Article citation info:
PARCZEWSKI K, WNĘK H. Using mobile scaled vehicle to investigate the truck lateral stability. Eksploatacjai Niezawodnosc – Maintenance and Reliability 2013; 15 (4): 414–420.

Krzysztof PARCZEWSKI
Henryk WNĘK

USING MOBILE SCALED VEHICLE TO INVESTIGATE THE TRUCK LATERAL STABILITY

WYKORZYSTANIE MOBILNEGO MODELU POJAZDU DO ANALIZY STATECZNOŚCI POPRZECZNEJ SAMOCHODU CIĘŻAROWEGO*

This paper presents the results of an attempt to transfer resistance to the side overturning of the vehicle to the mobile vehicle in the scale of ~ 1:5 on the real vehicle. Due to the substantial cost of testing and the danger of rollover real vehicle attempt was made to reproduce the behaviour of the vehicle, using the conditions of similarity. The paper presents methods of risk detection and control algorithms in stability systems equipped with a safety feature to prevent rollover. The analysis was based on the tests carried out at research training ground. Shows the results of tests on a real and a mobile smaller scale vehicles, and the values of obtained rollover risk indicators.

Keywords*: stability of truck movement, testing of mobile scaled vehicles, vehicle stability, testing, rollover risk indicators*

Praca przedstawia próbę przeniesienia wyników badań odporności na przewrócenie pojazdu na bok z mobilnego modelu pojazdu w skali ~1:5 na pojazd rzeczywisty. Z uwagi na znaczny koszt badań i niebezpieczeństwo przewrócenia pojazdu rzeczywistego starano się odwzorować zachowanie się pojazdu, wykorzystując warunki podobieństwa. W pracy przedstawiono sposoby detekcji zagrożenia oraz algorytmy sterowania układów stabilizacji toru jazdy wyposażonych w funkcję zabezpieczającą przed przewróceniem. Analizę przeprowadzono w oparciu o próby poligonowe. Przedstawiono wyniki badań pojazdu rzeczywistego i mobilnego modelu w mniejszej skali oraz uzyskane wartości wskaźników zagrożenia przewróceniem pojazdu.

Słowa kluczowe*: stateczność ruchu samochodu ciężarowego, badania mobilnych modeli pojazdów, badania stateczności pojazdów, wskaźniki zagrożenia wywrotem.*

1. Introduction

Rollover of the vehicle is about 2.5% of the total number of accidents, but they have about 20% of the total number of victims [5]. Such overturning of the vehicle occurs when the vehicle is rotated by ninety degrees or more relative to its longitudinal axis. Agency NHTSA (National Highway Traffic Safety Administration USA) [7] has evaluated a number of manoeuvres that may cause wheels lift on the road and rollover the vehicle. The results did not indicate in detail the manoeuvres that cause the loss of lateral stability of the vehicle. It should be noted that the study of large-scale test vehicle especially, are dangerous and costly. Attempts were made to determine the conditions of "limit" at which the vehicle is likely to rollover. To assess the vehicle stability both indicators are based on the mass and geometrical parameters of vehicles with varying degrees of simplicity and road tests are performed using standardized procedures vehicles developed by ISO and recommended by NHTSA, such as "J-turn", "fishhook" or other similar testing procedures used at various proving grounds, testing of vehicles (presented in part 3).

The aim of the experimental research was not only to determine the significance of the mass and geometrical parameters on the propensity for overturning the vehicle, but also to determine limit values of indicators of the motion condition at which the vehicle may rollover.

In the following chapters is a description of the requirements and standardized tests carried out during the test vehicles. Then is a description of selected tests and their results. The last chapter contains the analysis, conclusions and summary of the main points of work.

2. Justification for the use of vehicles at a scale to test stability

Susceptibility testing of real vehicles to overturn on its side is expensive and dangerous. Real vehicle dynamics studies are carried out on research proving grounds - separate complexes roads, implementing road tests under controlled and repeatable conditions. In the world there are dozens of research training grounds used by carmakers to test vehicles in the summer and winter conditions.

The test track should allow for a wide variety of vehicle testing, standardized or developed by the tire manufacturers, carmakers or their teams. The most common test tracks components include torah to high-speed driving, paths to test vehicle dynamics (acceleration and braking), tracks the stability and steerability test (plate with a radius of ~ 100 m), tracks the motion stability tests, the surface of different factor of friction coefficient, with varying elevation angle, tracks with different surfaces and with wavy surface. The study requires the construction of off-road vehicle tracks a number of parts allowing for testing of the capability such as wading, overcoming obstacles, moving on soft and muddy ground. Due to the ever increasing range of vehicle tests carried out test tracks, due to the safety requirements, the requirements for the same track and testing costs increase.

(*) Tekst artykułu w polskiej wersji językowej dostępny w elektronicznym wydaniu kwartalnika na stronie ***www.ein.org.pl***

Testing of vehicles on tracks research particularly defining characteristics movement stability, create rollover risk. The study of the stability of vehicles require special mounting arm, which supports the vehicle when wheels lift on from the road.

