameters \( \mu_{\text{max}} \) and \( f \) differ from the ones of the rollers.
In the next stage of the research, it is planned to extend the control algorithm to the simulation of driving over deformable soil and rotation of the wheels with different speeds (e.g., during curvilinear motion).

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