

# The Possibility of Directional Stability Improvement of Articulated Tool-Carriers

Marian J. Lopatka, Tomasz Muszynski

**Abstract**— Our research shown that there are two main reasons of poor directional stability of articulated tractors-toolcarriers: a small frame articulation which always occur due to different reason and limited perception of driver which has insensitivity zones of lateral and angular location. On this basis the principle for stabilization of steering system was worked out. In this paper the model of articulated tractor, assumption and result of simulating research such system are presented.

**Index Terms**—Articulated frame vehicle, directional stability, driver aid system, simulated investigation

## I. INTRODUCTION

Tractors with articulated frame are widely used on construction sites as a basis machines for loaders, handlers, lift-trucks and other tool-carriers because their advantages such as: excellent leading of front attachment and good visibility during steering (attachment turns together with front frame and wheels), high maneuverability (short turn radius), high lifting capability (rigid axis), good mobility (the same tracks of front and rear axles during turning) and simple and durable construction. For this reasons has dominated wheeled equipment weight above 10 tons.

But as a automated or robotized equipment articulated tractors has one feature which limits their usefulness – it is poor directional stability of movement called snacking phenomenon. It is permanent deviations from one side to another and needed correction of path movement develop new one. So, the top speed is limited to 5-6 km/h in remote or automated control systems.

Our research shown that there are two main reasons such behavior : a small frame articulation which always occur due to different reason [6], [8] and limited perception of driver which has insensitivity zones of lateral and angular location [7], [9].

On this basis the principle for stabilization of steering system was worked out.

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## II. CONCEPT OF STABILIZING SYSTEM

Conducted at MUT research [6],[7] shows that main reasons of the snacking phenomenon are :

- initial turns of articulated frame (below  $0,6^\circ$ ),
- residual steering moment in hydraulic cylinders.

In Ackerman steering system due to suitable geometry there are a few torques and forces that can acts straighten road wheels because of play in steering system and stabilize of vehicle movement [2]. Different geometry of steering and play absence in articulated tractors cause the stabilising function of steering system is fulfilled by the driver [7]. Their perception limits accuracy of vehicle leding, directional stability of movement and top permissible speed. In order to improve directional stability of articulated tractors the limited sensitivity of human brain must be supported by driver aid system.

Because the driver sensitivity of angular deviation is limited to about  $1^\circ$  and this is a main cause of poor directional stability of articulated tractor [9] – the driver aid system ought to straighten the frame and eliminate small turns of frame leading to deviations. For this reason new system consist sensor of frame articulation and additional hydraulic circuit for straighten the frame (fig.1). Such system should acts only when main steer valve not operates and their flow ought to be small as possible to not disturb main steering system operations in the case of damage and lowering the energy consumption and jerk accelerations.

As a first step the possibility of such system actions was investigated because the main problem for such solution is high torque of inertia of rotated front and rear parts of tractor. Additional hydraulic system was installed on 20 tons wheel loader and time domain characteristic was determined. For this test the gear pump with electric drive and 10% of nominal flow recommended by steering system manufacturer [3],[4],[11] was selected.

Results (fig.2 and fig.3) shown that during 0,3 s time of operation pump achieved almost nominal flow  $8 \text{ dm}^3/\text{min}$  and frame turns about  $0,5^\circ$  – value comparable to initial turns. It was positive because shown high speed of turn corrections almost without of any time lag and low power consumption at small pump delivery (it is very important from safety point of view because additional system working regardless of main steering system must be weaker) and there is possibility to operate in such system.

