

# DIRECTIONAL DYNAMICS PROBLEMS OF AN ARTICULATED FRAME STEER WHEELED VEHICLES

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## Abstract

One of the current trends in industrial wheeled vehicles is to increase their mobility, i.e. the ability to travel at speeds ranging from 50 even up to 100 km/h, e.g. military engineering machines. As a result of the high demands put on such vehicles several previously unknown problems with their dynamics have come to light.

The paper presents the results of experimental and MSC Adams view simulation tests for snaking movement of an articulated frame steer vehicle such as bucket loader. The model precisely accounts for machine units' mass and geometry along with their inertia. Additionally, the model includes hydrostatic steering gear submodel and its geometry and the submodel of a drive train with dynamic properties of multi-dimensional tyres as well. Experimental and virtual tests demonstrated that directional dynamics of articulated frame steer wheeled vehicles are largely affected by total (equivalent) steering gear torsional stiffness, dynamic properties of large-size tyres and type of vehicle's wheel drive. Simulation studies used the results of experimental studies whereby the simulation outcome provides interesting and original guidelines for articulated frame steer vehicles manufacturers.

**Keywords:** articulated frame steer vehicles, snaking, experimental and simulation tests, steering system stiffness, dynamic properties of multi-dimensional tyres

## 1. Introduction

In the literature one can find many classifications of vehicles, such as: automotive vehicles, transport vehicles, off-road vehicles, construction vehicles, municipal vehicles, agricultural vehicles, military vehicles, special-purpose vehicles and so on. In recent years in order to systematize the terminology the term industrial vehicles has been commonly applied to the group of non-automotive vehicles, especially in the foreign literature, e.g. in "The Magazine for Industrial Vehicle Technology, Design & Engineering". This group of industrial vehicles includes first of all mobile mechanical work machines (e.g. loaders), in military applications referred to as earthmoving machines, and transport vehicles (e.g. haulers). Generally, it is a category of vehicles ranging from forklift trucks (used for on-site material handling) to vehicles used in the particular branches of industry, e.g. in construction, mining, agriculture, forestry, etc. Considering the above, one can conclude that the term industrial vehicles will eventually be adopted also in the Polish specialist literature.

One of the current trends in this class of vehicles is to increase their mobility, i.e. the ability to travel at a speed of 50-60 km/h and even at 100 km/h (e.g. military engineering machines) [1-2]. As a result of the high demands put on such vehicles several previously unknown problems with their dynamics have come to light.

In the literature on the subject one can find several publications and papers on the stability of conventional automotive vehicles, e.g. [3-5]. Unfortunately, the reported results are not directly applicable to industrial vehicles which markedly differ from automotive vehicles. Depending on the operational requirements, e.g. concerning vehicle manoeuvrability, industrial vehicles have different steering systems [6].

Steering systems with wheel or axle steering, articulated frame steering and their combinations are referred to as geometric steering systems [6]. To make the picture complete, it should be added that there are also skid steering systems of wheeled chassis, in analogy to conventional tracked undercarriage systems.

Unlike the various industrial vehicle undercarriage systems, automotive vehicles typically have the conventional Ackermann front wheel steering system. Moreover, the significant differences between automotive vehicles and industrial vehicles mainly stem from the dynamic properties of industrial vehicles' large-size or inertial tyres and steering gear. The steering gear provides the vehicle with the desired drivability and additionally, guarantees the necessary motion stability and consequently, safety when travelling at high speed on public roads [6]. This double role of the steering gear is especially critical in industrial articulated frame steer vehicles in which the steering gear interacts with two large-mass vehicle frames connected by an articulation joint.

## 2. Problems with articulated industrial vehicle motion stability

An industrial vehicle moves in complex operating conditions in which many factors adversely affect the strategy of movement selected for the vehicle by the operator (Fig. 1).

