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DIRECTIONAL STABILITY IMPROVEMENTS WITH COUPLING FORCE CONTROL ON TRACTOR/SEMI-TRAILER COMBINATIONS

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Abstract

In a tractor/semi-trailer combination the tractor is usually equipped with anti lock device and the semi-trailer is less maintained than the tractor. Since sensing couplers are available from manufacturers, a well-adapted control algorithm can eliminate jack-knifing and high peaks in coupling force during braking. In order to improve the stability and reduce the forces on the kingpin, a three dimensional non-linear mathematical model has been developed for digital simulation. The model includes the following properties:

- non-linear characteristics of road–tyre interface,
- vehicle units regarded as rigid bodies,
- suspensions with non-linear spring and damping behaviour,
- detailed description of the brake system.

Special efforts were made to use appropriate characteristic curve for the elements in the brake system and these results were validated to experimental measurements. Though the study is limited to the examination of the behavior of the vehicle during braking in a turning maneuver, several test runs were made to reach the best dynamical performance and the minimum number of sensors. Test results are compared in conventional and controlled case.

Keywords: articulated vehicles, vehicle stability, digital simulation, jack-knifing, braking model.

1. Introduction

The requirements against transportation vehicles force engineers to re-examine and improve old conventional structures. The weight of trucks and trailers is growing, the safety rules are getting stricter. In the case of truck–semi-trailer combination, the usually less maintained semi-trailer can cause problems when its brake system is not correctly fitted to that of the haulage vehicle. Another source of problems when only the truck is equipped with an anti-lock device, and there is a great chance for the driver to loose the directional control during an emergency braking in a turn. The current investigation below sets for to find out whether significant stability improvement can be achieved by applying brake-control for coordinating the brake systems of vehicle units.

Heavy duty trucks in Europe are mounted with ABS by law. This system is capable to observe the dynamical state of the truck, the operation of the brake system and the brake control sign for semi-trailer. From the observed signals it can judge if the braking performance of the towed vehicle unit is poor or too high, and is able to modify the brake control sign. For example, the dynamic state of a truck can be followed by electric pressure sensors mounted on the air-springs. From these, one can read the current axle loads and the changes due to braking or downhill driving. Further information comes from strain gauges mounted on the horizontal pin of the coupler. An appropriate arrangement allows to measure the horizontal and vertical components of the force acting on the pin (see *Fig. 1*). The analog signs must be converted into digital information and be processed by a micro controller. This digital control unit — using an algorithm which is not detailed here — evaluates a control sign for semi-trailer brake control valve so that the coefficient of friction be equal on all axles of truck and semi-trailer.

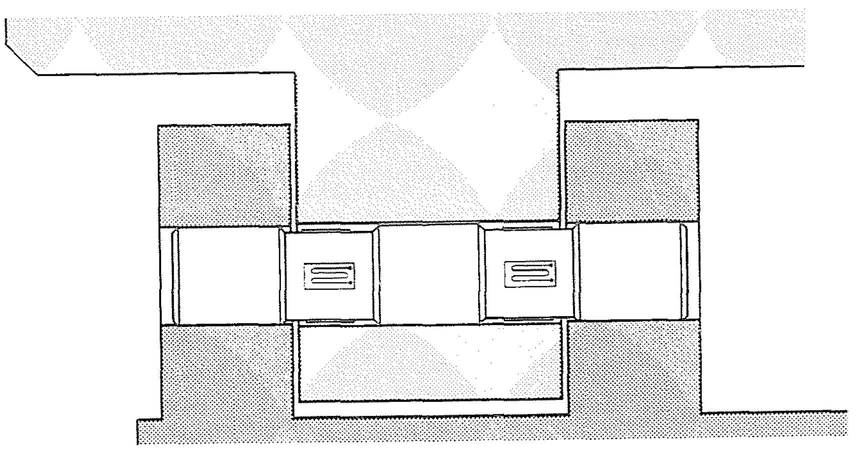

Fig. 1. Horizontal pin of the coupler equipped with strain gauge

2. Vehicle Model

The articulated vehicle is a complex mechanical system of rigid bodies connected by springs and dampers. For want of space, the details of mathematical analysis are not presented here, but the method will be briefly outlined below. *Fig. 2* shows the model used in digital simulation. The system consisting of rigid elements can be described by matrices in 3 dimensions. The equations governing the motion of the system can be written in the form