Fig. 1. The vehicle during the motion stability test

Therefore, searches are alternative methods to achieve similar results at lower cost. The use of the test vehicle on a scale of similar parameters, provides an alternative to the real vehicle test. If the scale is maintained similarity to the actual vehicle for example as the theory of Π-Buckingham, vehicle testing on a scale may replace costly and dangerous testing full-size vehicles. In addition, they provide the possibility of extending the scope of research and move closer to the conditions in which a loss of stability. The results of these tests can be used not only to evaluate the stability of a vehicle, but also for the analysis of: brake systems, suspension and vehicle protecting systems before rollover. Terms of similarity are shown by S. Brennan, S. Lapapong, S. Allyene, V. Gupta, E. Callejas, K. Romaniszyn [1, 2, 4, 5, 13] and by the authors [10, 11, 12]. Currently, there are 20 parameters compared to the size, mass and rigidity to the real and scaled vehicles.

Among the advantages of the use of vehicles on a scale motion stability experimental testing can distinguish [2, 6, 10, 12, 13, 15]:

- the cost of the scale vehicle tests is a much smaller than the full-size car, the same applies to supplies and spare parts,
- is much easier to make changes to the vehicle on a smaller scale,
- vehicle test on a smaller scale require less space (not required research track and can be done on a much smaller space),
- the possible overturning of the vehicle entails much lower repair costs and is much safer to use,
- the availability on the market of vehicles made in scale, radio-controlled or via cable, there is a wide range of models of different sizes and types that can serve as the basis for the construction of vehicles used in the tests.

3. Parameters of evaluation stability of the vehicle

The ability to maintain the desired trajectory is one of the most important aspects of automotive active safety. Every vehicle, along with his driver and their surrounding environment constitute a closed system of interaction that is unique. The task of assessing the behaviour of the vehicle stability is very difficult because of the large number of interacting components such as a driver – vehicle – trailer – shaping the way. Complete and accurate description of the behaviour of trucks with high located centre of mass must include the information received on the basis of the different types of research. Because this testing only set up a small piece of the entire field of vehicle behaviour, the results of this study can be considered to be relevant only in that area.

Experimental investigations allow an assessment of the stability and steerability of the vehicle in motion with constant and variable speed on the track straight and curvilinear, with or without taking into account the impact of the driver.

The most commonly used tests for testing the stability and steerability of vehicles includes:

- steady-state circular driving behaviour, in accordance with ISO 4138, ISO 14792 (trucks),
- double lane change manoeuvre, according to ISO 3888,
- single lane change manoeuvre, according to GOST P .2003, B32/03,
- step input manoeuvre with the linear angle escalation of the steering wheel, according to ISO 7401, ISO 14793 (trucks),
- sinusoidal input manoeuvre in the form of a one period (usually resulting in a single lane change manoeuvre), according to ISO 7401, ISO 14793 (trucks),
- continuous sinusoidal input manoeuvre, according to ISO 7401, ISO 14793 (trucks),
- pulse input manoeuvre, according to ISO 7401, ISO 14793 (trucks),
- random input manoeuvre, according to ISO 7401, ISO 14793 (trucks),
- manoeuvres developed by NHTSA: steady growth turning SIS (increasing steadily steer), "J-turn" and "fishhook".

The tests carried out for the full loaded vehicle. Height of the mass centre and weight distribution of the load should be set so as to reflect an interesting application.

The test apparatus used in the study should allow to monitor the measured values and their transcripts. The basic parameters for measuring the stability of the vehicle include: vehicle longitudinal velocity V_L, lateral speed V_Q, vehicle sideslip angle β, lateral acceleration a_y, roll angle φ and vehicle body roll rate $\dot{\phi}$, yaw speed $\dot{\psi}$, steering wheel angle δ_H. Installation of test equipment on the vehicle should be in accordance with the recommendations of the manufacturer and, if possible, provide a direct measurement. In the case of indirect measurement, perform the appropriate correction.

Conducting tests on the real vehicles is associated with a high risk of rollover. For this reason simulation studies of mobile scaled vehicles under the conditions of similarity lead to increase security and reduce the cost of research.

4. Rollover risk detection

The susceptibility of the vehicle to overturning on the side is usually determined by parameters of a vehicle, the quasi-static conditions, with various degrees of simplifying the analysis. Threshold is determined by the parameters of the vehicle rollover. The ability to rollover is determined by level of lateral acceleration, on the vehicle driven along a circular path, (assuming that the vehicle does not operate outside forces). On this basis it has been developed resulting in the definition of the threshold value of the vehicle rollover in steady state conditions in the circle path: *SSRT* (*static roll stability threshold*) – defined as the maximum value of lateral acceleration at which there will be no rollover of the vehicle.

Based on this definition resulted in different indicators of stability, of which the simplest are the *SSF*, *TTR*, RI_B, RT_{SVM}, *DSI* and others. Criteria for assessing risk of rollover of the vehicle is provided in [11, 16].

