Design of active stability control system of agricultural off-road vehicles

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Abstract

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Part of active stability control system design of an agricultural technological vehicle designated for working in mountain and foothill areas is described. The principle of active control of angular velocities of the centre of gravity has been used. During the manoeuvre, active tipping axes are identified by orientation of the weight vector. From experimental tests of a machine MT8-222 based on the Standard STN 47 0170, the real records of angular velocities were obtained. Tests were executed on the slope with an average slope of 32 degrees. From computation critical angular velocities were gained, by which the machine could get into the position of labile stability during the manoeuvre. The regulator was simulated in Matlab®, which controlled the actual value of angular velocity compared with the critical one. In case the boundary zone of critical angular velocity was reached, the regulator sends a signal to the fuel control system and then vehicle speed decreased. During experimental tests, the vehicle did not turn over. Therefore, the angular velocity was simulated by a generated function so that the functionality of the designed regulator was verified.

Keywords: vehicle dynamics; mathematical modelling; simulation

The influence of dynamic effects of the ride of an agricultural mechanism on a sloping terrain is the main cause of its rollover. Following the rollover of the machine, leakage of technical fluids, damage of the machine as well as injuries of the driver, frequently fatal, can occur. This is documented by records of the National Labour Inspectorate of the Slovak Republic, which registers 12,874 occupational injuries in agriculture from 2000, from which 66 cases were caused by agricultural mechanisms. Up to 13 accidents were caused by vehicle overturns. Safety at work and operational safety of agricultural mechanisms are monitored by the European Agency for Safety and Health at Work. The agency states that in the United Kingdom transport with agricultural vehicles was the main reason of fatal injuries of workers in agriculture during the years 1999 and 2009. The cases as being struck by a moving vehicle with a 25% share or trapped by something collapsing and overturning with a 7% share of all fatalities are registered (European Agency for Safety and Health at Work 2011). Because of this, the main priority of producers of agricultural machines nowadays is an implementation of intelligent stability control systems, mainly of those which work on a sloping terrain of more than 15 degrees.

A remarkable contribution in the research of stability of agricultural machines operation was achieved by Grečenko (1983, 1986) who inter alia presents that dynamic stability is part of the slope stability of agricultural vehicles. A significant contribution in the research of static and dynamic stability of agricultural off-road vehicles was presented by Šesták et al. (1987), who dealt with the static stability and slope stability of machines working on the sloping terrain. Fundamental results of the dynamic stability research were published in the works of Šesták et al. (1989, 1993). The methodology of determining the slope stability of agricultural vehicles in Slovakia was presented by Šesták et al. (2000). Also, Spencer and Gilfillan (1976), Chisholm (1979a,b), and Swanghart (1987) markedly contributed with their works to the problem of vehicle dynamic stability and to the research of the dynamics of vehicle rollover on the slope. With the arrival of modern information technology begins an era of more intensive research and modification of existing methods and the specification of some terramechanics theories. Pacejka (2005) and Genta (2006) also contributed with notable works. The phase of increasing the operational safety of agricultural vehicles begins with implementation of intelligent systems into vehicle control systems, with connection to gyroscopic sensors and accelerometers. The utilization and suitability of use of accelerometers in vehicle dynamics is described by MacDonald (1990). Contemporary development is oriented mainly towards modelling and dynamic simulations in various CAD/CAM programs, which allow motion simulations of vehicles with more degrees of freedom. Input forcing functions appear as an appropriate supplement. For simulation of dynamics of the vehicle Fiat Arborio et al. (2000) use the stability analysis in Matlab and Adams Car. Strassberger and Guldner (2004) describe the "Active Stabilizer Bar System" developed by BMW and called dynamic drive. Dynamic drive significantly reduces roll angle during cornering. Zolotas et al. (2006) discussed various issues related to creating models for controllers design, which can be applied to complex dynamic systems. The prediction of vehicle dynamics on soft soil in the Adams system is published also by Fassbender et al. (2007). A complex system of active stability control ADAS (Advanced Driver Assistance Systems) is described by McNair (2007), where the system directly communicates via satellite connection. Pro/Engineer is used for vehicle dynamics simulation by Bradley et al. (2009). Mikleš et al. (2011) dealt with constructional parameters of forest machines, which directly influence the centre of gravity of the machine, and with the determination of static and dynamic stability of forest machines. At modern machines there are used modern active technologies such as Electronic stability control (ESC). ESC is safety feature that detects and prevents (or recovers from) skids. It can help keep the driver from losing control of the car in a panic swerve or when driving on slippery roads (Gold 2013). Mitsubishi Motors Corporation (2013) uses Active Stability Control (ASC) which helps drivers maintain in adverse weather conditions and during emergency manoeuvres. In BMW vehicles is Dynamic Stability Control (DSC) used. DSC adds safety to facilitating vehicle control even in adverse driving conditions or on tough surfaces. It ensures the highest possible levels of stability when driving and it maximizes traction of all wheels when setting off or accelerating (BMW 2013).