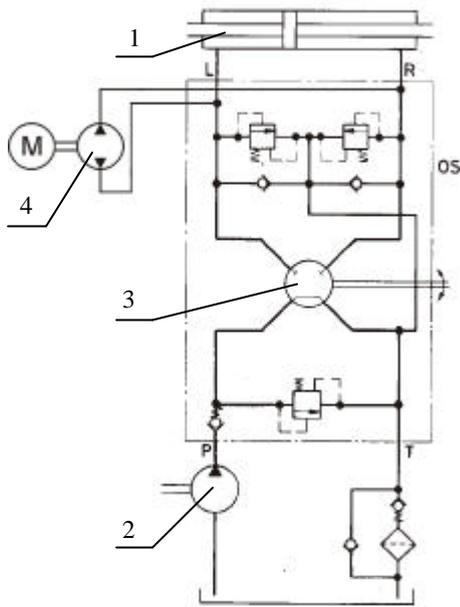


Fig. 1. Diagram of hydraulic steering system with additional driver aid system : 1 – steering cylinder, 2 – main pump, 3 – steering valve, 4 – additional pump with electric drive

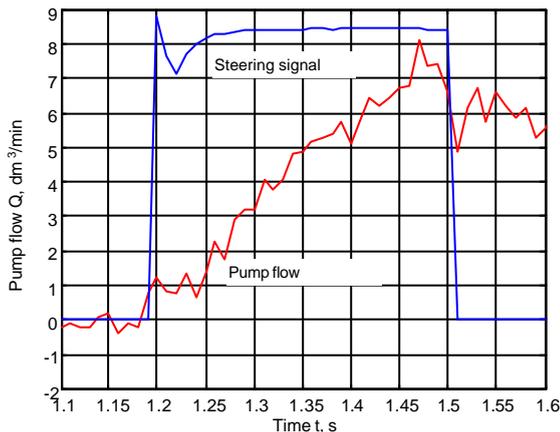


Fig. 2. Additional gear pump delivery during 20 t loader frame turns

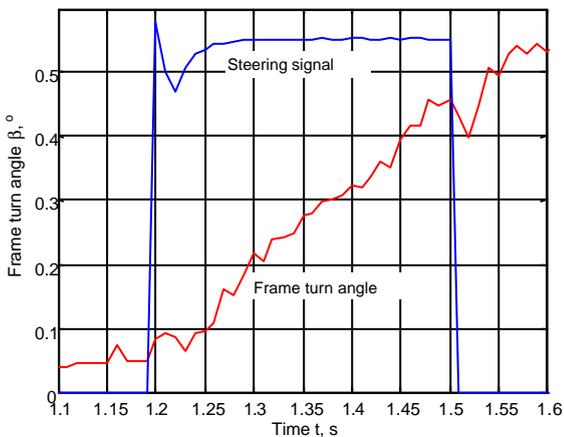


Fig. 3. Frame turns as influence of additional gear pump operation

### III. MODEL OF ARTICULATED TRACTOR AS A STEERED VEHICLE

For evaluating operational efficiency such system and estimating their main parameters the simulation model of articulated tractor as a steered vehicle was needed. Available models of articulated tractors for example [1,5,10] are aimed on investigation of the critical speed phenomenon and for controlled movement simulation the new model was evaluated.

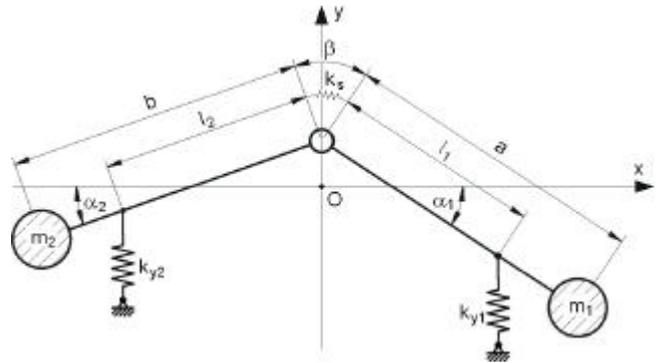


Fig. 4. Model of articulated vehicle frame turns

As a steering signal the frame turn angle  $\beta$  (fig. 4) was selected. Because lack of suspension system in such tractors only tires and hydraulic steering system affect and disturb the trajectory of movement. For these reasons building a simplified linear model and taking for consideration the lateral stiffness of tires, their damping effect and hydraulic steering cylinders properties the equations describing such system are follow :

