Stability and Control of Movement of the Truck with Automatic Differential Locking System

V. Anchukov, A. Alyukov, and S. Aliukov

Abstract—**The article presents an algorithm for the automatic differential locking system of a six-wheel drive heavy truck. For the fully differential transmission, a control law for the 2 inter-axle and 3 inter-wheels locks of the differentials has been developed. The process of control is realized by multi-criterion analysis of the current mode of operation of the vehicle. The usage of blocked mechanisms for power distribution is one of the most popular and effective ways to improve the off-road vehicle performance. However, the lock of differential may adversely affect the stability and control of vehicle because of the unobvious redistribution of reactions acting on wheels, which consequently leads to poor performance and safety properties. Problems of rational distribution of power in transmissions of all-wheel drive vehicles, as well as research in the field of improving directional stability and active safety systems are among the priorities in modern automotive industry. To study dynamics of a vehicle with wheel formula 6x6 a mathematical model of the vehicle was developed in an environment of LMS Amesim software package. The model includes the realization of the features of all major mechanical units of a vehicle: engine, transmission, suspension, drive wheels. Besides, the model takes into account the so called "external" dynamics of the vehicle and includes interaction of the wheels and pavement and implementation of possible changes in environmental conditions. With help of the mathematical model we have managed to estimate the trajectory and directional stability of all-wheel drive trucks with lockable differentials for different operating conditions. The results allowed us to develop the most effective, in terms of stability and control, algorithm for control of the power distribution system.**

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Index Terms— **Automatic differential locking, control of movement, all-wheel drive vehicles, heavy vehicle transmission**

I. Introduction

The method of implementation of rigid kinematic constraints in transmission is quite effective in terms of increasing of possibility of a wheeled vehicle [13,14]. However, the movement with locked differentials has several disadvantages, the most significant one includes deterioration of controllability and course stability of the vehicle [1]. Accordingly, to maintain and enhance these important performance properties, it is necessary to search optimal laws of power distribution. As a rule, the system of active power distribution includes the following main elements: means of collecting and evaluating information on the current operating mode, the control unit, actuators, power-sharing arrangements and other necessary mechanical elements.

Thus, the problem of control of power flow in the transmission of the wheeled vehicle is quite difficult and multidisciplinary. Significant investigation in this area belongs to V.V. Vantsevich, Josko Deur, Vladimir Ivanović, Matthew Hancock, Francis Assadian. For example, in [9] V.V. Vantsevich, Dennis Murphy and Gianantonio Bortolin made estimation of energy efficiency and fuel economy of a vehicle type of 6x6 by method of combination in transmissions of different differential mechanisms of power distribution. The authors suggested three drivetrain configurations and by drawing up mathematical models they carried out a comparative analysis with help of criteria of energy efficiency in the case of slippage of wheels.

The use of the locking differential mechanisms in wheeled vehicles to improve mobility and other dynamic characteristics is very important [1], it is devoted a large part of the following work.

Paper [3] presents a mathematical model of four types of active locking differentials: ALSD; clutch TVD superposition; stationary clutch TVD; TVD 4WD. The authors verified the developed mathematical model of differential ALSD experimentally. The results indicate a high degree of accuracy of the model. Description of experiment and summary are presented in [8].

In the paper [6] the authors developed a mathematical model of the semi active differential with electromagnetic actuator. The simulation results were presented and conclusions about the usefulness of this type of mechanism for power distribution were formed as well.

The next step of the development of the locking differentials was usage of intelligent control systems, working independently, without any direct participation of a driver. This issue is dedicated to the paper [11]. The authors suggested a model of autonomous controlled mechanism of power distribution. Also it was developed an algorithm to control of this mechanism. The algorithm allows controlling power flow transmitted to drive wheels, and, as a result, improving mobility of a wheeled vehicle. Besides, it was represented the results of simulation of motion of four-wheel vehicle with usage in transmission the above-mentioned mechanisms as two cross-axle and one inter-axle differentials. Estimation of their influence on the overall dynamic performance of the vehicle was done.

Similarly, in the papers [1,2] it was described the mathematical model of the lockable differential with a system of control. Effectiveness of this solution was evaluated.

Continuous and intensive growth of the number of vehicles in different countries and improvement of traction-speed qualities of the vehicles led to a sharp increase in the accident rate in road transport. One of the main directions of the improvement of the vehicles and the most important direction of the improvement of trucks is increasing safety of the traffic.

Active safety systems aimed at preventing emergency situations and help drivers in case of extreme modes of movement.

II. Description of the Mathematical Model

Mathematical model of process of movement of a vehicle should include all of the major subsystems of the vehicle together with the actual characteristics of the particular object of study [15-17]. Only in this case it is possible to obtain reliable results and their further use in the process of design of the vehicle. Estimated vehicle scheme 6x6 is shown in Figure 1.