Some of the factors, e.g. the driving direction, driving speed  $V_0$ , two-wheel or all-wheel drive, the driving force and the braking force, change according to the driver's will. A large number of the factors, such as: the distribution of masses  $m_p$ ,  $m_t$ ,  $m_u$ , wheelbase  $L$ , wheel track  $B$ , the location of articulation joint PS and oscillation joint PW, steering gear's torsional stiffness and damping  $k_s$ ,  $c_s$  or traction and stiffness and damping  $k_{z,i}$ ,  $k_{y,i}$ ,  $c_{z,i}$ ,  $c_{y,i}$ ,  $c_{\alpha,i}$  of the large-size tyres, are connected with the vehicle specifications determined early on in the design process. These factors affect, among other things, the oscillation ( $\pm\Delta\gamma$ ) of the machine frames in the articulated joint, the determined trajectory of motion and the number of operator corrections needed to keep the vehicle within the prescribed path. The remaining factors, e.g. terrain unevenness and slope, vehicle movement resistances, wind, etc., are a group of disturbances and as a rule have a random character.

It should be mentioned that because of the flexibility of the steering gear and large-size tyres in articulated frame steer vehicles, as opposed to non-articulated (rigid-body) vehicles, several of the above factors generate oscillations of the main vehicle frames in articulation joint PS by angle  $\pm\Delta\gamma$  and oscillations of, e.g., the rear axle in oscillation joint PW, resulting in vehicle snaking. If, for the analysis, geometric centre  $O_p$  of the front axle (Fig. 1) is assumed as a representative point, the principal indicators of snaking for an articulated frame steer vehicle are: amplitude  $A_w$  of horizontal axle deviation  $O_p$  and frequency  $1/T_w$  of its occurrence.

Moreover, when the vehicle's design specifications and the driver's qualifications and psychophysical predisposition are taken into account, the above critical factors affecting the articulated frame steer vehicle cause a deviation of the actual trajectory of motion from the prescribed one, defined as a change in a vehicle's course angle  $\Psi$  (Fig. 1).

Snaking and the spontaneous change in a vehicle's course angle  $\Psi$  make it difficult, or even impossible, to meet the vehicle's roadworthiness certification requirements [12-13]. According to the requirements [12-13], the steering system should ensure that the vehicle can drive forward along a straight line at maximum speed  $V_{o(max)}$  on a road section with a dry horizontal level hard surface, with minimum length  $S_{min}=100m$  and a width equal to 1.25 of the external tyre track  $W$  (Fig. 2). On this road section the operator is allowed to make the standard steering gear corrections. In order to prevent the steering gear course corrections from causing additional adverse dynamic oscillations of the articulated frame steer vehicle's main frames the steering system must meet proper requirements. Such an innovative adaptive steering system for the industrial vehicle, developed and tested in the Division of Off-Highway Vehicles Engineering at Wrocław University of Technology, is presented in [14].



Unfortunately, no comprehensive and exhaustive numerical or full-scale experimental studies into the above problems can be found in the literature. The existing papers, e.g. [1-2] and [7-10], do not address the many practical questions. The latest work in this field is [11].

The present paper is an attempt to significantly extend the knowledge about the snaking of wheeled articulated industrial vehicles.

### 3. Experimental and virtual studies of flexible properties of steering gear and tyres of articulated frame steer bucket loader

The Division of Off-Highway Vehicles Engineering at Wrocław University of Technology carried out experimental studies on the Ł220 articulated frame steer bucket loader (belonging to the Division), driving at different speeds on a concrete ground.

The results of the studies confirmed the significant influence of the flexibility of the steering gear and that of the large-size tyres on the motion stability of such a vehicle.

In order to determine the influence of solely the steering gear on the snaking process the articulation joint was mechanically locked whereby the vehicle-with-stiff-frame effect was obtained for the machine in this particular option of measurement. In the course of the tests the following were measured: angle oscillations  $\pm\Delta\gamma$  of the machine frames in the articulation joint, the forces in the steer actuator-by means of special measuring pins [15, 16], the displacements of the piston rods of the actuators, the torque generated in the steering gear and the change in the angle of the machine 's course.

Total (equivalent) steering gear torsional stiffness  $k_s$  of loader Ł220 was determined while the latter was standing still, by generating torque  $M_s$  in the articulation joint and measuring articulation angle  $\gamma$  of the vehicle's frames. The measurements were performed including or excluding the hydraulic pipes and including or excluding the contact of the large-size tyres with the ground (Fig. 3).