$$m_1 \ddot{y} - m_1 a \ddot{\alpha}_1 + m_2 \ddot{y} - m_2 b \ddot{\alpha}_2 + k_{y1}(y \cdot \mathbf{a}_1 \cdot l_1) + k_{y2}(y - \mathbf{a}_2 \cdot l_2) = -c_1(\dot{y} - a\dot{\alpha}_1) - c_2(\dot{y} - b\dot{\alpha}_2) \quad (1)$$

$$J_1 \ddot{\alpha}_1 - m_1 a \ddot{y} + m_1 a^2 \ddot{\alpha}_1 + k_s(\mathbf{a}_1 + \mathbf{a}_2 - \mathbf{b}) + k_{y1}(\mathbf{a}_1 l_1 - y) = c_{l1}(\dot{y} - a\dot{\alpha}_1) + c_s(\dot{\mathbf{b}} - \dot{\alpha}_1 - \dot{\alpha}_2) \quad (2)$$

$$J_2 \ddot{\alpha}_2 - m_2 b \ddot{y} + m_2 b^2 \ddot{\alpha}_2 + k_s(\mathbf{a}_1 + \mathbf{a}_2 - \mathbf{b}) + k_{y2}(\mathbf{a}_2 l_2 - y) = c_{l2}(\dot{y} - b\dot{\alpha}_2) + c_s(\dot{\mathbf{b}} - \dot{\alpha}_1 - \dot{\alpha}_2) \quad (3)$$

where:

$y$  – lateral deviation of pivot;

$a, a_2$  – angular deviation of front and rear parts of tractor;

$m_1, m_2$  – mass of front and rear parts of tractor;

$J_b, J_2$  – moments of inertia of front and rear parts of tractor;

$c_b, c_2$  – damping coefficients of front and rear tires;

$c_s$  – damping coefficient of hydraulic cylinders;

$k_b, k_2$  – stiffness coefficients of front and rear tires;

$k_s$  – stiffness coefficient of hydraulic cylinders;

$l_1, l_2$  – distance between front and rear axis and pivot;

$a, b$  – distance between front and rear center of gravity and pivot.

Assuming  $A_1 = \mathcal{L}(\mathbf{a}_1)$ ,  $A_2 = \mathcal{L}(\mathbf{a}_2)$ ,  $Y = \mathcal{L}(y)$ ,  $B = \mathcal{L}(\mathbf{b})$  after Laplace's transformation - the frame movement described by angular deviation of front and rear parts of tractor and lateral deviation of pivot as forced by frame turn angle can be evaluated as :

$$Y = \frac{A_1(m_1 a^2 s^2 + k_{y1} l_1 + c_1 a s) + A_2(m_2 b s^2 + k_{y2} l_2 + c_2 b s)}{m_1 s^2 + m_2 s^2 + k_{y1} + k_{y2} + c_1 s + c_2 s} \quad (4)$$

$$A_1 = \frac{Y(m_1 a^2 s^2 + k_{y1} + c_1 l_1 s) + B(k_s + c_s s) - A_2(k_s + c_s s)}{J s^2 + m_1 a^2 s^2 + c_1 l_1 a s + c_s s + k_s + k_{y1} l_1} \quad (5)$$

$$A_2 = \frac{Y(m_2 b^2 s^2 + k_{y2} + c_2 l_2 s) + B(k_s + c_s s) - A_1(k_s + c_s s)}{J s^2 + m_2 b^2 s^2 + c_2 l_2 b s + c_s s + k_s + k_{y2} l_2} \quad (6)$$

In this case according to fig. 5 - the lateral velocity of articulated tractor movement is described as :

$$v_{y1} = v(\mathbf{a}_1 - \mathbf{d}_1 + \mathbf{j}) \quad (7)$$

$$v_{y2} = -v(\mathbf{a}_2 + \mathbf{d}_2 + \mathbf{j}) \quad (8)$$

where:  $v$  – vehicle velocity;

$f$  – overall angular tractor deviation (fig. 5) for small angles of turns given by :

$$\mathbf{j} = \frac{y_1 - y_2}{L_0} \quad (9)$$

where:  $L_0$  – tractor wheel base;

$y_1, y_2$  – lateral wheels deviations;

$d_1, d_2$  – slip angles front and rear wheels given by :

$$\mathbf{d}_i = \frac{k_{yi}(\mathbf{a}_i l_i - y_0)}{k_d} \quad (10)$$

where :  $k_{di}$  – cornering stiffness of tires.

