# Simulation Modeling of the Multi-Axle Truck Dynamics

A. Keller, V. Sayakhov and S. Aliukov

***Abstract*— Experimental and theoretical researches and operation experience show that the acceleration dynamics, mobility and fuel efficiency of wheeled vehicles are considerably influenced by the circuit of power input to the traction wheels. It is particularly important to multiwheelers, for example, off-road dump trucks. To insure a high off-road performance of any vehicle the power distribution along the wheels must meet two basic requirements: 1) the wheel must operate in a free or close to it rolling mode with the minimum rolling resistance due to absence of an axial force and an absolute slip; 2) in case it is necessary to create an axial pulling power to overcome additional resistances (acceleration, rise, towed load), the power supplied to the wheel must not exceed the limit of the intensive ground failure as a result of excessive wheel slipping. In other words, wheel slipping must not exceed the value, at which the maximum pulling power is created. To obtain consistent results of the vehicle's dynamic processes the authors developed a mathematic model, which includes a description of all the vehicle's basic systems with real parameters. The created mathematical model allowed to implement the vehicle's motion processes for all the required motion modes in view of the dynamics of the vehicle's major units and subsystems. To model the vehicle dynamics we used CAD LMS AMESim package of physically-oriented mathematical modeling.**

***Index Terms*— All-wheel drive vehicles, control of movement, modeling, stability.**

## I. INTRODUCTION

OFF-ROAD dump trucks are generally operated at different road conditions, on asphalt, dirt road or sand. Mechanical power losses are registered in the vehicle at all the stages of power transfer. However, the power needed to overcome the motion resistance can be reduced by choosing an optimal traction wheel drive circuit. Thus, it is possible to increase the vehicle performance [1].

The power consumed on the rolling resistance can considerably grow in different road conditions. In this regard, the losses for the traction wheel rolling differ from the losses for the idle wheel rolling. It is connected with the fact that losses for outward rolling are added to hysteresis losses (equally inherent in the idle wheel). It, in its turn, depends on the torque supplied to the wheel. To reduce the wheel torque at preservation of the vehicle's traction force it is necessary to increase the number of traction wheels. Apart from the losses for the traction wheel slipping with regard to the bearing surface, it is also necessary to consider the power necessary for rotation of the units of the additional drive axle and the drive thereto. In this regard, the transmission losses are increased with the increasing number of traction wheels, and the losses for slipping of the traction wheels with regard to the bearing surface are decreased with the increasing number of the traction wheels.

It is possible to obtain reliable data on energy losses during the vehicle movement by means of experiments. Alongside with that, the experiments involve considerable financial expenses. In this connection, currently the researchers apply the simulation modeling method [2, 3, 4]. Mathematical models allow to obtain the necessary vehicle characteristics without any experiments at the design stage, that is why they have been actively used by motor manufacturers.

## II. OBJECT OF RESEARCH

The object of the research is KAMAZ-65222 off-road dump truck (Figure 1), specifications (Table I) [8] and design drive circuits (Figure 2, 3) [1].

Fig. 1. KAMAZ-65222 dump truck

The work was supported by Act 211 Government of the Russian Federation, contract № 02.A03.21.

The work was conducted with the financial support of the Ministry of Education and Science of the Russian Federation in the framework of the complex project to create a high-tech production "Creating high-tech production of a new generation of energy efficient transmissions for trucks and buses" under contract No. 02.G25.31.0142 dated "01" December 2015 between the Ministry of Education and Science of the Russian Federation and Public Joint-Stock Company "KAMAZ" in cooperation with the head executor of the research, development and technological work of civil purpose - Federal State Autonomous Educational Institution of Higher Education "South Ural State University (National Research University)".

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TABLE I
SPECIFICATIONS OF KAMAZ-6522

| | |
|---|---|
| Weight parameters and loads | |
| Curb weight, kg | 14350 |
| load on the rear bogie, kg | 8000 |
| load on the front axle, kg | 6350 |
| Full weight of the vehicle, kg | 34000 |
| load on the rear bogie, kg | 26000 |
| load on the front axle, kg | 8000 |
| Engine | |
| Engine model | 740.632-400 |
| Gearbox | |
| Transmission model | ZF 16S1820 |
| Drive gear | |
| Transmission ratio | 6,88 |
| Climbing angle, no less than, % | 25 |
| External turning radius, m | 12 |