However, during the real manoeuvres of the vehicle, a relatively rare set of motion conditions. Hence, there are some stretches of time in which the lateral acceleration threshold is reached, it does not mean that there will be a rollover. Since the acceleration of *SSTR* limited top range of stability of the vehicle, it is also limited at the bottom, to determine the extent to which it is possible to lose stability.

DRT (*dynamic roll stability*) – defined as the minimum peak lateral acceleration at which rollover occurs, while performing various manoeuvres of the vehicle (which can cause it to overturn). Figure 2 shows the dependence of the energy required for of the vehicle rollover as a function of lateral acceleration.

Point 0 in Figure 2 corresponds to driving on a straight road, the potential energy increases proportionally to the lateral acceleration until it reaches the point 1 – which corresponds to the lift on of one wheel on the road. Further increase in lateral acceleration causes the increase in potential energy to reach point 2 – equivalent to lift on of the other wheels (one side of the vehicle) on the road. At this point is reached the lateral acceleration (SSRT) necessary for vehicle rollover. Obtaining greater lateral acceleration results in a loss of stability of the vehicle. Similarly, the rollover occurs, even if the lateral acceleration is smaller, but the potential energy will increase to a point 3 (the point at which the vehicle rollover occurs, although there is no lateral acceleration).

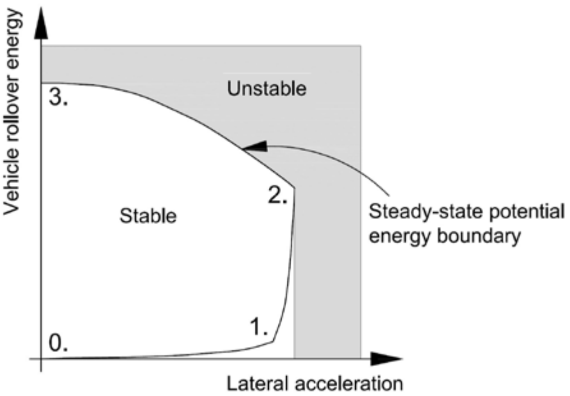

Fig. 2. The energy of the vehicle rollover as a function of the lateral acceleration [2]

It is assumed that the state of emergency occurs when it comes to interior wheels lift on of the of the vehicle from the road surface [2]. Detection of threat vehicle overturning is particularly important in the case of vehicle stability control systems is equipped with anti-roll function. These systems are usually activated only if necessary, adjust the track, the rest of the time do not affect on the vehicle performance. During rollover, the vehicle relatively quickly becomes unstable, which means that the rollover risk detection algorithm must be very sensitive, and stabilization of the system controller must be activated as soon as possible. As criteria values are used: the roll angle and roll rate (ϕ and $\dot{\phi}$), changing on one wheels axle loads (*LTR – load transfer ratio*), the critical energy rollover ($E_{critical}$) defined as the smallest energy required to lift on the wheel from the roadway and lateral acceleration limit value $a_{ycritical}$.

Below are shown a number of methods used to detect threat of the vehicle rollover, which can be used in the control algorithms in stability control systems. When choosing a method to take into account not only its effectiveness, but also the availability of the information needed to use in the drivers. These algorithms are based on parameters such as changing a single load wheel or wheels of one side of the vehicle, or the lateral acceleration acting on the center of mass and energy of the vehicle rollover. These methods can be divided into analyzing the causes and effects, resulting in a danger of the vehicle rollover. The method of using such a method causes a lateral acceleration and analyzing its derivative (dash). Analysis methods based on the effects of there are, methods based on the determination of the angle of the rolling, rollover energy, or analyzing the normal force acting on each wheel of the vehicle. It should be noted that to analysis can be used sensors of lateral acceleration in existing vehicle stability control systems. In the case of methods based on an analysis of wheel load deflection can use the parameters of the individual suspensions, which already requires the use of additional sensors. Methods based on estimating energy require an rollover vehicle parameters such as roll stiffness of the suspension, the suspension angular damping, the weight of the vehicle and traffic parameters like: the roll angle and roll ratio and others.

4.1. Changing the wheel loads

Changing the wheel load is an important indicator used in the analysis of the vehicle rollover. Side changing the wheel loads determines the change of normal forces acting on the wheel, caused by the lateral acceleration of the center of mass and its lateral shift in Y direction, due to suspension deflection. Figure 3 shows the impact of the phenomenon of shifting the center of mass to the suspension deflection.