The results of the tests show that the flexibility of the hydraulic pipes and that of the large-size tyres have a strong influence on total (equivalent) torsional stiffness  $k_s$  of the steering gear and consequently, on the snaking of vehicles of this class.

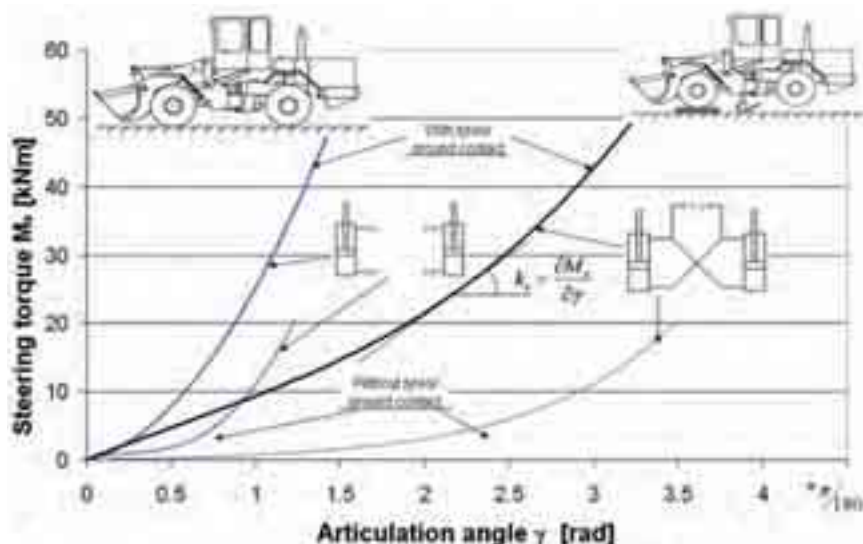


Fig. 3. Experimentally determined total (equivalent) steering gear torsional stiffness  $k_s$  of loader Ł220 with flexible hydraulic pipes and large-size tyres/ground contact taken into account and without these factors being taken into account

Unfortunately, neither the literature nor the manufacturer catalogues provide information on the changes in the volume of flexible hydraulic pipes (pipes swelling) as a function of their type, geometry and internal pressure. Therefore pipes of different types and with different geometry, i.e. with a single

fabric ply (1SN) and with two fabric plies (2SN), with nominal diameters: DN10, DN16, DN20 and length:  $L=0.45$  m,  $L=0.64$  m and  $L=0.84$  m, were tested on a special test stand [17].

Selected test results are shown in Fig. 4.

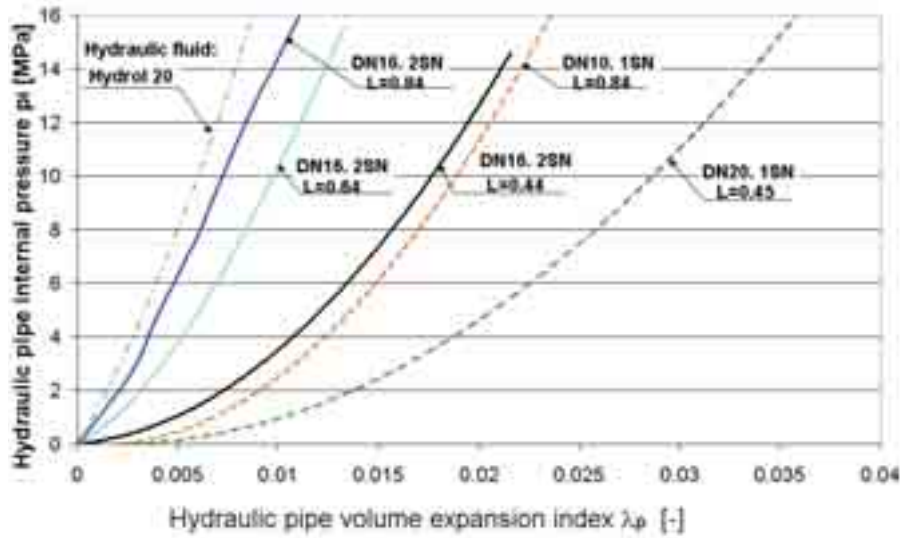


Fig. 4. Experimentally determined volumetric expansion index  $\lambda_p$  of flexible pipes versus pressure  $p_i$  of hydraulic fluid for different pipe structures with single fabric ply (1SN) and two fabric plies (2SN), and their different geometries

In order to qualitatively evaluate the flexibility of the tested pipes, pipes volume expansion index  $\lambda_p$ , was defined as:

$$\lambda_p = \frac{\Delta V_p}{V_0}, \quad (1)$$

where:

$V_0$  – the pipes initial internal volume [ $\text{m}^3$ ],

$\Delta V_p$  – a change in the pipes internal volume, resulting from pressure  $p_i$  of the hydraulic fluid [ $\text{m}^3$ ],

was introduced.