$$\mathbb{M}\ \ddot{\mathbf{q}} = \mathbb{F}\left(\mathbf{q}_s, \dot{\mathbf{q}}_s, \mathbb{F}_{q_s}\right), \tag{1}$$

where $\mathbf{q}_s$ is a column vector (of dimension 18x1) whose elements describe the state of the vehicle at any given time; $\dot{\mathbf{q}}_s$ is the first derivative of the state vector of $\mathbf{q}_s$; $\ddot{\mathbf{q}}_s$ is the second derivative of $\mathbf{q}_s$ with respect to time; and $\mathbb{M}$ is a (18×18) matrix, whose elements are functions of $\mathbf{q}_s, \dot{\mathbf{q}}_s$, and the parameters of the vehicle. The right hand side of *Eq.* (1) depends on $\mathbf{q}_s, \dot{\mathbf{q}}_s$ and generalized forces $\mathbb{F}_{q_s}$.

The equations of motion of the wheels are given by

$$\varepsilon_i = \frac{F_{xi} r_i - T_{bi}}{I_i}; \qquad (i = 1, 2, \dots, 6), \tag{2}$$

where F_{xi} is the longitudinal force at the road–tyre interface; r_i is the height of the axle, which is assumed to be equal to the effective rolling radius of a freely rolling wheel; T_{bi} is the brake torque and I_i is the mass moment of inertia of the wheel. The subscript i denotes that the above parameters apply to the ith wheel.

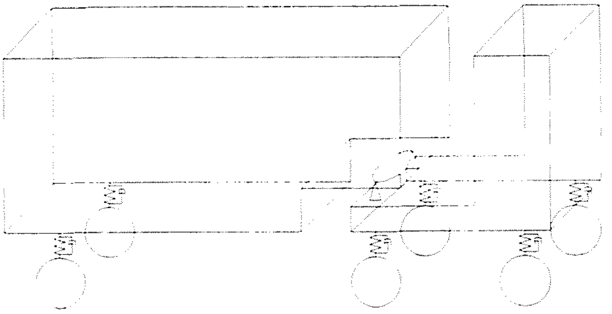

Fig. 2. Vehicle model for computer simulation

The set of equations represented by (1) and (2) can be integrated numerically if the tyre forces, brake moments and other forces acting on the vehicle system are defined by appropriate relationships.

A special procedure can be used to satisfy the kinematical constraints at the kingpin. Instead of using the technique of Langrangian multiplicators, very rigid springs and dampers were applied in constrained directions at the coupler. Hence, the coupling force is given from the relative displacements and distortions of tractor and semi-trailer pin locations.

To avoid oscillations in the numerical solution, the time unit used in integrating the equations of motion of the vehicle is about four times larger than that applied in integrating the equations of motion of the wheels. The equations of motion are integrated with the well-known Runge–Kutta method in order to save computational time.

2.1 Tyre Model

The tyre forces generated at the road–tyre interface, which depend on the interaction of the tyre with the road, were found by the 'magic' formula (see PACEJKA, 1987). However, the analysis of the tyre forces was further simplified. These forces depend on the nature of the road-tyre interface, slip, slip angle, normal load and the velocity of the wheel along the wheel plane. The slip S is given by

$$S = \frac{V - r\omega}{V}, \tag{3}$$

where r is the rolling radius of the wheel, ω is the angular velocity of the wheel, δ is the steering angle and V is the forward velocity of the wheel.

The slip angle α is defined as the angle between the direction of motion of the center of wheel and the wheel plane. Thus, the slip angle is given by

$$\alpha = \tan^{-1} \frac{V_y}{V_x} - \delta, \tag{4}$$

where V_x is the velocity of the wheel along its longitudinal axis, V_y is the velocity of the wheel along its lateral axis.

The longitudinal and lateral force on the tyre is given by

$$F_x = \mu_1 \sin\left(\mu_2 \tan^{-1}(\mu_3 S_x)\right) \frac{|S_x|}{\sqrt{S_x^2 + S_y^2}} F_z, \tag{5}$$

$$F_y = \mu_1 \mu_4 \sin\left(\tan^{-1}(\mu_3 \mu_5 S_y)\right) \frac{|S_y|}{\sqrt{S_x^2 + S_y^2}} F_z, \tag{6}$$

where μ_i are constants describing the road and tyre characteristics, F_z is the wheel load, S_x and S_y are the longitudinal and lateral components of the slip vector.