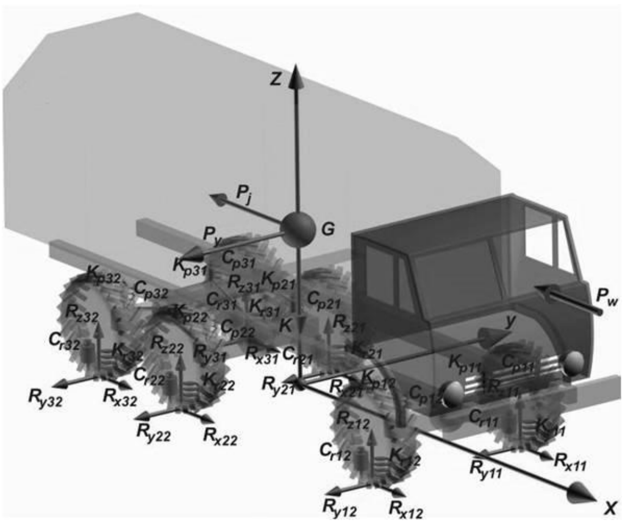


Fig.1. Estimated vehicle scheme 6x6

Assessment of dynamic load acting on transmissions is done by means of simulation in LMS Imagine Lab AMESim package. The developed mathematical model shows the process of movement of a vehicle on required modes, taking into account the specific features of dynamics of components and assemblies.

The body of the vehicle is a solid one to which forces and displacement determined by power unit and external environment are transmitted. The equations of motion of the body relative to a fixed coordinate system in the form of Lagrange type II are the following [1]:

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

$$T_1 = \frac{\left[\begin{array}{l} \sum_{i=1}^{2} R^i_{y12} a + \sum_{i=1}^{2} R^i_{y2} b_2 + \sum_{i=1}^{2} R^i_{y3} b_3 + 0,5 (R^{right}_{x1} + R^{left}_{x1}) B_1 + \\ +0,5 \left(\sum_{j=2}^{3} R^{right}_{xj} + \sum_{j=2}^{3} R^{left}_{xj} \right) B_2 \end{array} \right]}{J_z}$$

$$T_2 = \sum_{j=1}^{3} \sum_{i=1}^{2} R^i_{xj} / G$$

$$T_3 = \sum_{j=1}^{3} \sum_{i=1}^{2} R^i_{yj} / G$$

here

ω is velocity of rotation of body relatively vertical axis;

G_S is sprung weight of vehicle;

G is total mass of vehicle;

h_{COG} is height of center of gravity;

λ is angle of deflection of body deflection in horizontal plane;

V_X is speed of vehicle in the longitudinal direction;

V_Y is speed of vehicle in the transverse direction;

J_Z is moment of inertia of body relatively vertical axis;

R_X is longitudinal reaction of wheel;

R_Y is transverse reaction of wheel;

a is distance from the front axle to the center of gravity;

b_2 is distance from center of gravity to the central axis;

b_3 is distance from center of gravity to the rear axle;

B_1 is front track;

B_3 is rear track.

Engine-transmission unit includes a motor, working according to partial speed characteristics, transmission, yielding cardan shafts with consideration of gaps in joints, five differential mechanisms (two axle and three cross-axle ones) and main transmissions on each axis.

Suspension takes into consideration kinematic links and dynamic of work process, beam axle, transmitting forces and vibrations to the vehicle body. Implementation of the interaction of the wheels with supporting surface is done by means of the built-in model of tire named Rocard/Brossard, which is based on the following assumptions:

• deformation of the center line of tread is a result of lateral and longitudinal slip;

• the model does not account for the tire grip in the longitudinal and transverse directions;

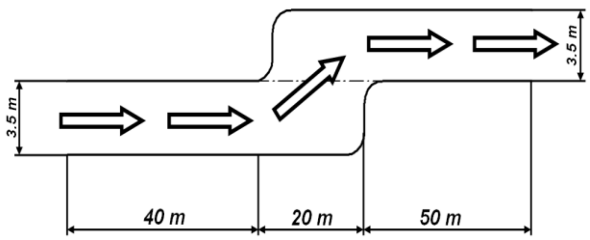
• vertical load is evenly distributed on the contact patch;
• in the case of slipping of the tread with respect to the road surface, deformation of the tread is uniform;
• magnitude of normal force and relative speed of the sliding do not affect friction factor.

Maintaining a predetermined driving mode is provided by the system "virtual driver", which is a set of measuring tools of condition of the car. The "virtual driver" system solves the following tasks: control of gas and clutch pedals, gear changes, braking and holding the desired trajectory.