Considering the above and taking into account the nonlinearity of the phenomenon, the pipes' flexibility bulk modulus  $B_p$  can be written as:

$$B_p = \frac{\partial p_i}{\partial \lambda_p}, \quad (2)$$

other

$$B_p = \frac{\Delta p_i}{\Delta \lambda_p}, \quad (3)$$

where:

$p_i$  – the pipes internal pressure [ $\text{N}/\text{m}^2$ ].

In order to determine the flexibility of the large-size tyres of the Ł220 loader, their radial and lateral stiffness was tested [11].

The Division of Off-Highway Vehicles Engineering at Wrocław University of Technology has many years of experience in virtual prototyping of complex mechanical and mechatronic constructions, especially in the numerical modelling of industrial vehicles (e.g. Fig. 5) in DADS and MSC ADAMS software environments [6, 11, 18].

For example, recently a complex model of wheeled articulated frame steer loader Ł220 manufactured by the Construction Machinery Plant Fadroma in Wrocław has been created (Fig. 5).



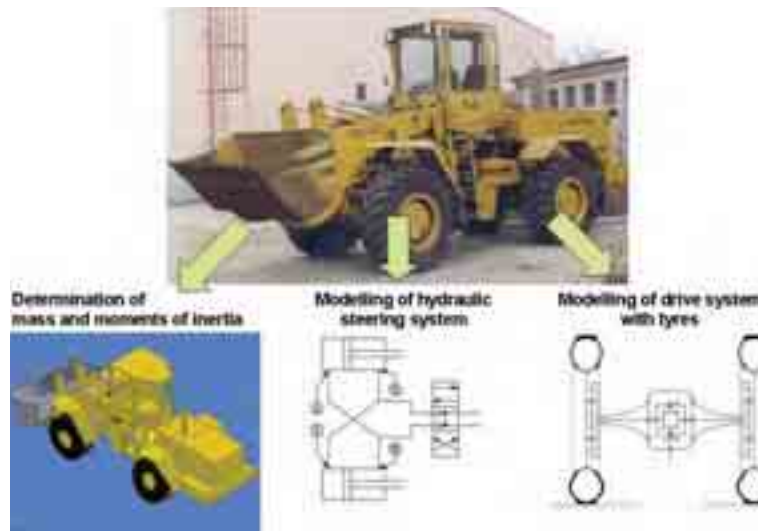


Fig. 5. Loader Ł220 and its simulation structural model

The loader is one of the test machines belonging to the Division. Specialist software and the documentation obtained from the manufacturer were used to identify the geometry, the mass, the centre of gravity and the inertia of the particular components of the machine.

The hydraulic steering system was modelled taking into account the flexibility of the hydraulic fluid and that of the flexible hydraulic pipes. The parameters were determined on a measuring stand specially built for this purpose [11]. Also the geometry of the location of the hydraulic actuators in the steering gear and that of the actuators themselves were taken into account. The model of the hydraulic steering system was experimentally verified by applying it to the real system.

Additionally, the drive system, consisting of differential gears and reduction gears in the wheels equipped with flexible large-size tyres, was modelled. The comprehensive dynamic model of the Ł220 loader was experimentally validated on a test track. Apart from the flexibility of the large-sized tyres, also their inertness, caused by the transition states of the loads generated in the course of tyre-ground interactions, was included in the dynamic model of the large-size tyres (Fig. 5) [6].

A technical stability criterion based on standards [12] and [13] was used for studying the vehicle motion trajectory. According to the standards, a vehicle is stable until the geometric centre of the front axle (a selected representative point) reaches value  $O_p$  of the maximum allowable lateral displacement (Fig. 2). On the basis of [12] and [13] it was determined that the lateral displacements of the Ł220 loader should not exceed 280 mm.