MATERIAL AND METHODS

Spatial identification of vehicle. The mathematical model of the vehicle was created with six degrees of freedom, whereby the mass of the whole machine is represented by a single point, namely

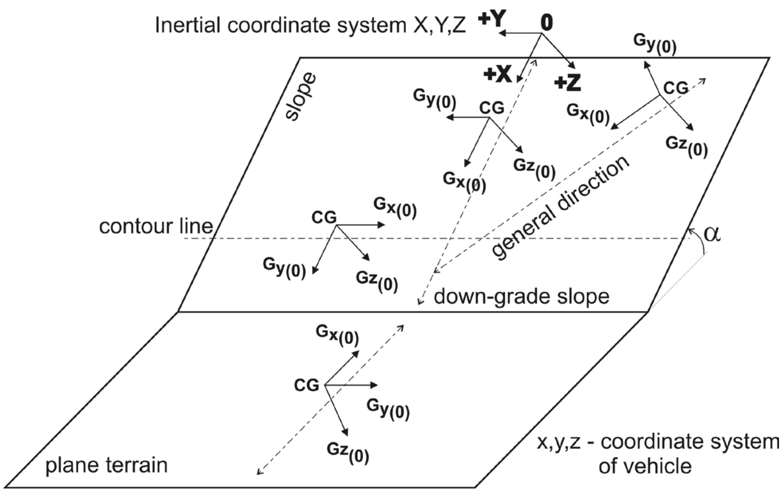

Fig. 1. Spatial identification of the vehicle

α – angle of slope; CG – centre of gravity; Gx,Gy,Gz – components of weight vector the centre of gravity. We followed the Standard SAE J670_200801 (2008) in terms of the terminology of vehicle dynamics. In Fig. 1, there is depicted a model of the slope with a defined inertial coordinate system (X, Y, Z). The vehicle and its moving on the slope is represented by the motion of the centre of gravity, in which the coordinate system of the vehicle (x, y, z) is oriented. Particular orientations of the weight vector as well as its components are defined according to Fig. 1.

In terms of spatial identification of the vehicle and transformations in the inertial system and in the coordinate system of the vehicle, there were chosen Euler's parameters. A symbolic matrix notation is:

$$[\Lambda'] = \frac{1}{2}[\Omega] \times [\Lambda] \tag{1}$$

where :
[Λ'] – matrix of parameters derivation
[Ω] – matrix of angular velocities
[Λ] – matrix of parameters

By solution of this system of differential equations, we obtain direction cosines of the transformation matrix between vectors determined in the coordinate system fixed to the vehicle and inertial space. These are:

$$\begin{aligned}
&a_{11} = \lambda_0^2 + \lambda_1^2 - \lambda_2^2 - \lambda_3^2, a_{12} = 2(\lambda_1 \times \lambda_2 + \lambda_0 \times \lambda_3)\\
&a_{13} = 2(\lambda_1 \times \lambda_3 - \lambda_0 \times \lambda_2), a_{21} = 2(\lambda_1 \times \lambda_2 - \lambda_0 \times \lambda_3)\\
&a_{22} = \lambda_0^2 + \lambda_2^2 - \lambda_3^2 - \lambda_1^2, a_{23} = 2(\lambda_2 \times \lambda_3 + \lambda_0 \times \lambda_1)\\
&a_{31} = 2(\lambda_3 \times \lambda_1 + \lambda_0 \times \lambda_2), a_{32} = 2(\lambda_2 \times \lambda_3 - \lambda_0 \times \lambda_1)\\
&a_{33} = \lambda_0^2 + \lambda_3^2 - \lambda_1^2 - \lambda_2^2
\end{aligned} \tag{2}$$