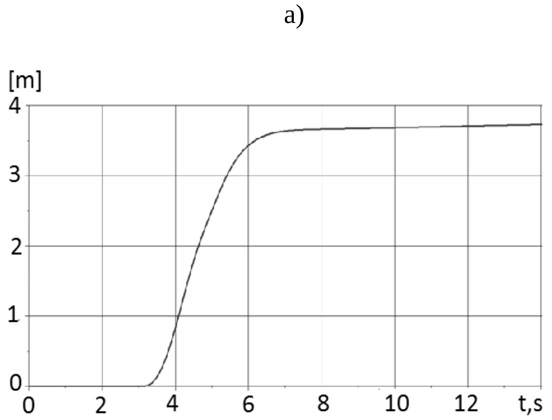
III. MANOEURING "PERMUTATION" AND "CLOSED PERMUTATION"

To perform settlement cases corresponding curvilinear motion of the mathematical model, there were selected two typical maneuvers, commonly used to evaluate the stability and control of vehicles, namely: "permutation 20 m." and "closed permutation". Maneuver "permutation 20 m." is described in ISO 31507-2012, maneuver "closed permutation" or «double lane-change» is described in ISO 3888-1: 1999 (E). Initial data for calculation of the two cases are shown in Table I, the results are illustrated in Figures 2-5.

As seen from these graphs, the trajectory of the vehicle is allowed to move in the given conditions. In the case of the maneuver "permutation 20 m." moving the vehicle center of gravity in the lateral direction was 3.6 m For maneuver "closed permutation" it was 4 m. Comparing these results with the options of the marking for performing each maneuver, it can be concluded that each maneuver is executed in the normal mode.



a)



b)

Fig.2. Scheme of marking of maneuvering area (a); trajectory of movement of the truck when maneuvering "permutation 20 m." (b)

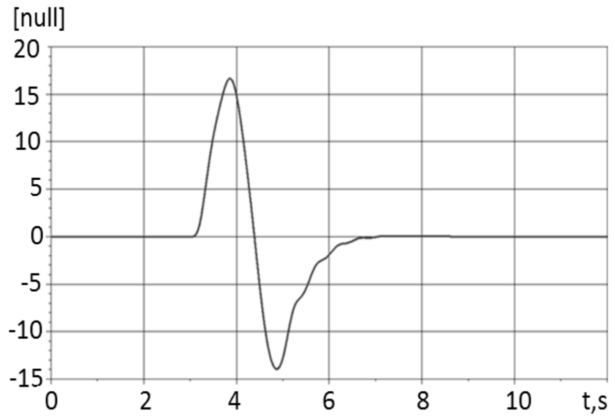
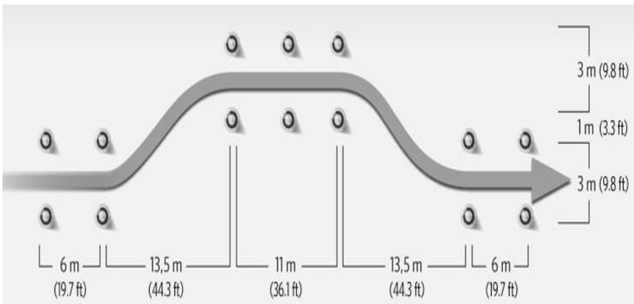
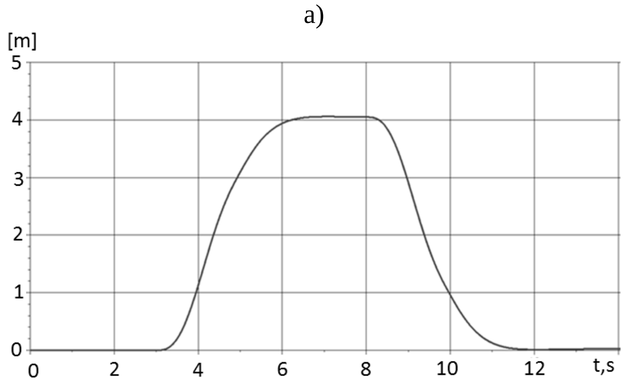


Fig.3. Change of yaw rate during movement



a)



b)

Fig.4. Scheme of marking of maneuvering area "closed permutation" (a); trajectory of movement of the truck when maneuvering "closed permutation" (b)

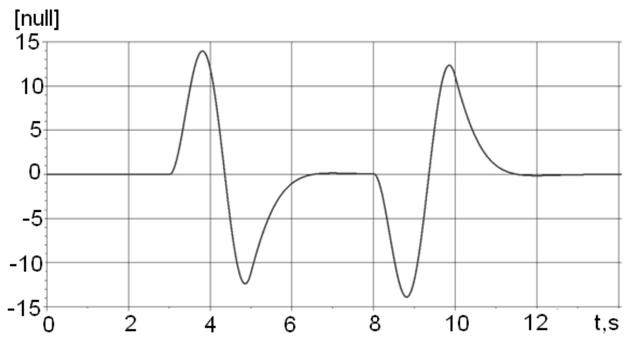


Fig.5. Changing the yaw speed while driving

TABLE I
INITIAL DATA FOR THE SETTLEMENT CASE "PERMUTATION 20 M." AND "CLOSED PERMUTATION"

Mode of movement	
The initial state of truck	Driving at a constant speed of 40 km/h
The coefficient of adhesion support surface	0,8
Transmission Mode of work of transmission	
Status of differentials	All are unlocked
№ of gear	7 (i = 2,2)
Gear shift	Absent
The power distribution between the front axle and the truck	1:2
Throttle position	Full fuel supply