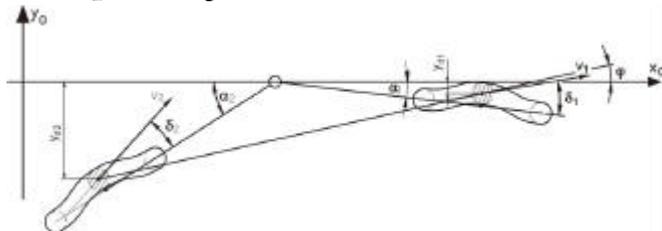


Fig. 5. Model of articulated vehicle movement

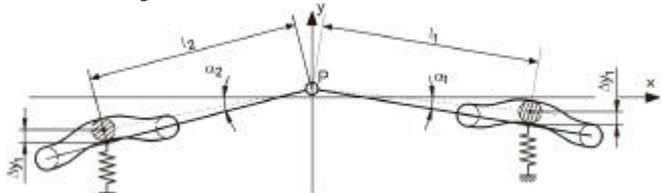


Fig. 6. Influence of tire stiffness on articulated vehicle movement

But lateral movement of pivot is smaller because lateral stiffness of tires (fig. 6) about :

$$\Delta y_i = \mathbf{d}_i \cdot r_d \quad (11)$$

where :  $r_d$  – radius of tire deformation (fig. 7).

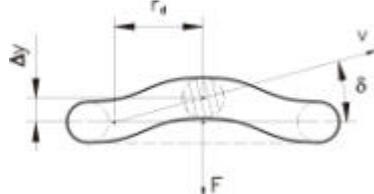


Fig. 7. Influence of tire stiffness on slip angle

And because lateral ground reactions are equal :

$$R_i = k_{di} \cdot \mathbf{d}_i = k_{yi} \cdot \Delta y_i \quad (12)$$

finally slip angles can be described as :

$$\mathbf{d}_1 = \frac{(\mathbf{a}_1 l_1 - y_0)}{r_d} \quad (13)$$

$$\mathbf{d}_2 = \frac{(\mathbf{a}_2 l_2 - y_0)}{r_d} \quad (14)$$

Connecting these equations in Matlab-Simulink environment the model of articulated tractor as steered vehicle was build.

#### IV. MODEL VERIFICATION

For estimating usefulness of presented model their verification on real articulated tractor was conducted. Tests depended on comparison and assessment of conformity registered and simulated trajectory of movement and lateral deviations of front and rear parts of tractor. As exemplar the lateral deviations registered during 20 t loader trials (fig. 8) was used. Force for simulation was worked out on the basis of registered frame steer angle (fig. 9) but the curvature was simplified (fig. 10). All model parameters was determined for mentioned 20 t articulated wheel loader (Tab. I). Calculated trajectory for these parameters and force are shown on fig. 11.

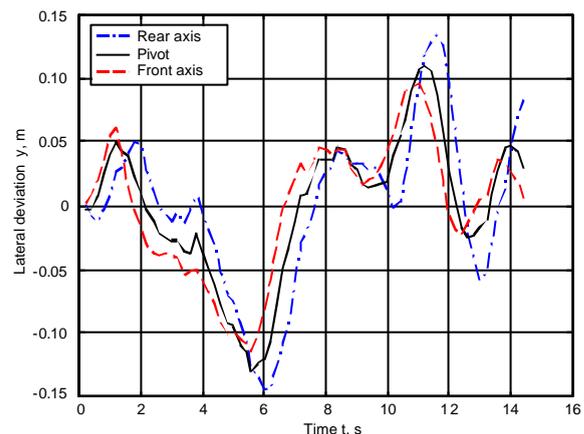


Fig. 8. Example of registered lateral deviations during articulated loader straight movement at speed 28 km/h