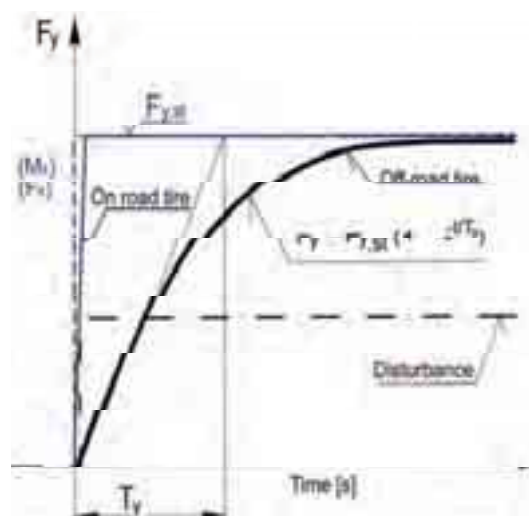


Fig. 6. Comparison of dynamic properties of car tyres and large-size tyres of industrial vehicles

In both the numerical studies and the experimental studies on the test track the vehicle moved on an obstacle course consisting of single obstacle acting on only its right front wheel. The reason was to prevent any superimposition of the oscillations of the main frames in the articulated joint, which could occur when the vehicle's right front wheel and its right rear wheel drove in immediate succession over the obstacle.

Exemplary results of the simulation studies are shown in Fig. 7-9.

Figure 7 shows the effect of the percentage of the free air in the hydraulic fluid on the oscillations of the machine's frames in the articulation. An increase in the amount of gases in the hydraulic fluid is known to result in a significant reduction in the flexibility bulk modulus of the fluid whereby the stiffness of the whole steering system decreases.

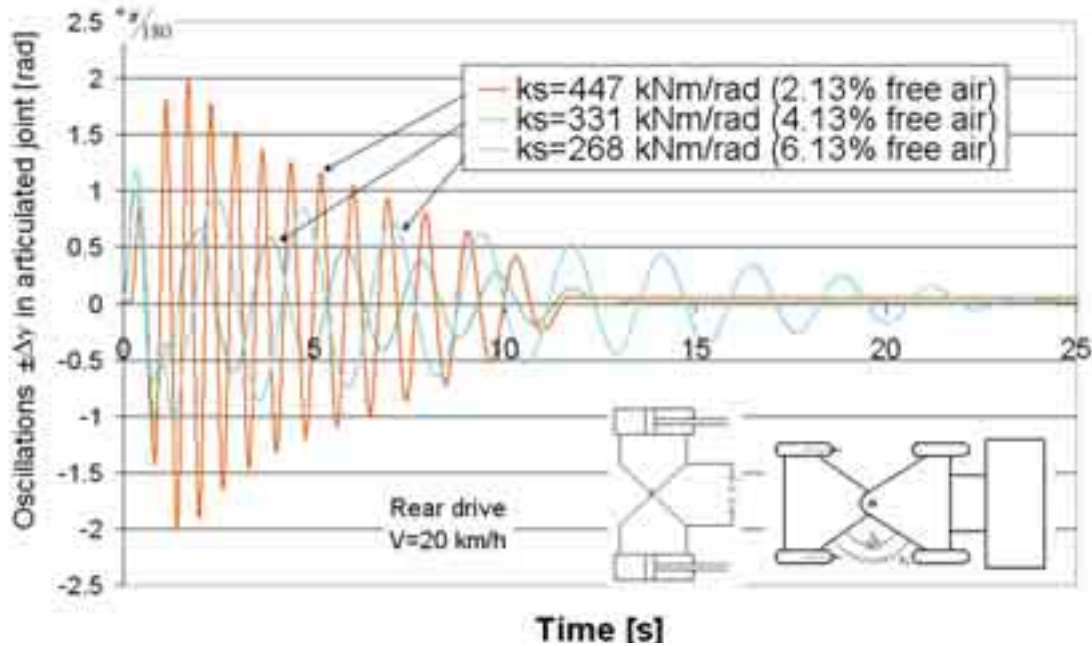


Fig. 7. Simulation results showing effect of equivalent torsional stiffness  $k_s$  of steering gear on oscillations  $\pm\Delta\gamma$  of frames of loader L220