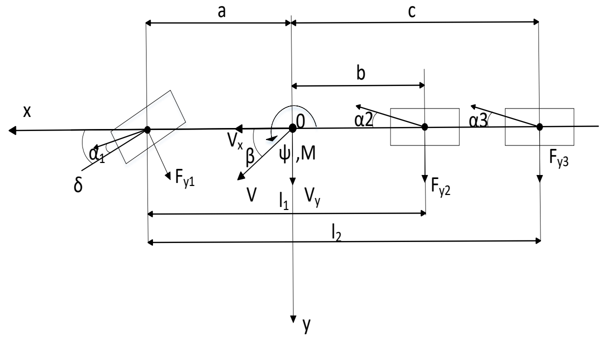


Fig.6. "Bicycle" model of three-axle vehicle

IV. Assessing Impact of Introduction of Rigid Kinematic Connection on Stability of the Vehicle

Locking of differentials while driving leads to non-obvious redistribution of reactions on vehicle wheels, which in turn can entail some negative consequences, such as loss of directional stability of motion. To maintain the directional and trajectory stability of the truck we make a function algorithm for system of stabilization of the directional stability based on differential braking. For calculations we use the so called "single-track" model of the truck, which is shown in Figure 6.

Let us describe all the physical quantities shown in Figure 6.

X,Y are coordinates of motion of center of mass of the vehicle, m;

ψ is yaw angle of the vehicle relatively to the center of mass, rad;

β is vehicle slip angle relatively to the center of mass, rad;

a,b,c are distances between axes of the wheels and the center of mass, m;

l1 is distance between the first and the second wheel axis, m;

l2 is distance between the first and the third wheel axis, m;

Fy1, Fy2, Fy3 are transverse forces arising during turning, H;

δ is angle of turning of the front wheels, rad;

α1, α2, α3 are slip angles of wheels, rad;

Vx is longitudinal velocity of the vehicle relatively to the center of mass, m/s;

Vy is transverse velocity of the vehicle relatively to the center of mass, m/s;

M is torque of the center of mass relatively to vertical axis, H·m.

The trajectory is described by the equations:

$$\dot{X} = V_x \cos\left(\psi + \beta\right),$$
$$\dot{Y} = V_x \sin\left(\psi + \beta\right).$$

As expected, the speed remains constant (*Vx* = const), then the trajectory depends on the angles ψ and β. According to Newton's second law for rotational and translational motion of the center of mass we have:

$$J\ddot{\psi} = M + aF_{y1} - bF_{y2} - cF_{y3},$$
$$mV_x\left(\frac{\partial\beta}{\partial t} + \dot{\psi}\right) = F_{y1} + F_{y2} + F_{y3} + F_{bank},$$

here *J* is moment of inertia of the center of mass relatively vertical axis, H·m2;
m is mass of the truck with respect to the center of mass, kg.

The differential braking of wheels is one of the main methods for many control systems of vehicles, including directional stability. The essence of this method will be discussed below, based on Figure 7.

The essence of this method lies in the fact that braking left (right) wheels on the rear axles of the truck, the forces on the right (left) wheels become larger opposite ones. Thus, it is created a torque (M_f) acting on the truck around its center of gravity. Torque applied to the center of mass of the truck, is produced by differential braking of the rear wheels. Link between this torque and the braking force is defined as:

$$M_f = \frac{\left(\left(F_{2,R} - F_{2,L}\right) + \left(F_{3,R} - F_{3,L}\right)\right)(b+c)}{4}.$$

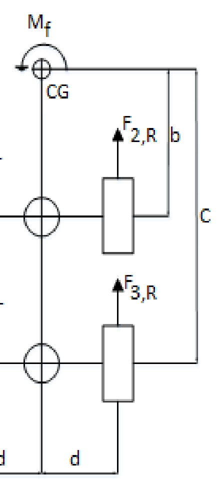


Fig.7. Geometrical parameters of rear wheels of three-axle truck

Figure 8 illustrates a functional scheme of the system of stabilization of directional stability.

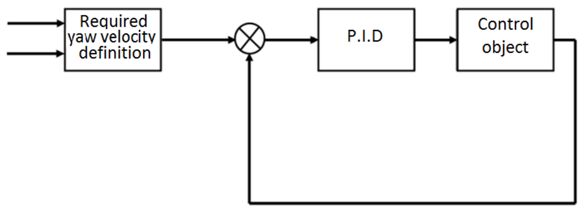


Fig. 8. Structural control scheme using the directional stability of the system with feedback on the angular velocity

Here $\varepsilon = \dot{\psi}_p - \dot{\psi}$ is error between actual and desired the actual and the desired angular speed;

$u = k_p\varepsilon + \frac{k_i}{s}\varepsilon$ is control action regulating the supply of pressure on the brake cylinder;

k_p, k_i are coefficients of proportional and integral controllers.