2.2 Brake Model

In a vehicle combination consisting of a haulage truck and a semi-trailer, the brake system of the haulage truck comprises a control valve which allows the adaption of the brake pressure imparted to the brakes of the semi-trailer to the brake pressure transferred to the brakes of the haulage truck.

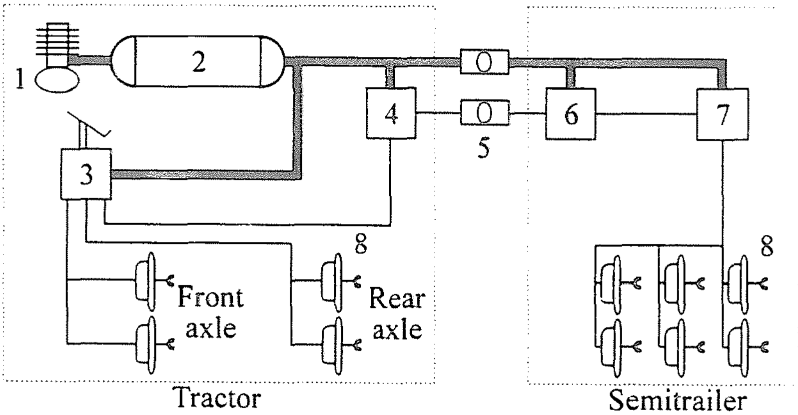

Fig. 3. Arrangement of the elements of the brake system

1. Compressor;
2. Reservoir;
3. Brake control valve;
4. Semi-trailer control valve;
5. Coupling block;
6. Semi-trailer brake valve;
7. Release valve;
8. Brake-cylinders.

The adaptation is carried out by means of an electrical control unit in which the actual dynamic state of the vehicle during a braking sequence is observed and treated as under- or overbraked case. A control member on the control valve is actuated automatically so that the semi-trailer is supplied with a higher or a lower brake pressure relative to the haulage vehicle depending on whether the actual coupling force was lower or higher than expected. The adaptation means that the brakes of the haulage truck and of the semi-trailer can be used better under various operating conditions.

For modeling, a simplified brake system was assumed (See *Fig. 3*). The semi-trailer brake control valve was assembled with an electric regulator to adjust the semi-trailer control pressure corresponding to the current dynamic state. A sample diagram of brake-cylinder pressures can be seen in *Fig. 4*. In the calculation the analogy from electrotechnics was used: brake pressure $\approx$ voltage and mass flow $\approx$ current (*Fig. 5*). For a short section