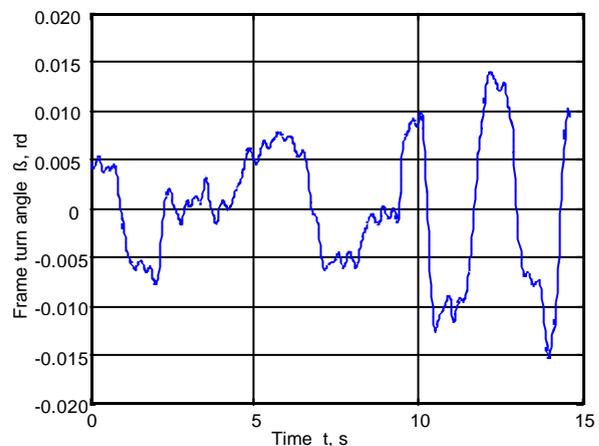


Fig. 9. Frame turn angle registered during articulated loader straight movement at speed 28 km/h

Conducted test shown not to high conformity because on simulated paths of movement front and rear axis there are no distinct difference which are observed in real motion. Lateral deviations of rear axle are about 5 cm higher – it is important because angular deviations are strongly depended on them.

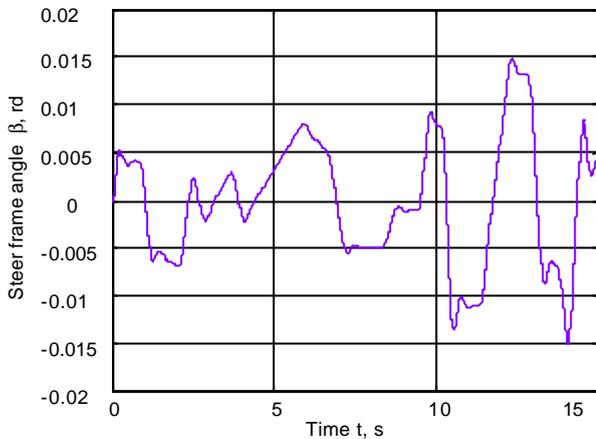


Fig. 10. Simplified force used for simulation of articulated loader straight movement at speed 28 km/h

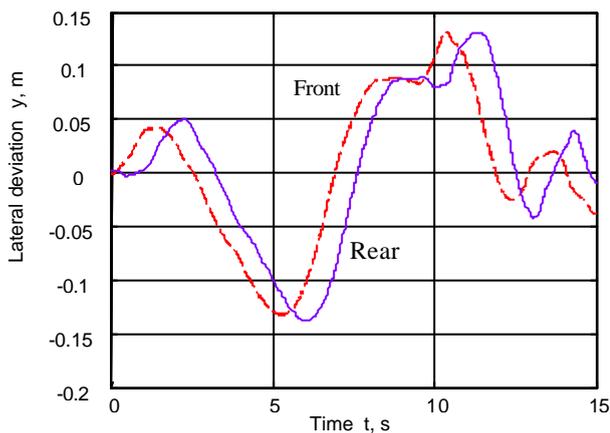


Fig. 11. Simulated lateral deviations during articulated loader straight movement at speed 28 km/h

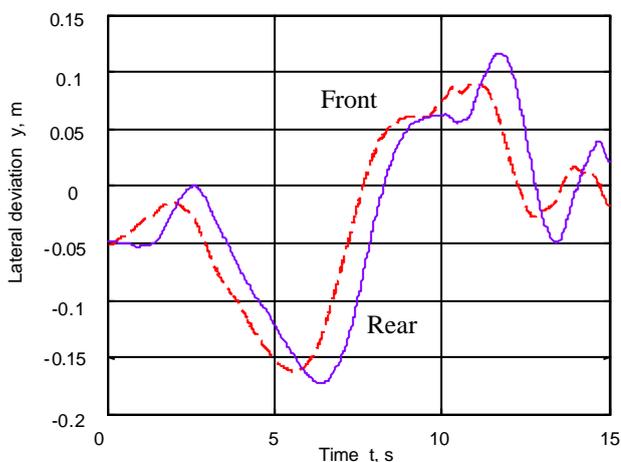


Fig. 12. Simulated lateral deviations during articulated loader straight movement at speed 28 km/h with improvement model of articulated tractor