To assess workability, we ask the system of stabilization of directional stability to perform the maneuver "permutation 20 m." with different coefficients of adhesion under the left and right wheels. Description of the design case is shown in Table II.

Figures 9-10 illustrate graphs of changing the transverse coordinate position of the center of gravity in space of driving for two cases: the movement without stabilization system of directional stability and movement when this system is available.

These results clearly demonstrate the effectiveness of the system. Movement of the vehicle under the given conditions without sideslip stabilization system leads to a serious deviation from desired trajectory and therefore, to an unsatisfactory result maneuver.

As shown in Figure 10, the use of sideslip stabilization system functioning on the principle of differential inhibition, promotes satisfactory maneuvers "permutation 20 m." on a given mode of movement.

TABLE II
INITIAL CONDITIONS OF THE SETTLEMENT CASE "PERMUTATION 20 M." WITH DIFFERENT COEFFICIENTS OF ADHESION UNDER THE VEHICLE SIDES

Mode of movement	
The initial state of truck	Driving at a constant speed of 40 km/h
The coefficient of adhesion support surface	0,8; 0,1
Transmission Mode of work of transmission	
Status of differentials	All are unlocked
№ of gear	7 (i = 2,2)
Gear shift	Absent
The power distribution between the front axle and the truck	1:2
Throttle position	Full fuel supply

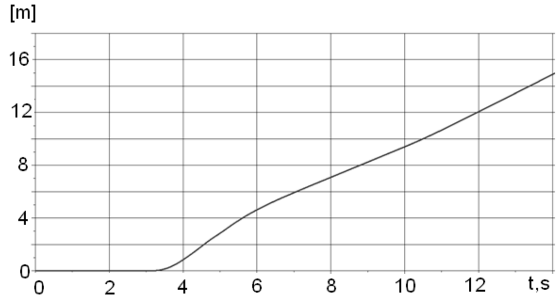


Fig.9. Trajectory of movement of vehicle without sideslip stabilization system

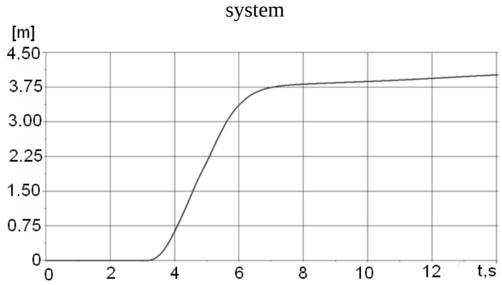


Fig.10. Trajectory of movement of the vehicle with the sideslip stabilization system

Effect of blocking differential links on the dynamics of the vehicle we estimate with help of the following case: the vehicle is moving at a constant pace on support base with a constant coefficient of adhesion, then one side comes over on the site with lower value of the coefficient of adhesion. The results are illustrated in Figures 11-13.

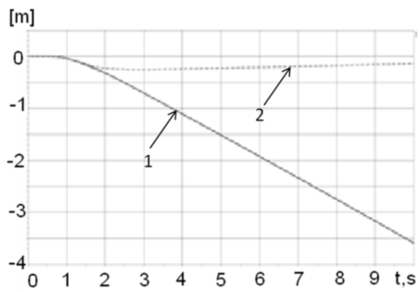


Fig.11. Trajectory of movement of the vehicle without sideslip stabilization system (1) and with the system (2)

Graph of trajectory of the vehicle (Figure 11) demonstrates the process of movement of the vehicle from desired trajectory, caused by the introduction of rigid kinematic connection. By the nature of changing graphs of rotation speed and torques of output shafts of the transfer case it is possible to observe the process of working the system of stability. Designed algorithm can be considered as effective because its usage led to withdrawal from the desired trajectory of less than 0.5 meters, which is valid for driving on public roads with a width of one lane of 2.75 m.

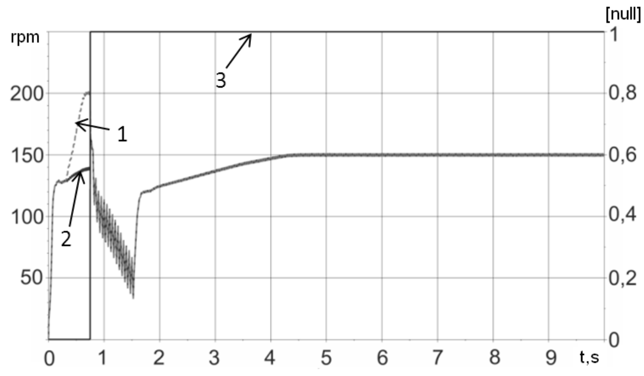


Fig.12. Changing speed of rotation of output shafts of transfer case (1, 2). Signal of switching differential lock - 3.