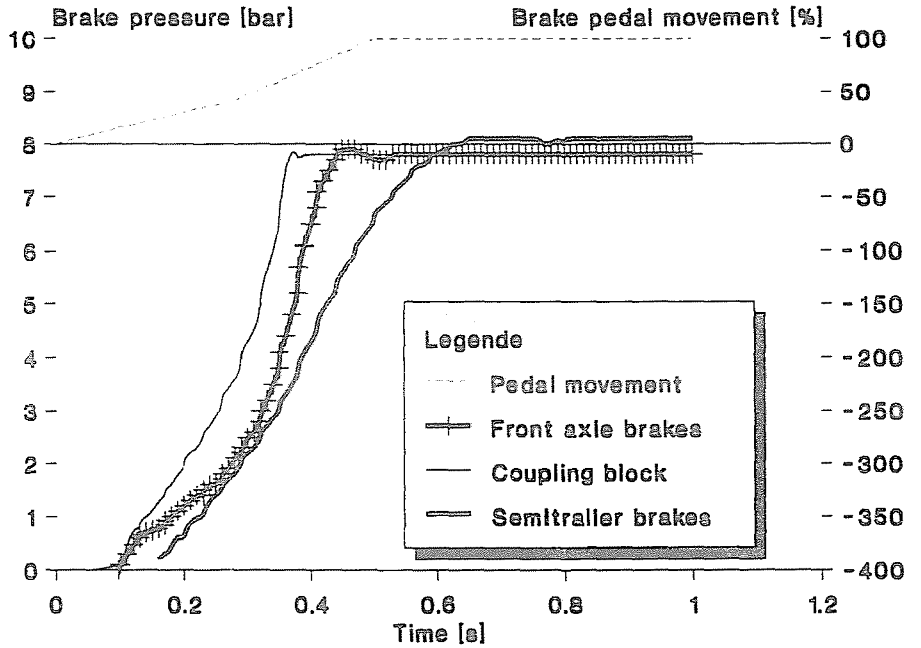

Fig. 4. Sample diagram of brake pressure changes in time

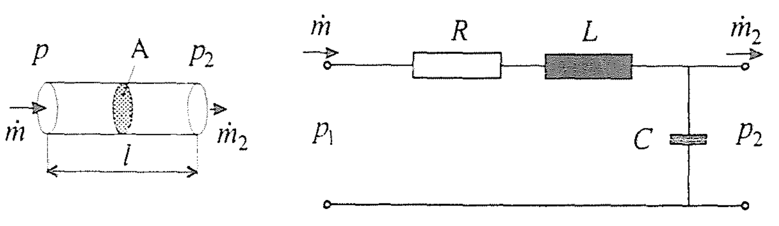

Fig. 5. Similarity from electrotechnics

of brake pipe with length of l, we can write that the pressure difference is

$$p_1 - p_2 = \frac{8\pi v l}{A^2} \dot{m}_1 \,. \tag{7}$$

So in the case of laminar flow, the resistance has the form of

$$R_{LAM} = \frac{8\pi v l}{A^2} \,, \tag{8}$$

where v is the kinematic viscosity and

$$v = f\left(\frac{p + p_0}{p_0}\right) ; \tag{9}$$

A is the cross-sectional area of the pipe and l is the length of the pipe section.

In the case of turbulent flow, the pressure difference is

$$p_1 - p_2 = \lambda \frac{l}{d} \frac{\dot{m}_1}{2\rho A^2} \dot{m}_1 , \tag{10}$$

where λ is the coefficient of resistance, d is the diameter of the pipe and ρ is the density of the compressed air:

$$\rho = f\left(\frac{p + p_0}{p_0}\right) . \tag{11}$$

Hence, the resistance can be written as

$$R_{TURB} = \lambda \frac{l}{d} \frac{\dot{m}_1}{2\rho A^2} . \tag{12}$$

Since the force

$$K = A\,(p_1 - p_2) \tag{13}$$

acts on the air elements of mass

$$m = \rho A l , \tag{14}$$

the acceleration (b) will be

$$b = \frac{\mathrm{d}v}{\mathrm{d}t} = \frac{1}{\rho A} \frac{\mathrm{d}\dot{m}}{\mathrm{d}t} . \tag{15}$$

The governing equation here is

$$p_1 - p_2 = \frac{l}{A} \frac{\mathrm{d}\dot{m}}{\mathrm{d}t} , \tag{16}$$

so the L inductivity is given by

$$L = \frac{l}{A} . \tag{17}$$

In the case of isothermal state-change we can use

$$L = \frac{4}{3}\frac{l}{A} \tag{18}$$

instead. The mass of gas in the pipe element is as a function of p_2 pressure in the case of isothermal state-change from the general gas law

$$m = \frac{p_2 A l}{RT}, \tag{19}$$

where R is the gas constant and T is the absolute temperature. Introducing the velocity of sound

$$c_a = \sqrt{\kappa RT}, \tag{20}$$

we have

$$m = \kappa \frac{Al}{c_a^2} p_2, \tag{21}$$

where κ is the adiabatic constant (1.4 for air). The relation among mass, mass flow, pressure and pneumatic capacity is as follows:

$$p_2 = \frac{c_a^2}{\kappa A l} m = \frac{c_a^2}{\kappa A l} \int (\dot{m}_1 - \dot{m}_2)\,\mathrm{d}t; \tag{22}$$

$$p_2 = \frac{l}{C} \int (\dot{m}_1 - \dot{m}_2)\,\mathrm{d}t; \tag{23}$$

$$C = \kappa \frac{Al}{c_a^2}. \tag{24}$$

The whole air-brake system of a vehicle combination can be modeled as a set of pipeline elements connected to each other. The volumetric changes in the brake-cylinders must be also taken into consideration. The reservoir and the valves can be treated as special boundary conditions for the related differential equations. For instance, the pressure at the reservoir is always constant ($p = 7.5$ bar). After the appropriate discretization in the space and in the time domain, we get

