Article citation info:
Dusza M. Freight car model with Y25 bogies stability analysis. Rail Vehicles/Pojazdy Szynowe. 2023;3-4:13-24. https://doi.org/10.53502/RAIL-175725

Rail Vehicles/Pojazdy Szynowe

www.railvehicles.eu

Freight car model with Y25 bogies stability analysis

Mirosław **Dusza** [a, *]

[a] *Faculty of Transport, Warsaw University of Technology, Poland*

ARTICLE INFO

Received: 10 October 2023
Revised: 22 November 2023
Accepted: 22 November 2023
Available online: 27 November 2023

KEYWORDS

Y25 bogie
Lateral stability
Nonlinear critical velocity
Suspension
Derailment quotient

Y25 (25Tn) bogies have been used in freight cars for over 50 years. It is a proven design with a positive opinion in many European countries. In addition to the generally positive opinion about the operational properties of the Y25 bogies, there are sometimes publications whose authors notice certain imperfections related mainly to the so-called running instability, resulting in derailment under certain motion conditions. This paper presents results simulation of motion tests of a freight car with Y25 bogies. A freight car model was created using the VI-Rail software. The research focus on determining the critical velocity and examining nature of the model solutions in the overcritical velocity range. Straight track as well as curved track motion is analysed. Empty and full wagon properties are considered. Impact of changes the suspension system selected parameters values on vehicle model solutions stability as well as the vehicle impact on track was analysed.

1. Introduction

Systematic improvement of the efficiency of rail transport concerns mainly passenger vehicles. Velocity of motion is increases, reliability of control systems is increases too, safety and comfort of traveling are improved. Development works concern both, vehicles and the entire track infrastructure [9]. Following the development of passenger rail vehicles, freight vehicles are also improved. Freight vehicles most often use the same track infrastructure as passenger vehicles. There are many features that distinguish the construction of passenger and freight vehicles. Generally cargo vehicles have a simpler design than passengers one. The advantage of design simplifications is ability to reduce cost of vehicle manufacturing and achieve high reliability. Disadvantage, however, are limited possibilities of intervention aimed at improving the vehicle properties while maintaining its basic structure. Typical structure of a modern rail vehicle consists of bogies and a body. These are unified constructions usually, which facilitates vehicle servicing. The basic elements of the bogie structure are: frame, wheelsets, axle boxes, suspension and brake equipment. Many components of the bogies are most often manufactured by specialized companies and assembled in the factory producing the wagons. Suspension elements are also among them. A metal spring elements (coil or leaf springs) are used in freight vehicles usually. Friction dampers with dry friction are most often used in order to damp vibrations.

Freight car (gondola car tape) with two Y25 bogies (Fig. 1) is analysed in presented article. Manufacturers of these bogies are scattered in many places, so there are some differences in construction details. However, the basic structural arrangement is fixed [3]. Characteristic features of this bogie are: wheelbase – 1.8 m, bilinear characteristic of the first suspension stage, the Lenoir type suspension on the first stage with friction dampers, where damping force depending on the wagon load. Two assemblies of coil springs placed vertically between each axle box and bogie frame provide substantially vertical stiffness (Fig. 1b). This is due to the basic load of the vehicle and its own mass (gravity force). In addition to gravitational forces, perpendicularly to the vertical direction acted forces may appear during the vehicle motion. It means the longitudinally and laterally acted forces have to be carried by the same suspension elements. The design of wheelsets guiding allows lateral displacements of

* Corresponding author: miroslaw.dusza@pw.edu.pl) (M. Dusza)

the wheelsets with axle boxes in the range of ±10 mm versus bogie frame. In this range of displacements, the stiffness in lateral direction is provided by the same sets of vertical coil springs. There is therefore a structural relationship between vertical and lateral stiffness.

Lack of separate springs and damping elements in directions perpendicular to the vertical is a significant simplification of the freight vehicle bogie structure. However, it forces vertical elements to provide (at least within a limited range of displacements) stiffness and damping in the longitudinal and lateral directions. The analysed Y25 bogie has axleboxes located by horn guides [6], so the suspension longitudinal deflections are strictly limited. Then, the vertical springs and dampers have to act in vertical and lateral directions. According to some users opinion [3], vehicles with Y25 bogies has got two disadvantageous features: not achieved excellent running quality and its running stability is not satisfactory. Presented research carried out on vehicle model with Y25 bogies, contribute to works aimed at obtaining information on how to improve the stability of vehicles in the operating velocity range (up to 120 km/h), without interfering with the basic Y25 bogie structure, and to do it with a small expenditure.