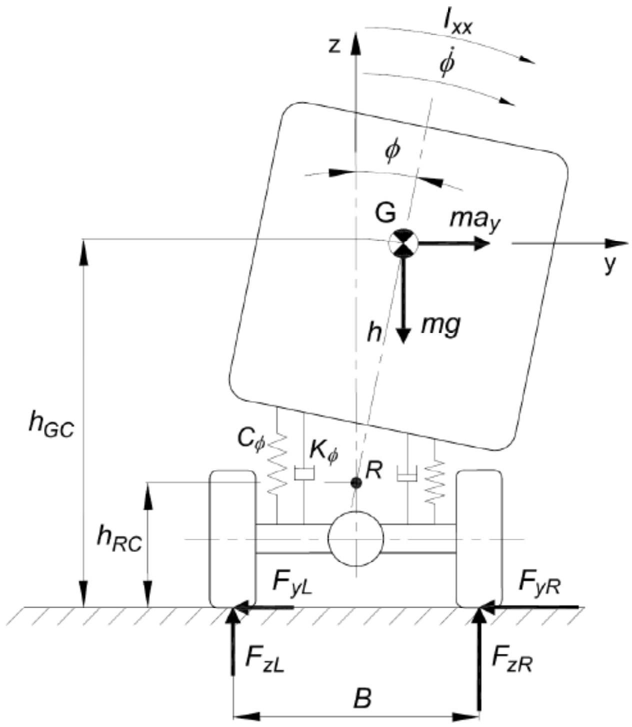

Fig. 3. Changing the wheel loads under the action of lateral force (shown in the transverse plan view)

Rate of change of wheel loads, (*LTR*) is defined as the ratio of the difference between the normal forces of the right and left side of the vehicle to their sum.

$$LTR = \frac{F_{ZR} - F_{ZL}}{F_{ZR} + F_{ZL}} \tag{4.1}$$

In the steady state, where $LTR = \pm 1$ wheels lose contact with the ground, which is read as a critical situation and can be used to control the anti-roll system of the vehicle. Under the unstable conditions the *LTR* limit should be lower.

4.2. Roll angle and roll rate

If the vehicle is equipped with sensors to measure the roll angle and roll rate (ϕ and $\dot{\phi}$), rollover threat detection can be performed with a simple analysis of these parameters. The simplest way to analyze this term limit roll angle $\phi_{critical}$ at which the driver, the fulfilment of the condition $|\phi| > \phi_{critical}$, starts to work.

As in the previous solution may be used to control the roll rate. In this case, the driver can be activated after fulfilling both conditions: $|\phi| > \phi_{critical}$ and $\dot{\phi} \cdot sign(\phi) > 0$.

4.3. Methods based on the energy

Overthrow threat detection and activation of the anti-roll driver can be implemented based on the energy of the vehicle rollover. “Emergency” is defined as when the inner wheel is lift on from the substrate. This energy consists of the potential energy accumulated in the suspension deflection and kinetic energy. Therefore, it is determined by the relationship:

$$E = \frac{1}{2} \cdot C_\phi \cdot \phi^2 - m \cdot g \cdot h \cdot (1 - \cos\phi) + \frac{1}{2} \cdot (I_{xx} + m \cdot h^2) \cdot \dot{\phi}^2 \quad (4.2)$$

The critical value of the rollover energy Ecritical can be defined as the minimum energy required to lift on the wheels of one side of the vehicle from the road. To rollover the vehicle, the total torque against roll axis, for the centre of mass motion, must be greater than the moment caused by the normal wheels force remain in contact with the road. Critical situation can be defined by the inequality moments acting on the vehicle:

$$\frac{1}{2} F_z \cdot B < F_y \cdot h_{GC} + C_\phi \cdot \phi + K_\phi \cdot \dot{\phi} \quad (4.3)$$

The critical value of the rollover energy Ecritical is determined by minimizing against roll angle ϕ and roll rate $\dot{\phi}$.

4.4. Methods based on the analysis of lateral acceleration

Vehicle roll stability analysis, based on the forces of inertia d’Alembert ($-m \cdot a_y$) acting on the centre of mass and the causing increase in the overturning moment. On vehicles equipped with stability control system lateral acceleration is measured and used it as a pointer indicating to the threat of the vehicle overturning, it becomes very attractive. Taking into account the impact of the suspension deflection complicates the analysis, so the number of solutions it has been ignored, resulting in low values of lateral acceleration limit. For a more detailed analysis allows to determine the value of the derivative of lateral acceleration (spurt). An additional complication is the fact that the measurement of acceleration has a significant noise and its elimination requires additional treatments.

4.5. System control algorithms prevent vehicle roll

Typically were used two control algorithms. One is based directly on the limit of the selected index: *LTR* wheel load changes, the rolling angle ϕ, lateral acceleration a_y or rollover energy E stored in general as $R_{critical}$ (labelled as $\hat{R}$). Dynamic switching control strategy (the second control algorithm) is based on a derivative of the ratio *R*. The idea is that the controller is operating in full if $R > \hat{R}$ and when it derivative rises $\dot{R} \cdot sign(R) > 0$ and its works only partially, when the derivative decreases $\dot{R} \cdot sign(R) \le 0$.

The first algorithm was written in the form:

$$F_{xT} = \begin{cases} 0 \quad dla \quad |R| \le \hat{R} \\ -m \cdot a_{x\max} \quad dla \quad |R| > \hat{R} \end{cases} \quad (4.4)$$

where a_y max – maximum attainable braking deceleration.