From parameters (Eq. 2), we obtain the transformation matrix as follows:

$$[M_T] = \prod_{i=1,2,3,\dots}^{n} \begin{bmatrix} a_{11} & a_{12} & a_{13} \\ a_{21} & a_{22} & a_{23} \\ a_{31} & a_{32} & a_{33} \end{bmatrix}^{-1} \tag{3}$$

Components of the angular velocity vector of the centre of gravity in the inertial system are determined as follows:

$$[\Omega_{Ts}] = [M_T] \times [\Omega_T] \tag{4}$$

where:
$[\Omega_{Ts}]$ – matrix of angular velocities of the centre of with respect to the inertial coordinate system,
$[\Omega_T]$ – matrix of angular velocities of the centre of gravity with respect to the coordinate system of the vehicle

The dynamic analysis of automotive vehicle stability involves also determining the components of the vehicle weight in the initial position and also during the ride.

Components of the vehicle weight in the initial position (vehicle does not move) are determined as follows:

$$[G_0] = [T_x] \times [T_y] \times [T_z] \tag{5}$$

where:
$[G_0]$ – matrix of vehicle weights in the initial position on the slope,
$[T_x]$, $[T_y]$, $[T_z]$ – matrices of rotations with respect to the x, y, z axes
$[G]$ – matrix expressing the total weight of the machine at rest on a flat ground

$$[T_x] = \begin{bmatrix} 1 & 0 & 0 \\ 0 & \cos\alpha & \sin\alpha \\ 0 & -\sin\alpha & \cos\alpha \end{bmatrix}$$
$$\begin{bmatrix} G_{X(0)} \\ G_{Y(0)} \\ G_{Z(0)} \end{bmatrix} = \begin{bmatrix} \cos\beta & 0 & -\sin\beta \\ 0 & 1 & 0 \\ \sin\beta & 0 & \cos\beta \end{bmatrix} \times \begin{bmatrix} \cos\gamma & \sin\gamma & 0 \\ -\sin\gamma & \cos\gamma & 0 \\ 0 & 0 & 1 \end{bmatrix} \times \begin{bmatrix} 0 \\ 0 \\ G \end{bmatrix} \tag{6}$$

The following transformation is used for computing the weight components during the ride:

$$[G_i] = [M_{T(i)}] \times [G_0] \tag{7}$$

where: $[G_i] = \begin{bmatrix} G_{x(i)} \\ G_{y(i)} \\ G_{z(i)} \end{bmatrix}$ – matrix of the components of vehicle weights during the ride in the i-th instant of time

Stability criterion. In the process of dynamic stability determination, it is important to identify tipping axes, which are dependent on the construction of the given machine (rigid vehicle, vehicle divided by axial or vertical pin).

The vehicle is in an unstable position when the value of kinetic energy determined with respect to the chosen tipping axis is greater than the value of potencial energy spent on the centre of gravity displacement into a limiting static position. Stability vanishes at the moment when the vehicle begins to turn over about some of the tipping axes. The loss of control occurs when the vehicle does not conform to driver's interventions. Vehicle control is fully influenced by the properties of the base and radial force or the vertical load of the wheels. In a real description of the dynamic stability criterion, we followed the work of Šesták et al. (1993), where determination of the dynamic stability criterion, obtained from vehicle operation, involves changes in the direction of the weight vector with respect