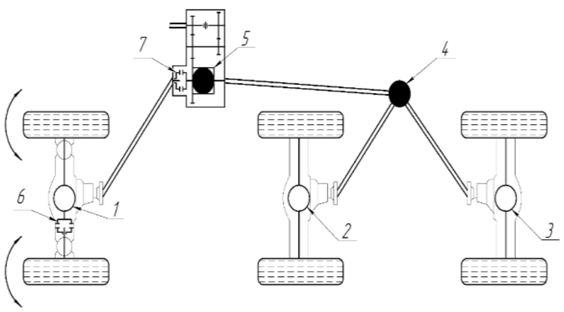


Fig. 2. Design drive circuits
(1,2,3 – Simmetrical differentials; 4 – Simmetrical locking differential; 5 – Asymetrical locking differential; 6,7– overriding clutches).

## III. DESCRIPTION OF THE MATHEMATICAL

To obtain reliable results of the vehicle's dynamic processes the mathematical model must include a description of all the vehicle's basic systems with real parameters of the research object. The design pattern of a vehicle with forces acting in the linear motion mode is shown in Figure 3.

To model the vehicle dynamics we use CAD (package of physically-oriented mathematical modeling) LMS AMESim. The created mathematical model allows us to implement the vehicle's motion processes for all the required motion modes in view of the dynamics of the vehicle's major units and subsystems.

The general view of the model is shown in Figure 4. This model consists of all the most important subsystems: engine, transmission, suspension and body. It describes all the necessary physical characteristics of the vehicle and the environment: weight and inertia, force of the wheel's interaction with the road carpet, aerodynamic resistance, etc.

The transmission units and motion conditions influence the truck body. Let us write the equations for the body's motion [4]:

$$\begin{cases} \dot{\omega} = T_1 - \dfrac{G_s h_{COG} \lambda (\dot{V}_x - \dot{V}_y \omega)}{J_z} \\ \dot{V}_x = V_y \omega + T_2 \\ \dot{V}_y = V_x \omega + T_3 \end{cases}$$

$$T_1 = \frac{\begin{bmatrix} \sum_{i=1}^{2} R_{y12}^{i} a + \sum_{i=1}^{2} R_{y2}^{i} b_2 + \sum_{i=1}^{2} R_{y3}^{i} b_3 + \\ +0{,}5\left(R_{x1}^{right} + R_{x1}^{left}\right) B_1 + 0{,}5\left(\sum_{j=2}^{3} R_{xy}^{right} + \sum_{j=2}^{3} R_{xy}^{left}\right) B_2 \end{bmatrix}}{J_z};$$

$$T_2 = \frac{\sum_{j=2}^{3} \sum_{i=1}^{2} R_{xj}^{i}}{G};$$

$$T_3 = \frac{\sum_{j=2}^{3} \sum_{i=1}^{2} R_{yj}^{i}}{G}.$$

where, $\dot{\omega}$ – vehicle's slew rate; $G_s$ – total sprung weight; $G$ – vehicle's weight; $h_{COG}$- height to the center of gravity; $\lambda$ – banking angle; $V_x$- longitudinal speed; $V_y$ – lateral speed; $J_z$ – body's inertia moment; $R_x$- wheel's longitudinal response; $R_y$- wheel's lateral response; $a$ – distance between the front axle and the center of gravity; $b_2$ – distance between the center of gravity and the intermediate axle; $b_3$ - distance between the center of gravity and the rear axle; $B_2$- width of the front track; $B_3$ – width of the rear track.

"The driver model" consists of a set of logical elements. It solves the following tasks: position control of the throttle blade, clutch pedal, brake pedal, choosing the transmission speed of the gearbox and the turn angle of the steering wheel.

The power unit and the transmission include: engine, which characteristics depend on the position of the throttle blade, configured to evaluate fuel efficiency; clutch model, connection/disconnection are controlled by the clutch pedal; model of the 16-speed synchronized gearbox and the transfer case with asymmetric locking differential and ability to disconnect of the front axle, as well as drive axles with symmetric inter-wheel differentials and the symmetric locking inter-axle differential of the intermediate and the rear axle.

The front axle suspension model and the rear bogie center point suspension model consider the dynamics of the solid axle's operating process, quench oscillations of the carrier system and transfer power and moments thereto, which occur when the wheels interact with the road. The embedded model of Brixius/Dugoff tires [5, 6] ensures interaction of the wheel with the deformable bearing surface, while the dependency of the friction coefficient on the spin is a hyperbolical function.