In the second case, the algorithm was described in the formula:

$$F_{xT} = \begin{cases} 0 \quad dla \quad |R| \le \hat{R} \\ -m \cdot a_{x\max} \quad dla \quad |R| > \hat{R} \cup \dot{R} \cdot sign(R) > 0 \\ -\frac{|R| - \hat{R}}{R_{\max} - \hat{R}} \cdot m \cdot a_{x\max} \quad dla \quad |R| > \hat{R} \cup \dot{R} \cdot sign(R) < 0 \end{cases} \quad (4.5)$$

These algorithms can also be used to adjust the vehicle wheel steering angle [9].

$$\delta_R = \begin{cases} 0 \quad dla \quad |R| \le \hat{R} \\ k_R \cdot sign(R) \cdot (|R| - \hat{R}) \quad dla \quad |R| > \hat{R} \end{cases} \quad (4.6)$$

where k_R – correction factor of wheel steering angle.

4.6. The limit values of indicators

Correct operation of the drivers requires an estimate indication limits for their activation.

Changing a wheel load of the vehicle LTR

The first is a change of the wheel loads LTR. This indicator varies in the range from 0 to 1, the value 1 is obtained at the time of lift on the wheels from the road. In the most general form of this relationship is as follows [9]:

$$LTR_{critical} = \frac{F_{ZR} - F_{ZL}}{F_{ZR} + F_{ZL}} = \frac{2 \cdot m_s}{m \cdot B} \cdot \left[((h_{GC} - h_{RC}) + h_{GC} \cdot \cos\phi) \cdot \frac{a_y}{g} + h_{GC} \cdot \sin\phi \right] \quad (4.7)$$

In the next part shows the values of R obtained from the road tests of scaled and the normal size vehicle.

The roll angle

Another indicator is based on vehicle roll angle ϕ. It can be relatively easy to determine in the steady state conditions, with the formula:

$$\phi = \frac{mg \cdot (h_{GC} - h_{RC})}{K_\phi - mg \cdot (h_{GC} - h_{RC})} \cdot \frac{a_y}{g} \quad (4.8)$$

In a dynamic dependence on roll angle is much more complicated. It is therefore recommended that you use the real roll angle obtained from measurements. Typically, the limit value of the roll angle, there are 5 ÷ 7 degrees.

Energy rollover

Index based on the analysis of rollover energy is used to determine the normalized condition of the vehicle rollover. It has been proposed by Johansson and Gäfvert [3], and is defined by the relationship:

$$ROW_1 = 1 - \frac{E_{critical} - E}{E_{critical}} \quad (4.9)$$

The critical situation is achieved for $ROW_1 \ge 1$. The inclusion of the controller is done when it reaches the limit value of ROW_1 less than 1, this value can be determined experimentally. This implies that the controller should begin operation after exceeding the limit value of indicator $ROW_1 \le ROW_{1\text{limit}}$. Attempt to estimate this parameter is shown below.

In the references, most of the materials can be found on the lateral acceleration limits. Figure 4 shows the recommendations proposed by the NTRCI (National Transportation Research Center) [8]. Limit value of lateral acceleration for a truck with a high center of gravity, are dependent on the suspension and tires stiffness, and the car body treated as a rigid amount of 0.50 g, and for a car after taking into account the susceptibility of the suspension and tires −0.35 g.

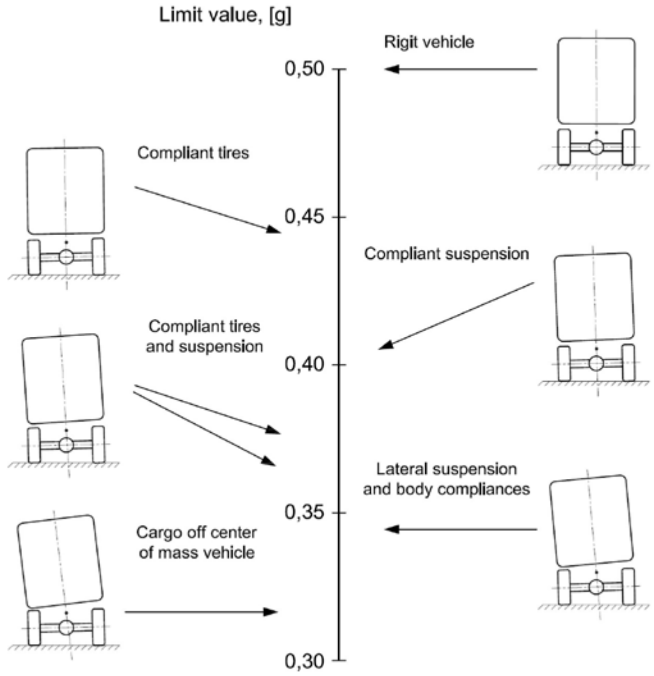

Fig. 4. Limit values threat of rollover in the steady conditions [14]

Due to the fact that the vehicle height of center of gravity, rigidity of the suspensions and their characteristics and the using of stabilizers, considerably influences on the lateral acceleration limit value. In the analysis of vehicles with other characteristics parameters must be made appropriate adjustments. In further analysis includes the impact of these factors on the limit rate values.