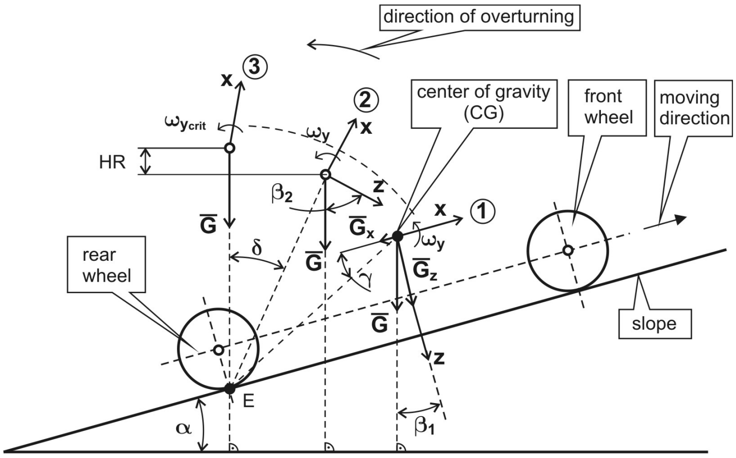

Fig. 2. Overturning dynamics
ω_y – angular velocity; $\omega_{y\text{-}crit}$ –critical angular velocity; HR – vertical difference; α – angle of slope; β, γ, δ – instantaneous angles of rotations of the automotive vehicle; E-point – rear tipping axis in the plane; $\overline{G}$ – weight vector; $\overline{G}_z, \overline{G}_y$ – components of weight vector; *x, z* – coordinates of the vehicle

to the determining tipping axis and inertial effects. Weight components of the vehicle in the initial position on the slope can be defined according to Fig. 2. The real phase of overturn begins with the angular rotation of the front (rear) part of the vehicle, with respect to the rear (front) part, about the internal tipping axis. Tipping axes are represented by the joining lines of points, which are the contact points of the tyre with the base. In Fig. 2, the rear tipping axis in the plane is represented by the point *E*. In the mathematical description of stability criterions, there are considered external tipping axes because the loss of contact of the only wheel with the base is not the condition of total stability loss of the machine. The real description of stability criterions, obtained from the operation of the automotive vehicle, then involves changes in the direction of the weight vector and inertial effects. According to instantaneous angles of rotations of the automotive vehicle, components of the weight vector are obtained by transforming the weight vector components from the initial start position, as introduced in Eq. (7). Relations describing the stability with respect to the rear tipping axis are for the known weight vector *G* determined according to Fig. 2. The value of the resultant of the vehicle weight in the plane *TXZ* of the coordinate system fixed to the vehicle is:

$$G = \sqrt{G_X^2 + G_Z^2} \tag{8}$$

The instantaneous direction of the resultant, oriented from the *Z* axis of the coordinate system fixed to the vehicle, will be:

$$SH = \frac{\frac{\pi}{2} - \arctan\frac{CG_z}{X_r}}{\beta_1} \tag{9}$$

Then, we can define a conventionally used stability coefficient with the relation:

$$\beta_1 = \arctan\frac{G_X}{G_Z} \tag{10}$$

where:
CG_z, X_r – dimensions found on the machine

If $SH = 1$, the machine is at the limit of static stability; if $SH < 1$, the total loss of stability occurs. In this procedure, the coefficient *SH* then informs the researcher about the backup of real stability in regard to the limit static value in each instant of time of realisation. Stability criterions are further derived in terms of Fig. 2. The radius of rotation of the centre of gravity towards the point *E* is determined as follows:

$$\overline{ECG} = \sqrt{CG_z^2 + X_r^2} \tag{11}$$

By the influence of dynamic effects which turn the machine about the tipping axis represented by the point *E*, the position of the centre of gravity is changed from position 1 to position 2 by vertical difference *HR*, which can be expressed as:

$$HR = \overline{ECG}\left(\cos\delta - \cos\left(\frac{\pi}{2} - \beta_1 - \gamma\right)\right) \tag{12}$$

Seeing that the kinetic energy of the vehicle which performs general plane motion is given by the sum of kinetic energy concentrated in the centre of gravity and kinetic energy of rotational motion about the centre of gravity, and the rotational motion is related to the space which is moved with the centre of gravity, we can determine the kinetic energy in point 1. The required angular velocity is determined from the experiment, and it must hold that $\omega_y > 0$. Kinetic energy is then expressed by the relation:

$$KH = \frac{1}{2}J_y \times \omega_y + \frac{1}{2}\omega_y^2 \times \overline{ECG}^2 \times m \tag{13}$$

If we set $g = 10$ m/s^2, then after modification we obtain:

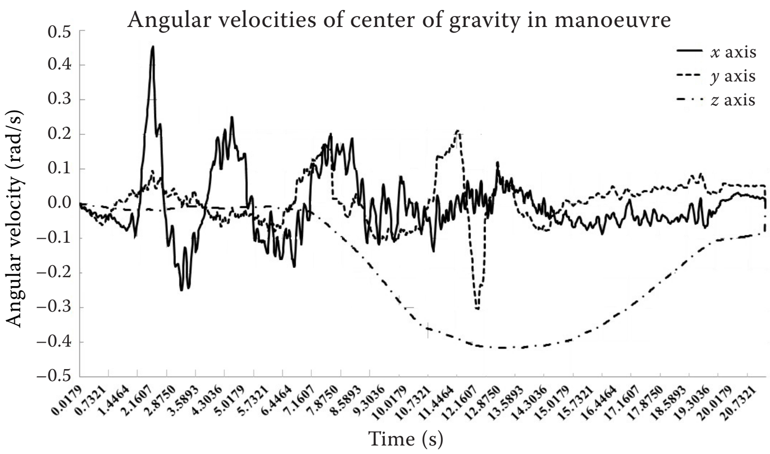

Fig. 3. Angular velocities of the centre of gravity during the manoeuvre

$$KH = \frac{1}{2}\omega_y^2 \times \left(J_y + 0.1G \times \overline{ECG}^2\right) \tag{14}$$

It is obvious that this kinetic energy is consumed during the process of the centre of gravity turning from position 1 to position 2 and this leads to $KH = HR \times G$.

In case the vehicle deflects from the slope, dynamic stability criterion DH can be set as follows. According to Fig. 2, we determine angle δ:

$$\delta_R = \arccos\left(\frac{KH}{G \times \overline{ECG}} + \cos\left(\frac{\pi}{2} - \beta_1 - \gamma_R\right)\right) \tag{15}$$

Next, we determine angle β_2:

$$\beta_2 = \frac{\pi}{2} - \delta - \gamma \tag{16}$$

After that, dynamic stability coefficient DH yields to:

$$DH = \frac{\frac{\pi}{2} - \gamma}{\beta_2} \tag{17}$$

If the position of the centre of gravity CG is in the limiting static position, it means that the straight line $\overline{ECG}$ is perpendicular to the horizontal plane of the slope and the point CG lies above the tipping axis (represented by the point E), then the point CG earns the potential energy PH, which is:

$$PH = m \times g \times \Delta h \tag{18}$$

where: $\Delta h = \overline{ECG} - \overline{ECG} \times \sin(\beta_1 + \gamma)$

and after modification:

$$PH = G \times \overline{ECG}\,(1 - \sin(\beta_1 + \gamma)) \tag{19}$$

Dynamic stability is violated if the angular velocity ω_y is so high that the kinetic energy causes the satisfaction of the condition $KH \geq PH$. From the introduced equations we can discover the limiting angular velocity ω_{HRA} at which the mentioned phenomenon could occur. Critical angular velocity is in the form:

$$\omega_{y_{crit}} = \sqrt{\frac{2G \times \overline{ECG} \times (1 - \sin(\beta_1 + \gamma))}{J_y + 0.1G \times \overline{ECG}^2}} \tag{20}$$

The condition of stability violation is $\omega_y \geq \omega_{y_{crit}}$. The mathematical formula for the other tipping axes is analogical.

Conditions of experiment. The listed angular velocities (Fig. 3) were obtained from the real experimental measurement. Driving manoeuvre was