Table I. Parameters of articulated tractor model

Parameter	Value
$m_1$	9000 kg
$m_2$	9500 kg
$J_1$	42000 kgm <sup>2</sup>
$J_2$	56000 kgm <sup>2</sup>
$k_1$	8,1·10 <sup>5</sup> N/m
$k_2$	4,1·10 <sup>5</sup> N/m
$c_1$	5·10 <sup>4</sup> Ns <sup>2</sup> /m <sup>2</sup>
$c_2$	5·10 <sup>4</sup> Ns <sup>2</sup> /m <sup>2</sup>
$k_c$	224·10 <sup>5</sup> Nm/rad
$c_c$	1·10 <sup>5</sup> Nms <sup>2</sup> /rad <sup>2</sup>
$l_1$	1,57 m
$l_2$	1,57 m
$a$	1,70 m
$b$	2,00 m

Undertaken analysis shown that reason of this small conformity is high moment of inertia of big size tires. Improved model (with influence of tire rubber high mass on tread) gives high accuracy of lateral deviations (fig. 12) and enabling investigation of driver aid system on simulations way.

### V. DRIVER AID SYSTEM EFFICIENCY CRITERION

Vehicles with Ackerman steering system are characterized by low driver effort level compared to vehicles steered with frame turns [2]. Lowering frequency of driver actions is main goal for new system. To accomplish this target the knowledge about articulated tractor driver behavior is needed. Assuming that main factor causing driver reaction are lateral and angular deviations sensed by him – the articulated loader was rebuild. The steering wheel was moved to the front of vehicle and mounted above the axis to accelerating sensing of deviation and gives clear ties between deviations and driver reaction.

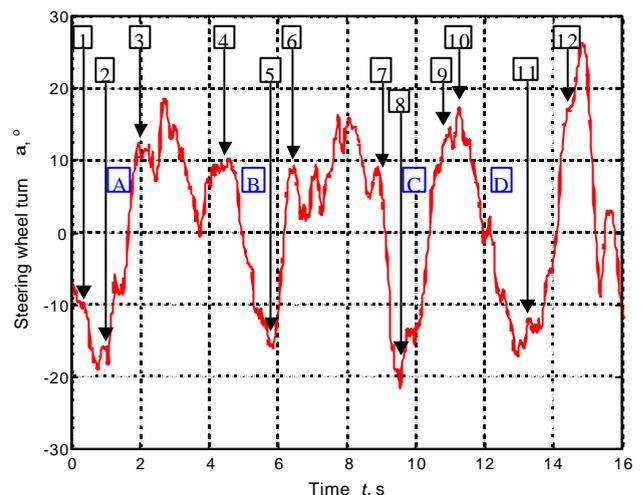


Fig. 13. Driver reactions as a steering wheel turn during modified loader straight movement at speed 28 km/h

For such modified loader investigation of driver reaction was conducted. They shown that after rebuilding there are period of high

and low driver activity which enabling the selection of starting point of reaction – fig. 13 (for example point 4, 7, 10). Looking at lateral driver deviation curvature (fig. 14) it can be pointed that driver act when lateral deviation is about  $\pm 0,2$  m (and this value was achieved only two times) but correlation with starting point is not clear. Better correlation is seen on fig. 15 where is shown angular deviation of driver. Average achieved deviation is about  $1,2^\circ$  and appear eight times in the same time. During car movement this deviation not exceed  $0,5^\circ$  [2]. It means that angular deviations are dominating in articulated vehicle movement and are crucial for driver effort. Eliminating residual frame angle with driver aid system the angular deviation can be limited and driver effort lowered.