5. Investigation of the scaled and normal size vehicle

5.1. The scaled vehicle

For calibration the dynamics of real vehicle motion is used for radio-controlled model car on a scale ~1:5. The scaled vehicle is equipped with an internal combustion engine of a cylinder capacity 26 ccm, centrifugal clutch, gearbox, center and main gearbox, and rear-wheel drive. In order to maintain the conditions of similarity in relation to the real vehicle's (special truck with a high located center of mass), a number of modifications that resulted among other changes: the wheelbase, center of mass, mass moments of inertia, suspension design front and rear axle and tire parameters. Scaled vehicle is equipped with appropriate measurement equipment allows the measurement and recording of relevant parameters of its motion. It was decided to use measuring equipment from Racelogic company – VBOX with the IMU module. Record the results of measurements were made on the Compact Flash memory card. Figure 5 shows the scaled vehicle with a installed measuring apparatus.

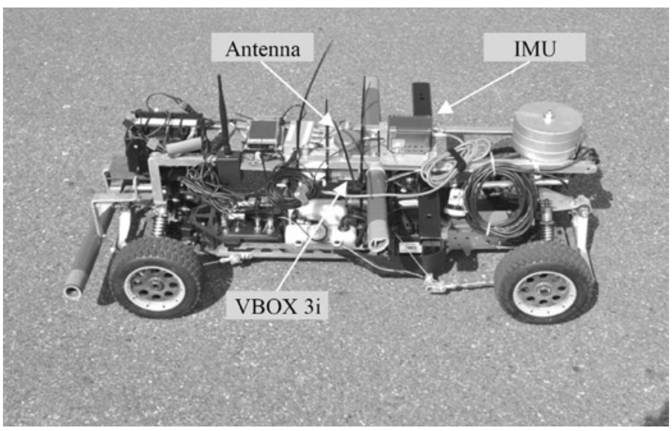

Fig. 5. Scaled vehicle with mounted the measuring apparatus

5.2. Field tests

For comparisons of selected two trials: driving in a circle with a fixed speed and manoeuvre extortion jump of the linear angle escalation of the steering wheel. Tests were carried out on the test track TATRA in Koprivnice (Czech Republic) in the case of a vehicle full scale [11, 12, 13] and at the airport in Kaniow near Czechowice-Dziedzice for scaled vehicle.

Figure 6 shows the path of the test drive in a circle and method of implementation the test of step input with linear angle escalation of the steering wheel.

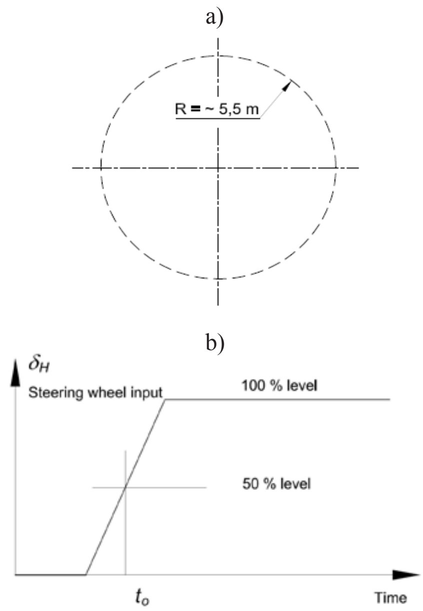

Fig. 6. Carried out tests: driving in a circle at a constant speed in a steady state conditions (a) and step input of the steering wheel (b)

5.3. Driving in a circle with a fixed speed

The trial was carried out at with velocity ~17 km/h and lateral acceleration ~4.5 m/s^2 (corresponding to real vehicle – speed ~40 km/h moving on the track with a radius of ~21.5 m). Figure 7 shows the course of the selected indicator and its derivative.

The graph shows that the dynamic component of the R ratio related to the derivative of the test under steady state conditions is rather small, and the components associated with the indicator has a value oscillating around the limit value (depending on equations (4.5) and (4.6)). In the case presented above may lead to activation of the vehicle stability control system.

5.4. Step input on the steering wheel

During the tests vehicle moved at a fixed speed on the straight, and then, in a designated area perform a rapid turn for a fixed steering wheel angle.