The developed model allowed us to evaluate efficiency of the truck's power distribution with 6x6 and 6x4 drive circuit with regard to the motion resistance powers [7]:

$$K_{ef} = \frac{N_{6\times4}}{N_{6\times6}},$$

where, $N_{6\times4}$ – total vehicle's motion resistance power with circuit 6x4 ;

$N_{6\times6}$ – total vehicle's motion resistance power with circuit 6x6.


ISBN: 978-988-14048-3-1
ISSN: 2078-0958 (Print); ISSN: 2078-0966 (Online)

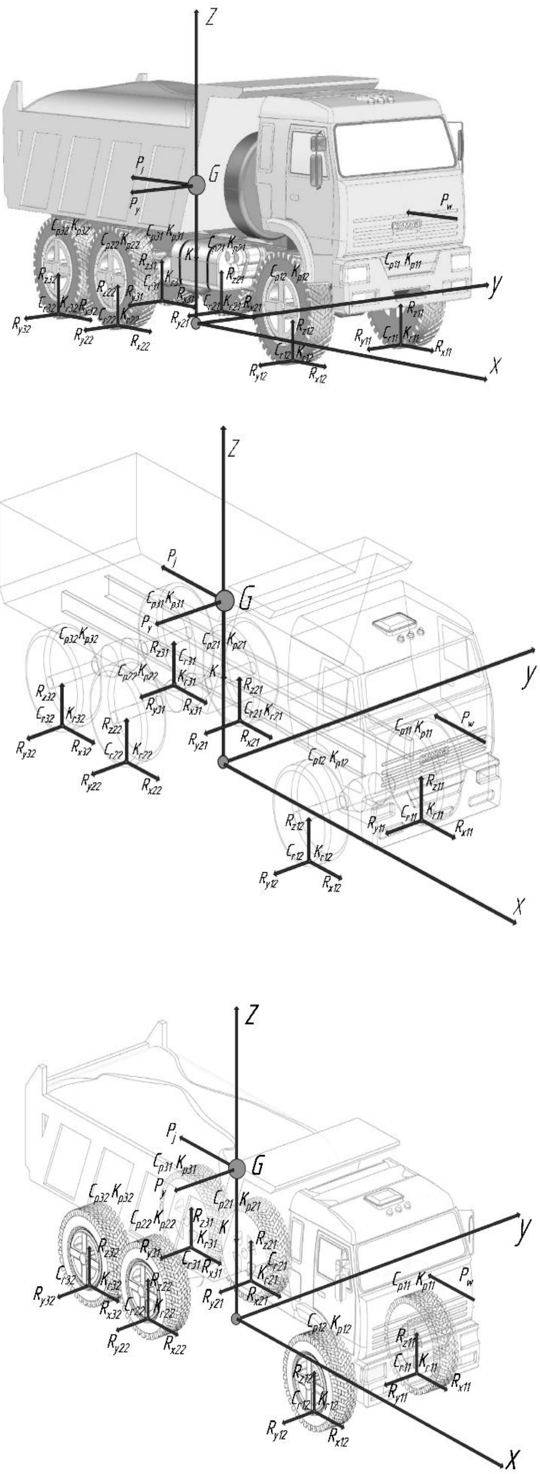


Fig. 3. Force pattern at linear motion
(G – Vehicle's center of gravity (CG); XYZ – axes of the global coordinate system; $P_j$ ,$P_y$ – axes of the local coordinate system beginning in the vehicle's CG; $R_{xij}$ – axial wheel force; $R_{yij}$ – lateral wheel force; $R_{Zij}$ – normal wheel force; $c_{rij}$ – wheel damping coefficient; $k_{ri}$ – wheel rigidity coefficient; $c_{pij}$ – suspension damping coefficient; $k_{pi}$ – suspension rigidity coefficient; $P_w$ – aerodynamic resistance force).

The main operating conditions of mine dump trucks [9,10] are descends and rises along roads of different slope levels and different support and adhesion properties, as well as roll-outs and turnovers to considerable angles. A full description of modeling scenarios is presented in Table II.