$$\dot{m}_{1,t+\Delta t} = \dot{m}_{1,t} + \left(\frac{p_{1,t} - p_{2,t}}{R} - \dot{m}_{1,t} \right) \frac{\Delta t}{\frac{L}{R}}, \tag{25}$$

and

$$p_{2,t+\Delta t} = p_{2,t} + (\dot{m}_{1,t} - \dot{m}_{2,t}) \frac{\Delta t}{C}. \tag{26}$$

To avoid numerical instability, $\Delta t \ll \frac{L}{R}$ value has to be used.

3. Simulation Conditions

Various operating conditions used in digital simulation, namely, road conditions, initial speed and testing procedure are described below:

3.1 Road Conditions

The digital simulation was performed on two roads, A and B. Road A is a typical high friction surface (dry asphalt). Road B is a typical medium friction surface (wet asphalt). The maximum μ values in μ-slip curves for road A and B are 0.7 and 0.3.

3.2 Initial Conditions

On roads A and B the simulated vehicle moved at an initial speed of 80 km/h. Then an appropriate steering procedure followed to bring the vehicle into a steady-state turn. Once the vehicle was in a steady-state turn, (i. e. the yaw rate of the tractor was constant) brakes were applied in such a way that the pressure in brake chamber reached the maximum value of 7.5 bar in 0.5 second.

3.3 Steering Procedure

On roads A and B, the front wheels of the tractor were steered through an angle of 0.05 radian so that the vehicle took a right-hand turn, and the lateral acceleration under steady-state conditions was about 0.15 g.

4. Criteria for Judging the Directional Behavior of a Tractor/Semi-trailer

It is widely recognized that during an emergency braking maneuver, an undesirable response of a vehicle. Directional instability or a loss of directional control may occur. Loss of directional control appears whenever there is a significant reduction in the side-force coefficient of the steerable wheels. Directional instability of a tractor/semi-trailer could be either aperiodic or oscillatory. It is doubtful whether oscillatory instability occurs. The aperiodic instability, which is of practical importance, is of two forms: one involving the tractor, which is known as jack-knifing and is associated with locking tractor rear wheels, and the other involving trailer swing, which is associated with locking the semi-trailer wheels. To determine from the time histories of the variables of the motion which kind of aperiodic instability has occurred, suitable criteria have to be used.

4.1 Loss of Directional Control

During the simulated braking in a turning maneuver, if the path of the center of gravity of the tractor is a straight line, it can be concluded that there is a loss of directional control. Obviously, if the intended path is a straight line, loss of directional control is of no consequence and the vehicle remains on the lane. However, when braking in a turning maneuver if a loss of directional control occurs, the vehicle will go out of the lane.

4.2 Jack-knifing

The time history of the yaw angle of the tractor is required to determine whether or not a vehicle has a tendency to jack-knife. The yaw angle of the tractor, (ψ), the yaw rate of the tractor, ($\dot{\psi}$), and the articulation angle, η, are defined in *Fig. 6*.

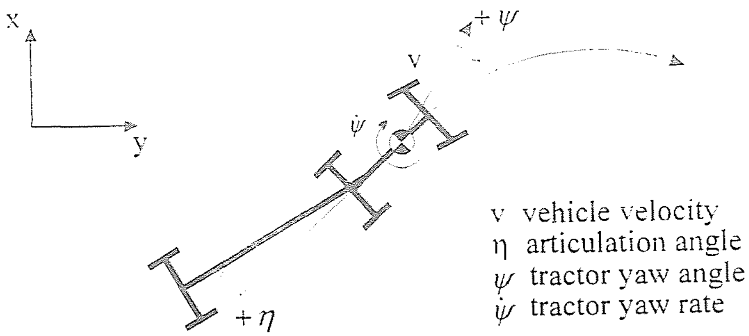

Fig. 6. Sign convention for motion parameters used in evaluating the directional performance of the vehicle

When the vehicle is in a steady state, right-hand turn without braking, the yaw angle of the tractor is negative, but constant; the yaw rate is also constant. However, when the tractor jack-knifes, its yaw angle becomes positive and increases rapidly. The yaw rate also increases rapidly as shown in *Fig. 7*. It may be noted that the rapid increase in yaw angle results in a rapid decrease of the articulation angle. In this case, the change in articulation angle is primarily caused by a change in the yaw angle of the tractor.