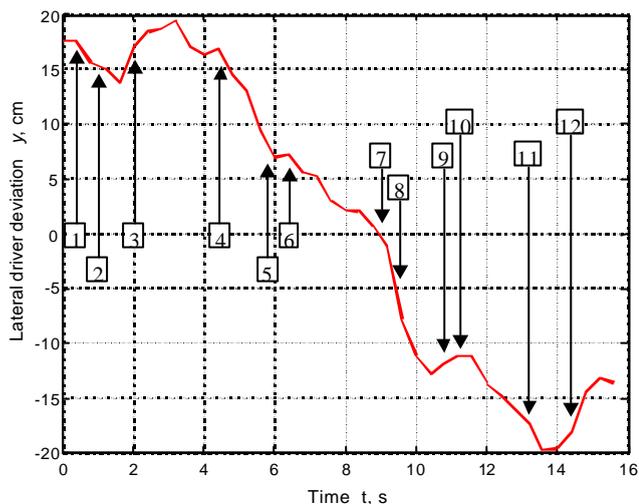


Fig. 14. Lateral driver deviations during modified loader straight movement at speed 28 km/h

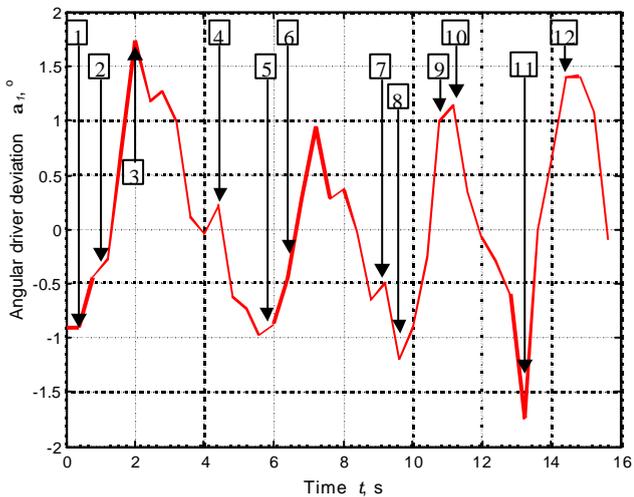


Fig. 15. Angular driver deviations during modified loader straight movement at speed 28 km/h

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Efficiency of such system can be evaluated with time period from start to achieving one of determined limits of deviations : lateral  $0,2$  m and angular  $1,2^\circ$ .

### VI. INVESTIGATION OF DRIVER AID SYSTEM EFFICIENCY

Using worked out model and admitted criteria the simulating tests was conducted for additional hydraulic system operation with different pump delivery. Dynamic parameters of hydraulic system was determined on the basis previous conducted tests (fig. 2). As a starting position was assumed that residual frame turn angle is  $0,6^\circ$  – half value of evaluated perception limit and sensors of deviations are over front axis.

Results shown that to small flow in hydraulic system can't straighten frame in desired time (fig. 16) and although driver effort is lowered, angular deviation is still growing and stay main cause for driver reaction. Higher flow (fig. 17) enabling straighten frame before achieving angular deviation limit and stabilization of movement. In this case driver reaction is caused similar to another vehicles by lateral deviation.

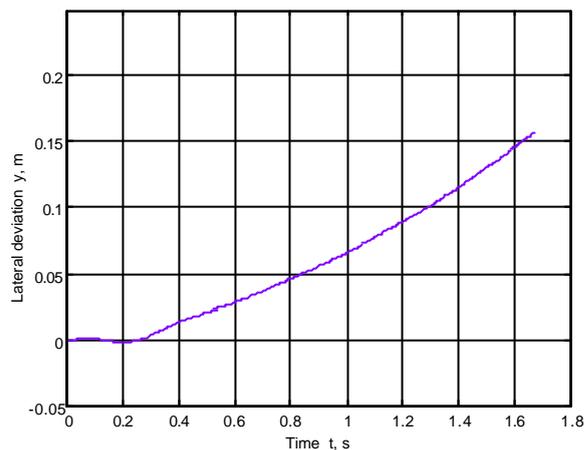


Fig. 16. Lateral driver deviations during loader straight movement at speed 60 km/h and operating driver aid system with pump flow  $4 \text{ dm}^3/\text{min}$