During the maneuver the derivative ratio $\dot{R}$ is significantly increased in the first part of the maneuver and then decreases. Indicator R

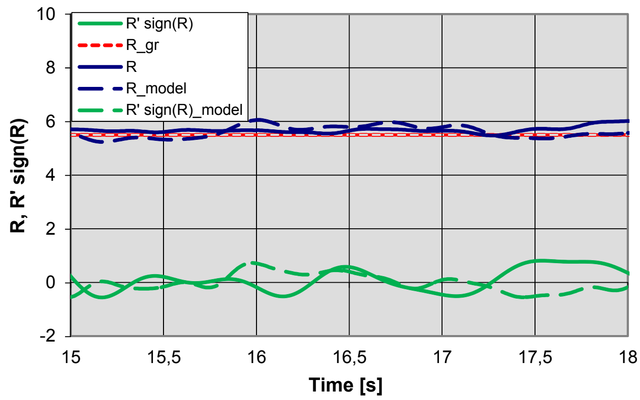

Fig. 7. The course of the selected indicator during the test circular driving at constant speed (the real vehicle marked with a continuous line, dashed line scaled vehicle)

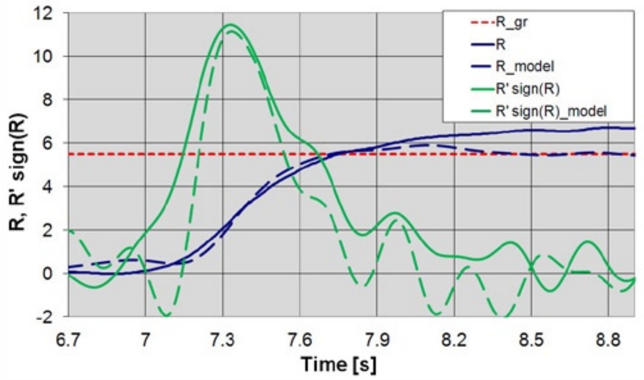

Fig. 8. The course selected indicator stepping while step input on the steering wheel (the real vehicle marked with a continuous line, dashed line scaled vehicle)

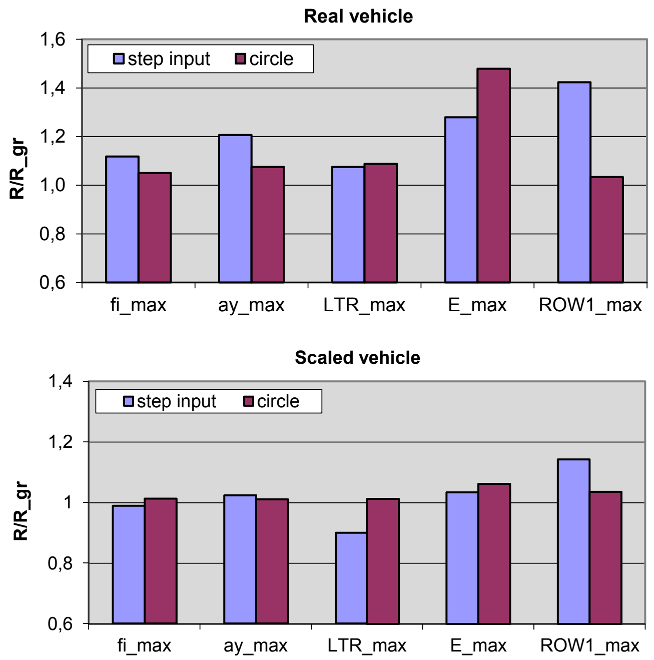

*Fig. 9. Normalized values of threat indicators of vehicle rollover R/R^gr^*

ratio increases, but the increase is delayed in relation to the derivative of ~0.6 ÷ 0.8 sec. This time delay associated with the inertia of the vehicle, it is often referred to as a time to lift on the wheel (figure 8).

Analysis of different risk rollover indicators R has clarified differences between the various indicators and their derivatives for real and scaled vehicle. In Figures 9 and 10 shows the values of the indicators and their derivatives.

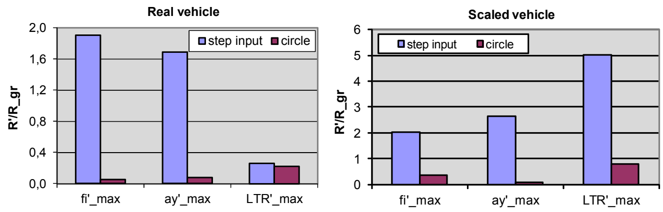

Fig. 10. Normalized values of vehicle rollover derivative risk indicators $\dot{R} / R_{gr}$

The comparison shows that both the real and the scaled vehicle have similar values of R for all parameters compared ϕ, a_y, LTR, E and ROW_1. Differences obtained for the test fixed driving in a circle, are respectively, 11, 15, 16 and 19% and for the jumping extortion on the steering wheel the differences are smaller, and are approximately 4, 6, 7 and 28%. Slightly larger differences have derived indicators $\dot{\phi}$, $\dot{a}_y$, and especially large differences are observed at a ratio of wheel unloading one side of the vehicle $L\dot{T}R$.

6. Summary and Conclusions

Comparison of the results of simulations of the scaled vehicle with the results of real vehicle measurements show good agreement of the analyzed parameters. Generally, this allows to conclude that the mobile scaled vehicle can be used to determine the control parameters for vehicles equipped with stability control systems with function of anti-rollover. There is a good agreement for the parameters of the roll angle, lateral acceleration and wheel unloading one side of the vehicle. Larger differences indicate: rollover energy and ROW1 rate. Similar ratios derived values obtained for the roll angle and lateral acceleration. Large differences exist in the comparison rate derivative LTR.