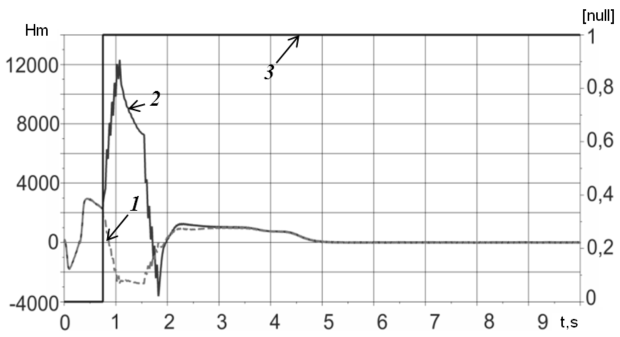


Fig.13. Changing speed of rotation of output shafts of transfer case (1, 2). Signal of switching differential lock - 3.

Based on our investigations we have developed the methodology of proper power distribution among wheels has been developed. The developed methodology is based on:

- the theory of operational properties of wheeled vehicles;
- the theory of rolling elastic wheels;
- optimal control theory;
- principles and methods of a systematic approach.

It has been done study of different methods of power distribution among the drive wheels of an all-wheel-drive truck, namely:

- method of partial solution;
- method of introducing a rigid kinematic connection;
- method of periodical action;
- and method of limit of excessive action.

We have proved the following.

On the mode of acceleration up to 30 km/h, a rigid kinematic connection among driving axles and wheels of unmanaged axels should be switched on. After reaching velocity of 30 km/h it should be provided disconnection of the rigid link and then a differential connection among drive wheels. Upon reaching of steady velocity and slippage of the drive wheels less than 5%, front axle should be switched off. With increasing of the slippage of the drive wheels more than 5%, full differential drive should be switched on.

V. Description of the Control Algorithm for Blocking Differentials

To increase the efficiency of the truck in mixed road conditions, the following algorithm for the operation of automatic differential locks is proposed:

1. The car is started with the transmission completely blocked. This will maximize the use of the coupling properties of the road surface and exclude the possibility of slipping the wheels during the start.
2. When the speed reaches 20 km/h, interlocking differentials are disengaged, and when the speed reaches 30 km/h, inter-axle locks are disabled. This condition is necessary to maintain the stability and controllability of the car at these speeds, because It is known that when maneuvering at high speeds, blocked differential connections can have a negative effect on these performance characteristics.
3. Regardless of the speed, the differential locks are disabled when the steering wheel is turned more than 30 degrees. The turning radius with locked differentials is less than with the unlocked ones, so this element of the control algorithm is applied.
4. When driving in settlements (GPS position analysis is used) for normal maneuvering and, consequently, ensuring the safety of all road users, all locking differentials are disabled.
5. With prolonged slipping of the wheels, accompanied by a three-time operation of the PBC system, regardless of the location of the vehicle, the corresponding differential locks are activated: when the axles are slipped, there are interaxial ones; when slipping wheels - first interaxle, and then inter-wheel.

VI. Simulation Results

To assess the functioning of the algorithm proposed above, a number of types of indicative calculation cases typical for the movement of a truck were adopted:

1. Acceleration to a speed of over 30 km/h with subsequent braking.

The most common modes of a truck’s movement are acceleration and braking [18,19]. The acceleration process is carried out with sequential gear shifting, braking after 25 s. movement and further continuation of the movement after 2 seconds braking. The corresponding results for vehicle speed, control signals for differentials locking and braking enabling, are shown in Figures 14-16.

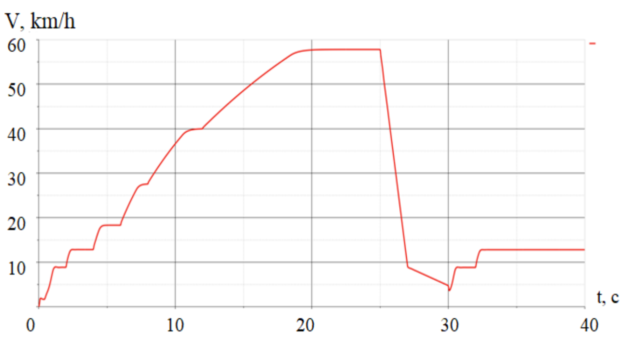


Fig.14. Vehicle speed changing during acceleration and braking

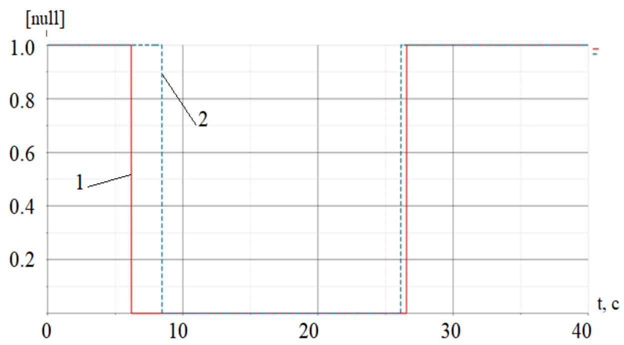


Fig.15. Enabling/disabling differentials locking
(1 – inter-wheel; 2 – inter-axle)