The studies showed that the decrease in the equivalent torsional stiffness of the steering system adversely affects the duration and amplitude of the oscillations of the machine frames in the articulation joint. The vibrations have a direct bearing on the trajectory of motion of the articulated frame steer vehicle and result in the vehicle's travelling shorter road length than that specified by standards [12] and [13]. For example, as a result of the increase in free air content from 2.13% to 4.13% the torsional stiffness of the steering system decreased by over 25% and the distance travelled decreased from 17 to 13 m whereby the covered minimum distance of 20 m required by the standards was not achieved.

Figure 8 shows the effect of a change in equivalent torsional stiffness from the minimum value below at which vehicle's jackknifing occurs to the maximum value at which the vehicle's front and rear frames can no longer rotate in the articulation joint on the trajectory of motion of the articulated frame steer vehicle. It appears from the simulations that an increase in equivalent torsional stiffness  $k_s$  of the tested vehicle's steering gear has a significant effect only up to a certain limit value above which it no longer significantly contributes to the lengthening of the distance covered in the standard allowable area [12, 13].

Moreover, it should be noted that below  $k_s=57\text{kNm/rad}$  vehicle jackknifing occurs during which the articulation angle changes so much that the vehicle tends to move approximately on a circle. It is also noteworthy that an increase in the stiffness of the steering gear contributes to an increase in harmful dynamic loads in the steering gear's components, considerably shortening the life of the latter.

Considering the above, there is an urgent need to formulate a method of determining the optimum equivalent torsional stiffness of the steering gear for a given articulated frame steer vehicle. Such a method is proposed in [6] and [11].

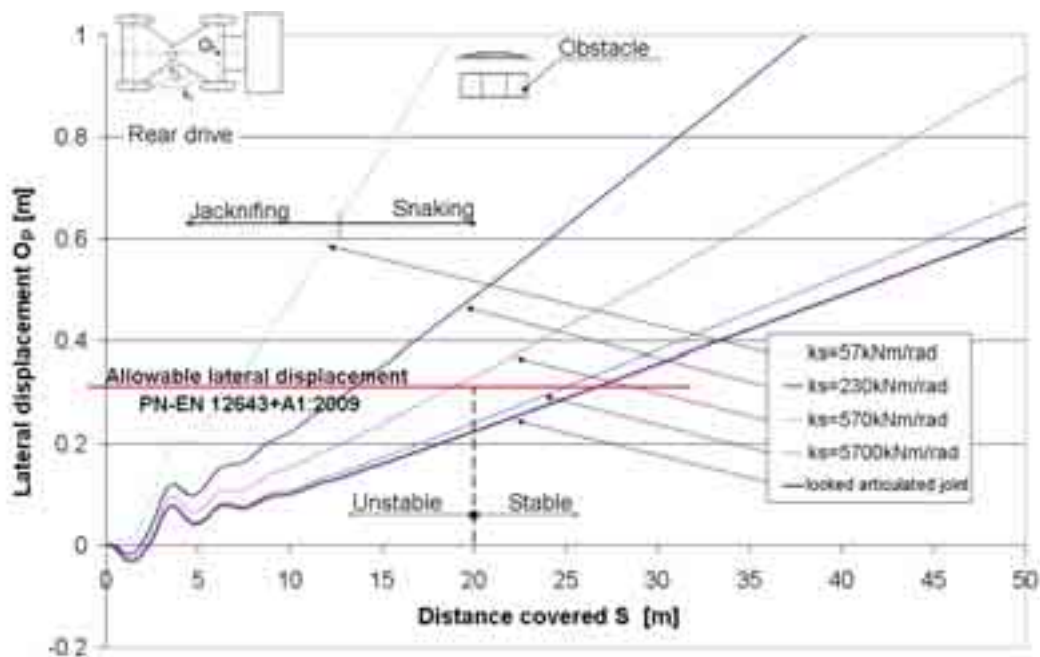


Fig. 8. Simulation results showing effect of (minimum to maximum) equivalent torsional stiffness  $k_s$  of steering gear on trajectory of motion of loader Ł220

The simulation studies also showed that the type of articulated frame steer vehicle wheel drive is a major factor having a bearing on vehicle snaking. The results of the relevant simulations are shown in Fig. 9.