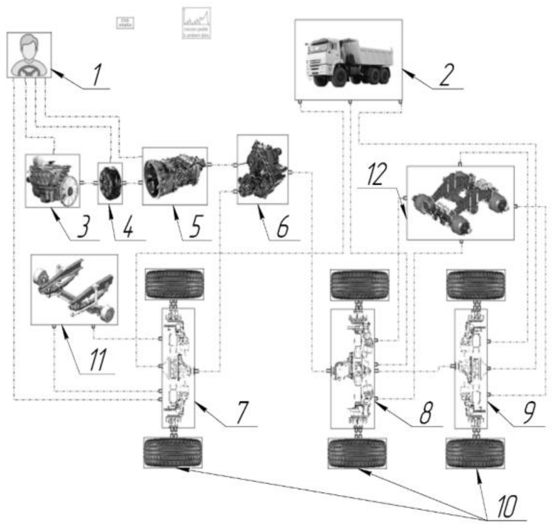


Fig. 4. Mathematical model made in LMS Imagine.Lab Amesim
(1 – Driver model; 2 – Vehicle chassis model; 3 – Engine model; 4 – Clutch model; 5 – Gearbox model; 6 – Transfer gear model; 7 – Front axle model; 8 – Intermediate axle model; 9 – Rear axle model; 10 – Model of the wheel and its interaction with the bearing surface; 11 – Front axle suspension model; 12 – Read bogie centerpoint suspension model).

TABLE II
MODELING SCENARIOS

| | Rise up the road slope (climbing aslope at the previously gathered speed) |
|---|---|
| Weight of the trailing load, T | 19,5; 9,725; 0 |
| Axle arrangement | 6x6; 6x4 |
| Climbing angle,% | up to 40 |
| Road conditions | Asphaltic concrete road; dirt road: dry rolled, after rain; dry sand |
| Position of the throttle blade | Full open |

## IV. RESULTS

As a result of modeling the movement along road sections with different types of road carpet and slopes we obtained dependencies of the power distribution efficiency coefficient on the wheel spinning and the weight of the trailing load (Figure 5).

In the course of an analysis of the obtained results we defined the threshold value of the wheel spinning of the rear bogie of the vehicle moving with the "disconnected" front axle, when it is expedient to change-over to the all-wheel drive power distribution pattern.

Based on the aforesaid, it is proposed to increase power and fuel efficiency of trucks by application of the automated all-wheel drive connection system.

The algorithm of the offered system is shown in figure 6. When the vehicle is moving, its motion speed and angle speeds of the traction wheels are continuously controlled. The control unit defines the traction wheel spinning by the following expression:

$$\delta = \frac{\omega \cdot r_d - \upsilon}{\upsilon},$$

where, $\omega$ – angle wheel speed;

ISBN: 978-988-14048-3-1
ISSN: 2078-0958 (Print); ISSN: 2078-0966 (Online)

$r_d$ – dynamic wheel radius (set by default);
$\upsilon$ – vehicle's longitudinal speed.

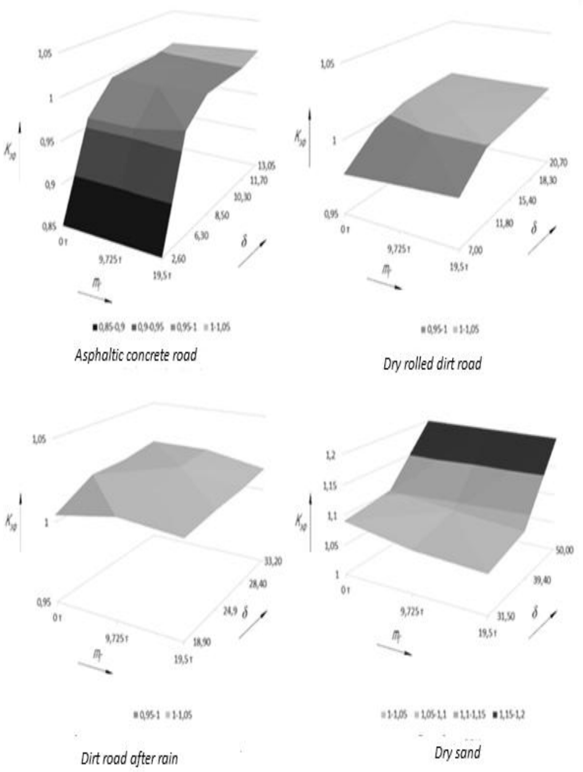


Fig. 5. Dependency of the power distribution efficiency coefficient on the wheel spinning and the weight of the trailing load

When the threshold wheel spinning value is reached, the control unit gives a signal to connect the all-wheel drive. The front axle is disconnected when the throttle angle begins to decrease (β).