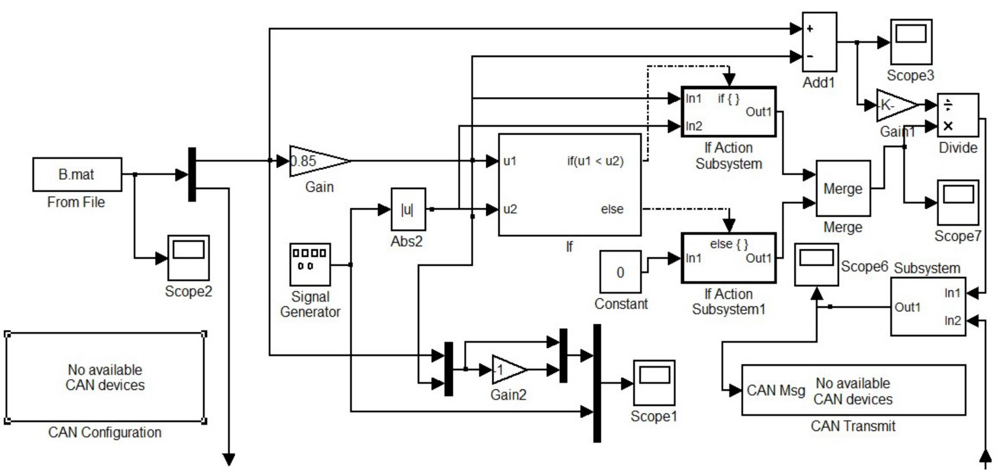

Fig. 4. Scheme of stability control zone

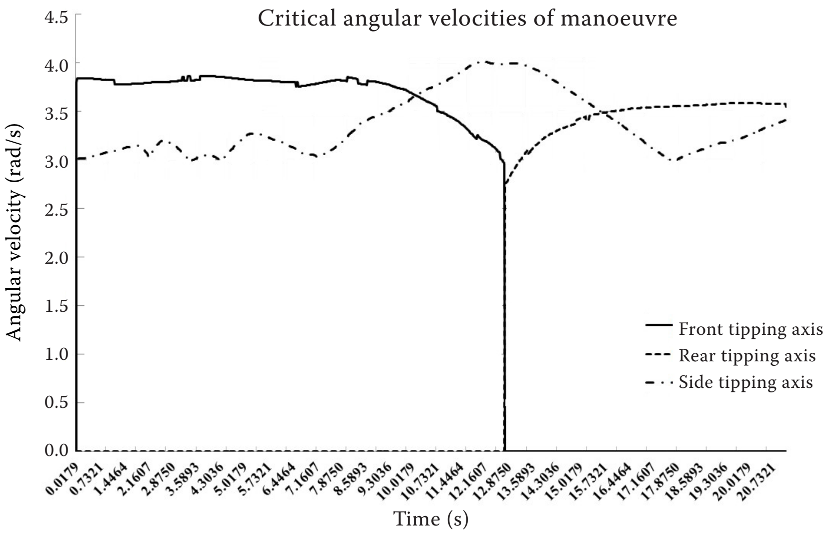

Fig. 5. Critical angular velocities during the manoeuvre

the ride on the slope along the down-grade slope with turning to the down-grade slope. The experimental test was performed following the Standard STN 47 0170. Priority was mainly given to the composition and humidity of the terrain on which the test was performed. The botanical composition of the terrain consisted of plants like *Festuca pratensis* Huds. 15%, *Poa pratensis* L. 30%, *Dactylis glomerata* L. 30%, *Arrhenatherum elativ* 5%, *Alopecurus pratensis* L. 5%, clovers 5%, and other plants 10%. The humidity of the terrain was determined according to the methodology of Jobbágy and Simoník (2006). The share of humidity was lower than 50%, which means that the surface of the terrain was specified as dry.

RESULTS AND DISCUSSION

In Fig. 4, there is depicted the scheme of stability zone control of the agricultural off-road vehicle. It is possible to describe the control principle as a verification of the measured angular velocity from accelerometers or from the gyroscopic sensor placed in the centre of gravity. Verification is in comparing with the critical angular velocity (Fig. 5) at which an unstable position occurs. The output of control is the limitation of engine power and also the velocity if the measured angular velocity comes near to the critical one. The zone of power control is determined in this way, as it is seen in Fig. 6.

This verification is involved in the part of the block scheme *If, If Action Subsystems*, where the real angular velocity is compared with the critical one. If the real angular velocity reaches 60% of the critical angular velocity, the regulation starts to work, and if it is not reached, then the system is not intervened. In order that the operating personnel still has the control of the vehicle, 15% of the engine power is taken away, which stabilizes the vehicle that in a minor extent reacts on the steep-