As it was to be expected the simulation results showed that the front axle drive is the most advantageous as regards snaking. The front wheel drive “tightens” the vehicle, making it easier for the hydraulic steering system to put the vehicle into the straight-ahead-drive position. A similar behaviour is observed during the travel of a car-trailer unit. The most disadvantageous as regards snaking is the rear wheel drive. In the case of the all-wheel drive without kinematic discrepancy in the drive train, the duration of the oscillations assumes values intermediate between those characteristic of the above two cases. Whereas in the case of the four wheel drive with kinematic discrepancy and circulating power, the results, depending on the type of the discrepancy, are closer to the ones for the front wheel drive or the rear wheel drive.

## Conclusion

The results of the experimental and simulation studies, reported in this paper represent only a part of the extensive programme of research on the dynamics of industrial vehicles, being carried out in the Division of Off-Highway Vehicles Engineering at Wrocław University of Technology. For instance, an original mathematical model (incorporating all the major factors) of the steering gear with two hydraulic actuators was presented in [6] and [11]. This model makes it possible to determine the total torsional stiffness of the steering gear already when designing the vehicle. Innovative technical solutions significantly improving the driving stability of articulated frame steer vehicles, based on the experimental and simulation results, have been developed and patents for them have been filed.

The solutions will be presented in the next papers.



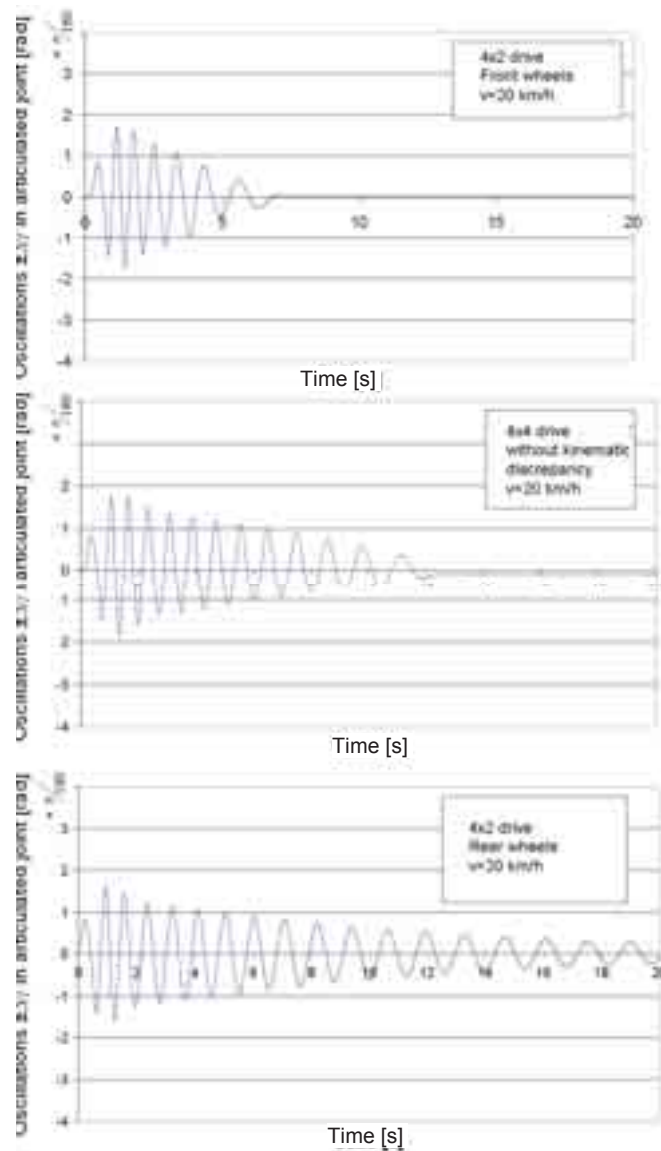


Fig. 9. Simulation results showing effect of type of vehicle's wheel drive on oscillations of frames in vehicle's articulation joint

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