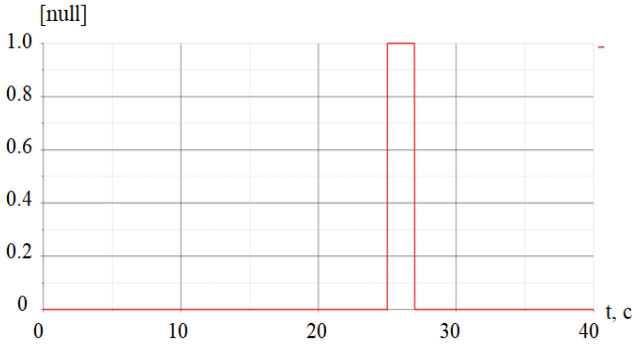


Fig.16. Control signal for braking

According to the results obtained, in particular the data in Figure 15, it can be concluded that during the acceleration, when the speed reaches 20 km/h, the blocking of the inter-wheel differentials is disengaged, and when the speed reaches 30 km/h, the inter-axle locks are disengaged. Further, during braking and lowering the speed below the set values, inter-axle and inter-wheel lock of the differentials are successively included.

2. Curved motion with steering wheel rotation.

The movement of the truck is carried out with a gradual from the rest position to a speed of less than 20 km/h. When time reaches 5 sec, the steering control input is applied to the steering wheel for further turning the car. The results are shown in Figures 17-19.

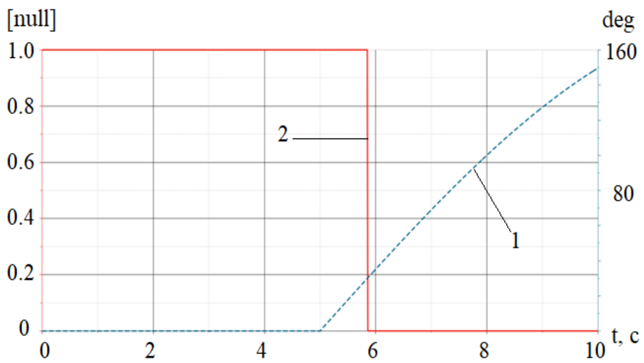


Fig.17. Steering wheel rotation (1) and inter-axle/inter-wheel differentials locks state (2)

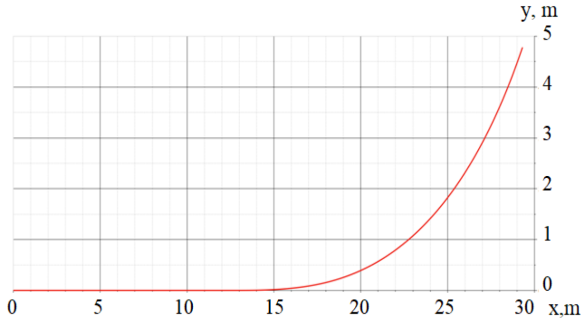


Fig.18. Vehicle trajectory in X-Y plane

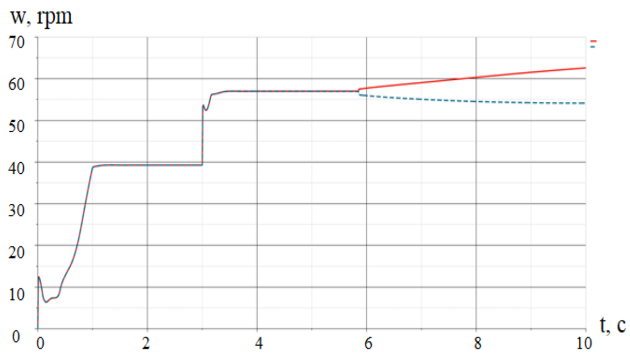


Fig.19. Wheels rotational speed

From Figures 18 and 19 it can be seen, that car turn is carried out with different speeds of rotation of wheels of the forward (operated) axis that testifies to the unlocked differential.

3. Uphill movement in city

In this case, the traffic in settlement is simulated. In accordance with the adopted algorithm, all differentials are initially unlocked, the car accelerates to a speed of more than 30 km/h, after 60 m of the path, the lift starts at an angle of 10 degrees with a semicircular entry at the beginning. The results are shown in Figures 20-13.

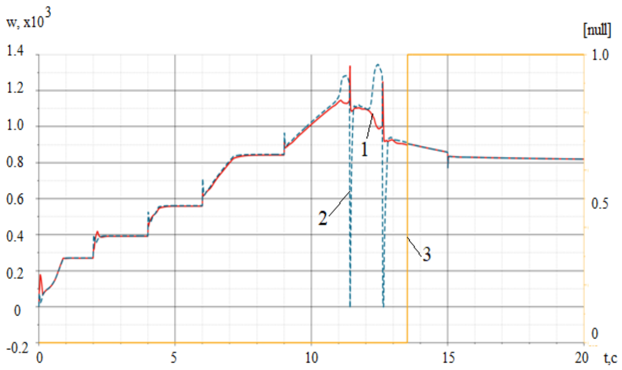


Fig. 20. Transfer case output shafts rotational speed (1 – front axle shaft; 2 – rear cart shaft) and control signal for inter-axle differentials locking (3)