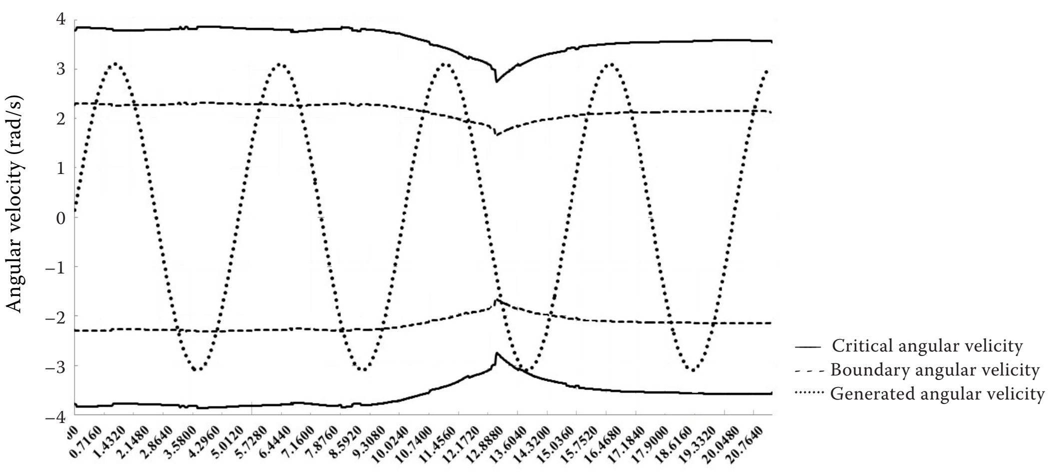

Fig. 6. Regulation of system

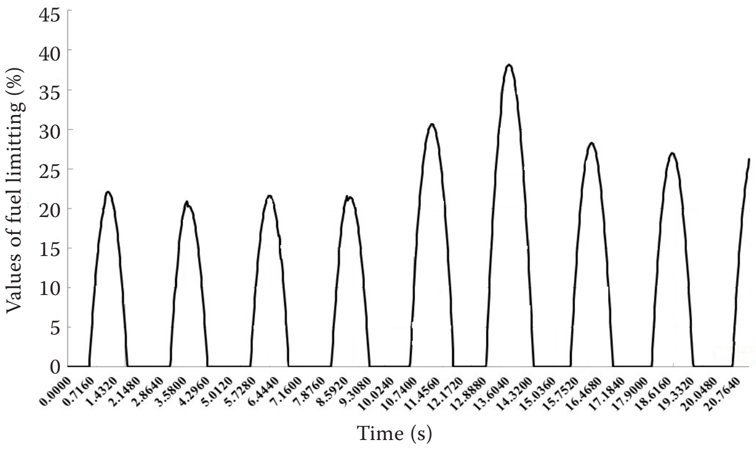

Fig. 7. Response signal of regulator

ness of the amplitude of excitement from the base. However, it is necessary to realise the control for two tipping axes of the machine so that the general stability is inherent. Connecting and validation is in charge of the block *Subsystem*, in which the signals of the control for particular axes of tilt are connected. At present, interconnection and limitation of engine power is possible to be solved via a control unit, which communicates through a *CAN* bus. In older mechanical systems, control would require an independent device, which would restrict the injection of the fuel into the engine. AP regulator is used for control. This type of regulator was chosen because of the absence of knowledge of the dynamical and operating parameters of the specific controlled machine.

Seeing that during the experimental tests the vehicle did not turn over, we simulated the angular velocity by the generated function $\sin(\omega \times t + j)$ to verify the functionality of designed regulator. The response is depicted in Fig. 7.

CONCLUSION

In this contribution, the part of the active stability control system of the agricultural off-road vehicle was presented. It is designated for operation in mountain and foothill areas, where the risk of overturn is very high. This is proven also by analyses of national institutions of occupational safety in the EU. The methodology of acquiring the angular velocities is based on the Standard STN 47 0170. The testing manoeuvres were realised on the machine MT8-222 Synona according to the performed agro-technical operation for the given machine. In the specified way, we obtained the critical angular velocities from experimental measurements. We projected a control system in Matlab, the functionality of which was verified by the generated function as no vehicle rollover occurred during the experiment. By verification we obtained the control signal, the intensity of which is expressed in percentage of the taken engine power of the machine. It is possible to connect the designed system with the control units of the fuel control system (e.g. EDC for Common Rail) through the CAN port. Proposed solution developed at our department can be used also for older machines, which do not include modern active stability control systems.

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