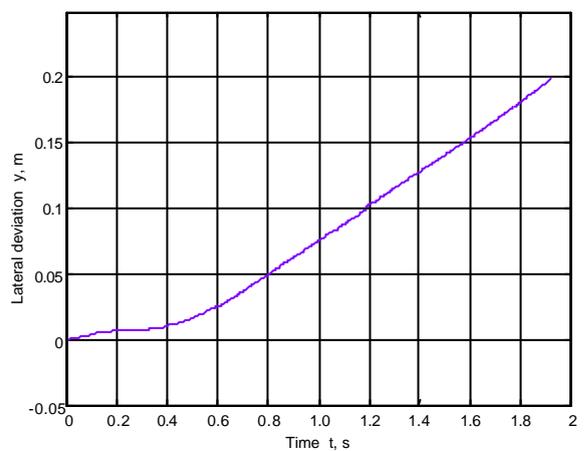


Fig. 17. Lateral driver deviations during loader straight movement at speed 60 km/h and operating driver aid system with pump flow  $10 \text{ dm}^3/\text{min}$

Efficiency of stabilizing system as function of pump delivery is presented in Table II. At relatively low speed (about 28 km/h) the system operating with 10 dm<sup>3</sup>/min nominal pump delivery can give high stability of movement – the driver reaction is needed after about 20 s. But when sensors are connected with rear frame of tractor their acting is delayed and disrupt. So, the efficiency of stabilizing system in this case is limited – Table III.

Table II. Time to achieving deviation limits at speed 28 km/h with sensors over front axis

Nominal pump delivery	Time to deviation $v = 0,2 \text{ m}$	Time to deviation $\alpha = 1,2^0$
$Q = 10 \text{ dm}^3/\text{min}$	20 s	-
$Q = 9 \text{ dm}^3/\text{min}$	13 s	-
$Q = 8 \text{ dm}^3/\text{min}$	11,6 s	-
$Q = 7 \text{ dm}^3/\text{min}$	7,3 s	-
$Q = 6 \text{ dm}^3/\text{min}$	7,2 s	-
$Q = 5 \text{ dm}^3/\text{min}$	27 s	-
$Q = 4 \text{ dm}^3/\text{min}$	4,4 s	-
$Q = 3 \text{ dm}^3/\text{min}$	3,6 s	-
$Q = 2 \text{ dm}^3/\text{min}$	2,9 s	-
$Q = 1 \text{ dm}^3/\text{min}$	-(4 cm)	1,54 s
$Q = 0$	-(2 cm)	1,36 s

Table III. Time to achieving deviation limits at speed 28 km/h with sensors over pivot and connected with rear frame

Nominal pump delivery	Time to deviation $v = 0,2 \text{ m}$	Time to deviation $\alpha = 1,2^0$
$Q = 10 \text{ dm}^3/\text{min}$	2,45 s	-
$Q = 9 \text{ dm}^3/\text{min}$	2,34 s	-
$Q = 8 \text{ dm}^3/\text{min}$	2,33 s	-
$Q = 7 \text{ dm}^3/\text{min}$	2,13 s	-
$Q = 6 \text{ dm}^3/\text{min}$	2,2 s	-
$Q = 5 \text{ dm}^3/\text{min}$	2,5 s	-
$Q = 4 \text{ dm}^3/\text{min}$	-(16 cm)*	1,67 s
$Q = 3 \text{ dm}^3/\text{min}$	-(14 cm)*	1,50 s
$Q = 2 \text{ dm}^3/\text{min}$	-(14 cm)*	1,45 s
$Q = 1 \text{ dm}^3/\text{min}$	-(13 cm)*	1,40 s
$Q = 0$	-(10 cm)*	1,17 s

## VII. CONCLUSION

In the case of articulated tractor opposite to Ackerman steered vehicle angular deviations are dominated. Moreover conducted tests shown that human perception is limited and deviation can reach : lateral 0,2 m and angular 1,2<sup>0</sup>.

Carried out research shown that limitation of snacking phenomenon during movement of articulated tool-carriers with driver aid system is possible. The principles of their operation was verified on simulation way. Developed unique model of articulated tractor as steered vehicle shown full usefulness for such simulation

and can be used in future works on developing the electronic control system and sensors necessary for demonstrator building.

Proposed system can increase productivity of articulated equipment due to higher travel speed but it is strongly recommended to connect sensors of deviation with front frame of vehicle.

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