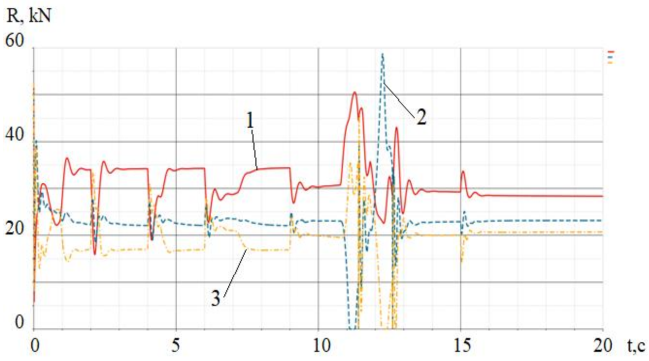


Fig.21. Wheels vertical reactions
(1 – front axle; 2 – middle axle; 3 – rear axle)

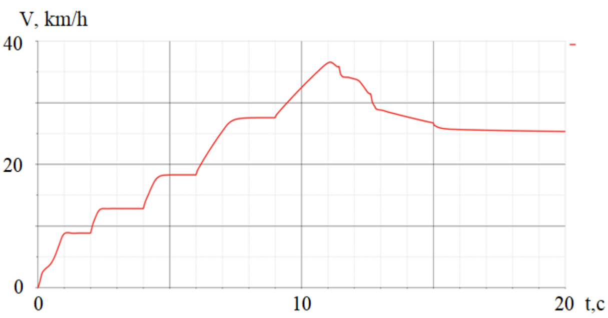


Fig.22. Vehicle speed when moving uphill

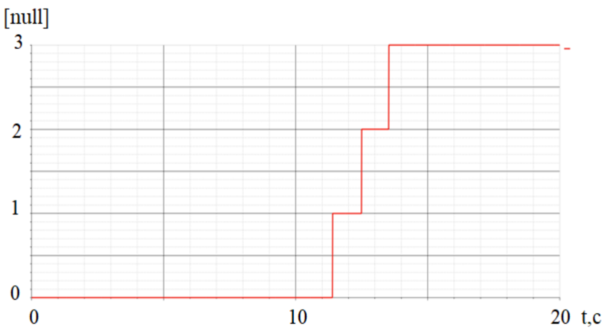


Fig.23. ASR enabling counter

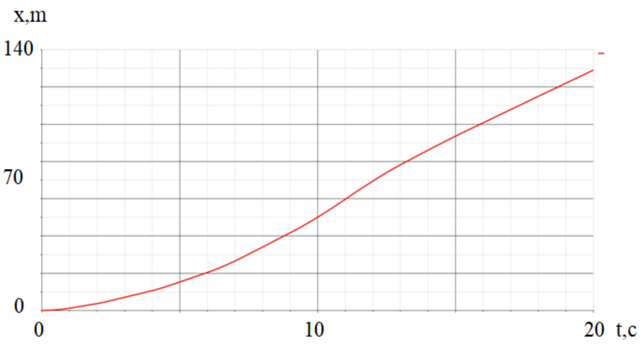


Fig.24. Lateral coordinate changing when moving uphill

According to the change in the speed of rotation of the output shafts of the transfer box (Figure 20), it can be seen that the wheels of the rear cart are slipping, this is a consequence of the consecutive short-term hanging of the wheels of the middle and rear axles (Figure 21) when the car enters the haulage area. When the axles are slip, the ASR system is triggered, however, its work is not enough to continue the motion, therefore, after the 3rd actuation (Figure 23), a control signal is sent to enable the inter-axle locks of the differentials (Figure 20). After that, the truck continues to move upward (Figure 24) at a constant speed (Figure 22).

V. Conclusion

1. The developed system to control power distribution of torque allows us to improve significantly vehicle's passability. At the same time on some driving modes there is a loss of directional stability of the vehicle when rigid kinematic connection is introduced during the driving. To solve this problem, a system of stabilization of directional stability was developed, which operates on the principle of differential braking wheels of the vehicle. Thus, it can be concluded that providing the necessary operational parameters of the vehicle is achieved by co-operation of the power distribution and the control system for stabilization of directional stability.

2. The algorithm for controlling automatic locking of differentials of a truck with a wheel formula 6x6 is developed. Means of simulation mathematical modeling confirmed the overall efficiency and effectiveness of the proposed algorithm for typical modes of vehicle operation, including in conjunction with regular active safety systems (ABS and ASR).

3. Simulation results proved that the proposed strategy for controlling power distribution mechanisms does not adversely affect the stability and controllability of the vehicle when rigid kinematic constraints are introduced in the course of the motion. Based on obtained results, the struckture of the software and hardware complex for automated control of truck differential locks has been formed.

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