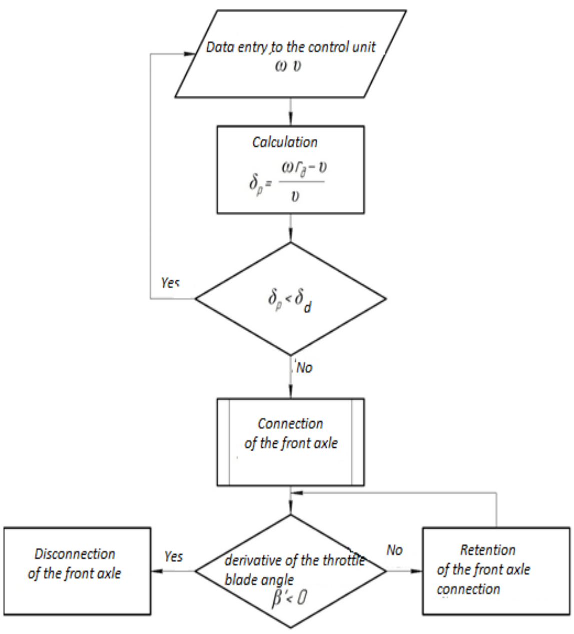


Fig. 6. Algorithm of the automated all-wheel drive connection system.

The efficiency of the offered pattern was determined by simulation modeling of the vehicle's movement along the standard route of the heavy building dump truck (Table III).

TABLE III
STANDARD ROUTE OF THE DUMP TRUCK

| # | Section | Road conditions | Length, km | Longitudinal slope, % |
|---|---|---|---|---|
| 1 | Mine face | Dry sand | 0,5 | 0 |
| 2 | Rise | Dry rolled dirt road | 0,5 | 17,5 |
| 3 | Rise | Dry rolled dirt road | 0,4 | 0 |
| 4 | Rise | Dry rolled dirt road | 0,3 | 8,8 |
| 5 | Rise | Dry rolled dirt road | 0,3 | 13 |
| 6 | Rise | Dry rolled dirt road | 0,1 | 25 |
| 7 | Surface | Asphaltic concrete road | 5 | 0 |

The standard route of the heavy building dump truck included removal of 20t of soil from the open pit mine to the waste pile. The route length comprised 7,1 km, including 30% impassability in the open pit mine and access roads thereto. The speed in the open pit mine was limited to 40 km/h, along the asphaltic concrete road – 60 km/h.

Based on the modeling results, we obtained values of the movement time along the route and the amount of fuel used during movement (Figure 7).

Application of the automated all-wheel drive connection system on KAMAZ-65222 dump truck increased the vehicle performance by 3,2% and 3,7% with regard to 6x6 and 6x4, accordingly.

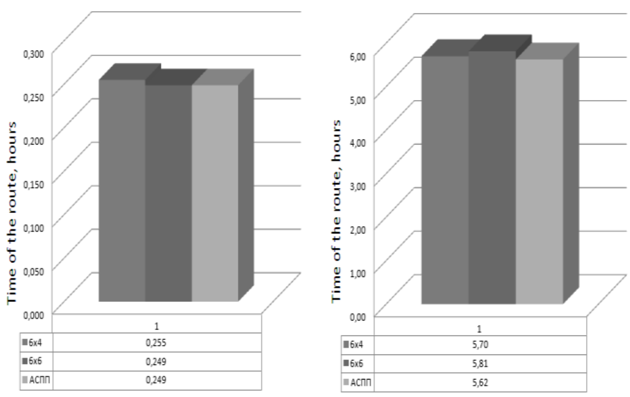


Fig. 7. Modeling results of the vehicle's movement along the standard route: Route passing time, h; АСПП – automated all-wheel drive connection system

## V. CONCLUSION

1. In this paper we developed a mathematical model of a vehicle, which includes a description of all the vehicle's basic systems with real parameters. The created mathematical model allowed implementing the vehicle's motion processes for all the required motion modes in view of the dynamics of the vehicle's major units and subsystems.

2. Based on the developed mathematical model we studied the vehicle's dynamics with application of the simulation approach and CAD LMS AMESim package of applied computer software. As a result of modeling the

movement along road sections with different types of road carpet and slopes we obtained the dependencies of the power distribution efficiency coefficient on the wheel spinning and the weight of the trailing load.

3. We defined the threshold value of the wheel spinning of the rear bogie of the vehicle moving with the “disconnected” front axle, when it is expedient to change-over to the all-wheel drive power distribution pattern. Based on the obtained results, we proposed to increase energy and fuel efficiency of trucks by application of the automated all-wheel drive connection system.

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ISBN: 978-988-14048-3-1
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