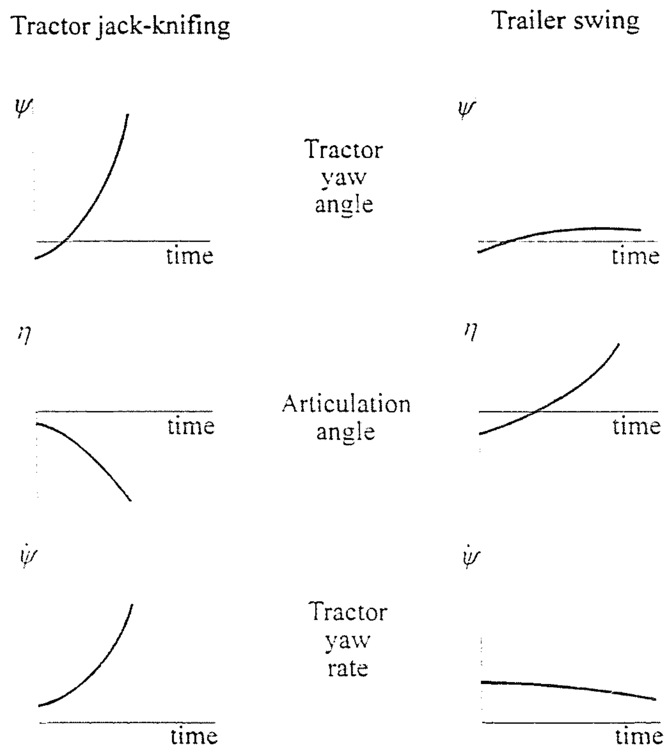

Fig. 7. Qualitative representation of tractor jack-knifing and trailer swing using motion parameters

4.3 Trailer Swing

The time history of the articulation angle is required to determine whether or not the trailer has a tendency to swing. When the vehicle is in steady-state, right-hand turn without braking, the articulation angle is constant and negative. However, if the trailer swings, the articulation angle becomes positive and increases rapidly without a corresponding change in the yaw angle of the tractor. The time history of the articulation angle when trailer swings is shown in *Fig. 7.*

5. Discussion of results

The results for the following cases are presented:
1. No braking,
2. Semi-trailer overbraked,
3. Semi-trailer poorly braked,

4. Controlled maneuver with initially overbraked semi-trailer,
5. Controlled maneuver with initially underbraked semi-trailer.