Based on the presented simulation tests of scaled and real vehicle there are the following conclusions:

- good agreement was obtained for parameters characterizing the risk of the vehicle rollover, both tests: driving in a circle with a fixed speed and step input of the steering wheel confirmed this compliance
- scaled vehicle can be used to create software systems for vehicle stability control systems equipped with anti-roll function to determine the limits of indicators characterizing the risk of rollover,
- the further testing of the scaled vehicle and work to preserve similarities more compared parameters to the real vehicle, and should contribute to reducing differences in the investigated indicators.

Work was established during the project realization R & D NCBR no. PB 5478/B/T02/2011/40 „Evaluation of the stability of the real vehicle on the basis of the scaled vehicle."

References

1. Brennan S, Alleyne A. The Illinois Roadway Simulator: A mechatronic testbed for vehicle dynamics and control. IEEE/ASME Transactions on Mechatronics 2000; 12, Vol. 5, No. 4: 349–359.
2. Dahleberg E. Commercial vehicle stability – focusing on rollover. Stockholm: Vehicle Dynamics, 2001.
3. Johansson B, Gäfvert M. Untripped SUV rollover detection and prevention. 43rd IEEE Conference on Decision and Control 2004; 14-17.
4. Lapapong S, Gupta V, Callejas E, Brennan S. Fidelity of using scaled vehicles for chassis dynamic studies. Vehicle System Dynamics 2009; 11 (47): 1401–1437.
5. Lapapong S. Vehicle rollover prediction for banked surfaces. Pennsylvania State University 2010.
6. Lozia Z. Ocena odporności pojazdu na przewrócenie na bok – wpływ stopnia skomplikowania modelu na wyniki obliczeń. Zeszyty Naukowe Instytutu Pojazdów Politechnika Warszawska 2010; 3(79).
7. NHTSA, Traffic Safety Facts 2003 – Final Report. U.S. Department of Transportation: National Highway Traffic and Safety Board 2004.
8. Pape D, Arant M, Nelson S, Franzese O, Knee H, LaClair T, Attanayake U, Hathaway R, Keil M, Ro K. Heavy truck rollover characterization (Phase B). NTRCI 2009.
9. Odenthal D, Bünte T, Ackermann J. Nonlinear steering and braking control for vehicle rollover avoidance. Proceedings of European Control Conference, Karlsruhe, Germany 1999.
10. Parczewski K, Wnęk H. Utilization of the car model to the analysis of the vehicle movement after the curvilinear truck. Eksploatacja i Niezawodnosc – Maintenance and Reliability 2010; 4 (48): 37–46.
11. Parczewski K, Wnęk H. Wykorzystanie kryteriów podobieństwa do analiz stateczności ruchu na podstawie mobilnego modelu samochodu ciężarowego. Logistyka 2012; 3.
12. Parczewski K, Wnęk H. Analiza wpływu parametrów masowych na stateczność ruchu samochodu ciężarowego w oparciu o badania mobilnego modelu pojazdu. Postępy Nauki i Techniki, SIMP 2012; 14: 208–223.
13. Romaniszyn KM. Mobilne modele samochodów do badań stateczności. Logistyka 2012; 3.
14. Winkler C. Rollover of Heavy Commercial Vehicles, University of Michigan Transportation Research Institute, Research Review Vol. 31 No. 4
15. Yih P. Radio controlled car model as a vehicle dynamics test bed. Mechanical Engineering Department. Stanford University 2000.
16. Yu H. Guvenc L. Ozguner U. Heavy-duty vehicle rollover detection and active roll control. Vehicle System Dynamics 2008; 6 (46): 451–470.

Key symbols

a_x	Longitudinal acceleration (along X axis)
a_y	lateral acceleration (along Y axis)
B	Wheel track
C_ϕ	Vehicle roll stiffness coefficient
$E_{critical}$	Vehicle rollover energy
F_{xT}	Braking force
F_y	Force acting respect to the Y axis
$F_{yL,R}$	Lateral force acting on left / right wheel
F_z	Force acting respect to the Z axis
$F_{zL,R}$	Normal force acting on left / right wheel
g	Gravitational acceleration
h	Distance between gravity centre and roll axis $h = h_{GC} - h_{RC}$
h_{GC}	Gravity centre height
h_{RC}	Roll centre height
I_{XX}	Vehicle mass moment of inertia respect to the X axis
K_ϕ	Vehicle roll damping coefficient
m	Mass of the vehicle
ms	Vehicle spring mass
δ_R	Wheel steering angle
δ_H	Steering wheel angle
ϕ	Vehicle roll angle
$\dot{\phi}$	Vehicle roll rate respect to the X axis

Krzysztof PARCZEWSKI, Ph.D. (Eng.)
Henryk WNĘK, Ph.D. (Eng.)
University of Bielsko-Biala,
Department of Combustion Engines and Vehicles,
Willowa 2, PL – 43 309 Bielsko-Biala, Poland
E-mails: kparczewski@ath.bielsko.pl, hwnek@ath.bielsko.pl