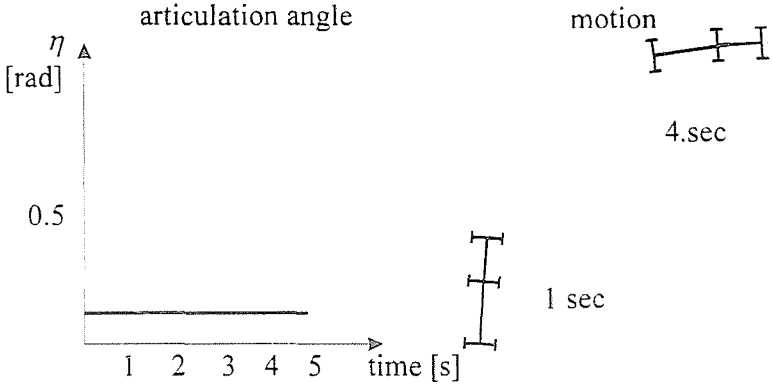

Fig. 8. Steady-state turn

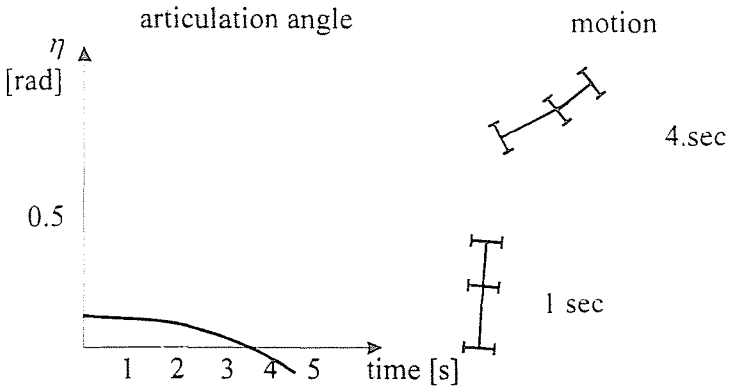

Fig. 9. Overbraked semi-trailer

For want of space, only results on road B are presented here. *Fig. 8* shows a steady-state turn with identical initial conditions except for the case when no brakes were applied. This path was used as reference for the other cases. *Fig. 9* shows the case of overbraked semi-trailer. It can be seen that the articulation angle changes its sign and the semi-trailer swings out of the lane. This occurs when the brake system of the semi-trailer works as if it were loaded but actually it is empty. In *Fig. 10* the semi-trailer is poorly braked compared to the haulage truck. This situation very often

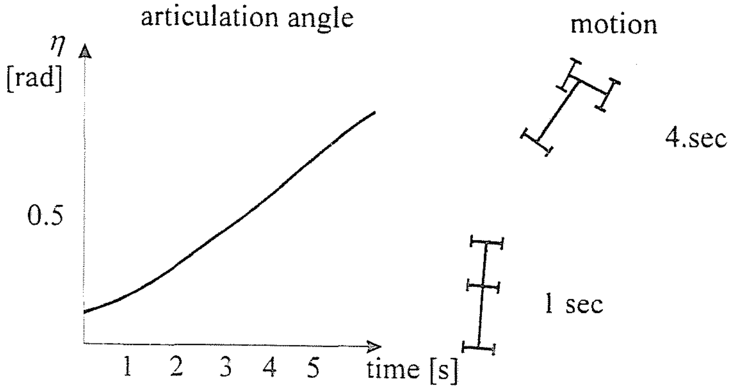

Fig. 10. Poorly braked semi-trailer

causes jack-knifing even under good driving conditions. The probability of jack-knifing grows when only the truck is equipped with ABS regulator. As shown in *Fig. 11,* significant improvement was achieved by controlling the brake signal of the semi-trailer. Although the vehicle slightly left its original, unbraked path, a proper steering action can correct this deviation.

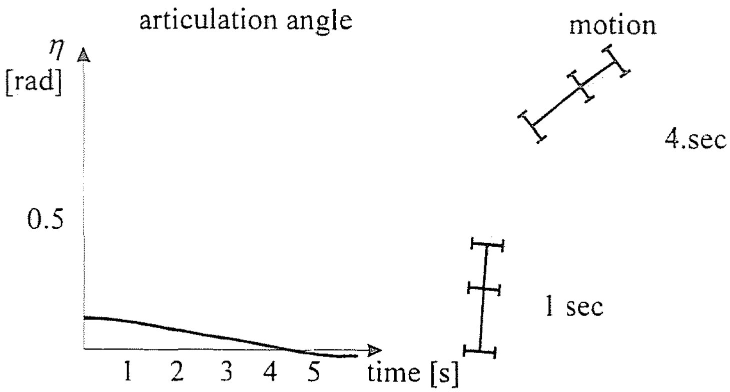

Fig. 11. Ideal braking

6. Conclusions

The results of this simulation study provide quantitative information on the effects of various adaptation of truck and semi-trailer brake system on the directional control and braking performance. They also provide an insight into the possible consequences of the failures of brake systems and anti-locking systems.

It appears that the use of anti-locking devices on all three axles of a tractor/semi-trailer is satisfactory from the standpoint of directional stability and control. In addition to the three-axle control arrangement, a two-axle control arrangement must be considered in which only the axles of haulage truck are equipped with ABS. In this case, there is a tendency of jack-knifing and semi-trailer swing, but appraising the dynamical state of the vehicle and tuning the brake system of the semi-trailer, these effects can be avoided.

As an additional benefit, the automatic control system provides a warning capability to indicate that the brakes have failed. A device should be designed that takes suitable action in case of failure and, at the same time, brings the attention of the driver to the alternative course of action through an appropriate warning signal.

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