Numerical model of the gondola car with Y25 bogies (Fig. 1) was prepared. The modeled vehicle dynamic was analysed by means of motion simulation. The research focuses on determining the critical velocity and the nature of solutions in the overcritical velocity range [4, 14–16]. First suspension stage vertical and lateral stiffness were separated in the created wagon model for research purposes. With a constant (nominal) vertical stiffness value, the lateral stiffness was changed. It was noticed that it is possible to significantly change the dynamic properties of the Y25 bogie by changing the lateral spring stiffness while maintaining a constant vertical stiffness value. Changes of the lateral stiffness influence on the vehicle-track forces and derailment quotient was also analysed.

2. The model

Vehicle-track model was created with the VI-Rail engineering software (Fig. 2). This is a discrete model of 412W Eaos cargo gondola car type (see Fig. 1). Bogie models are based on a Y25 (25TN in polish stand) construction. The complete wagon model is composed of 15 rigid bodies: loading platform (gondola), two bogie frames, four wheelsets and eight axle-boxes. Rigid bodies are connected with elastic-dumping elements having linear and bi-linear characteristics. Dry friction dampers are applied in the Y25 suspension structure. Suspension system parts with dry friction of Coulomb type are difficult to model. Difficulties results from the non-smooth, multi-valued, non-differentiable characteristic of Coulomb function during slip-stick transition [2, 11]. Regularisation methods proposed to solve this problem [7] leads to a long execution time during numerical simulation usually. In addition each dry friction damper acts in vertical as well as lateral direction in real Y25 suspension system. So, to precisely describe the friction forces, two-dimensional dry friction model should be adopted [11]. In order to avoid above mentioned problems and identify influence of each suspension parameters on vehicle properties, model is simplified. It means, two separate viscous dampers modeled with a big gradient function in vertical c_{zz} and lateral c_{zy} directions are applied. The sets of vertical coil springs on each axle box, execute the vertical k_{zz} and lateral k_{zy} stiffness. Each of the above mentioned parameters assume two values – one for empty the second for laden vehicle state (see Appendix). The wheelset's longitudinal guidance is very stiff. As it was mentioned, it is a kind of horn guides [6], so the longitudinal stiffness k_{zx} result from guides material deflection only and do not depend on vehicle load.

a)

b)

Fig. 1. The modelled vehicle (a) and Y25 bogie (b) side views

The coach model is supplemented by vertically and laterally flexible track model. Track parameters corresponding to European ballasted track of 1435 mm gauge were adopted. Nominal profiles of S1002 wheels and UIC60 rails with the inclination 1:40 were used. Track irregularities are not taken into account. Wheel-rail coefficient of friction 0.4 is applied. The non-linear contact parameters are calculated with the ArgeCare RSGEO software. For calculating tangential wheel-rail contact forces, simplified Kalker theory implemented as the FASTSIM procedure is used [8, 12]. Equations of motion are integrated with use of the Gear procedure with step-size and error control. It is well suited for solution of stiff problems such as the rail vehicle dynamic numerical analysis [6, 13]. Complete vehicle-track system has 76 kinematic degrees of freedom. Detailed values of the model parameters applied can be found in Appendix.

3. The method of research

The method of research consists in analysing model solutions for constant vehicle velocity value. The self-exciting vibrations theory is used to explain of wheelset oscillations (hunting motion). Vehicle velocity range starts from e.g. 10 m/s and ends with maximum value, for which stable model solutions can be observed. The observed parameters are usually the first wheelset lateral displacements y_{lw} as a function of distance or time. The „stable solution" term was adopted to describe the model solutions in which y_{lw} has a constant or periodic value of the limit cycle nature. Other forms of solutions are described as unstable, although they may meet the criteria of example technical stability [1]. Such criteria were also adopted in the analyses presented in this article.

Figure 3 presents examples of stable solutions typical for vehicle motion velocity lower than critical ($v < v_n$, Fig. 3a), and higher than critical value ($v > v_n$, Fig. 3b). In this example, the vehicle model motion takes place on a track consisting of straight section, transition curve and circular arc with a radius of R = 2000 m. The wheelset is shifted laterally during transition curve negotiation and $y_{lw} \neq 0$ in regular arc section due to track superelevation appliance. Values of the superelevation in particular track curves are collected in Table 1.

Table 1. Curve radii tested and track superelevation corresponding to them

Curve radius; R [m]	1200	2000	3000	4000	6000	∞
Superelevation; h [m]	0.150	0.130	0.110	0.077	0.051	0

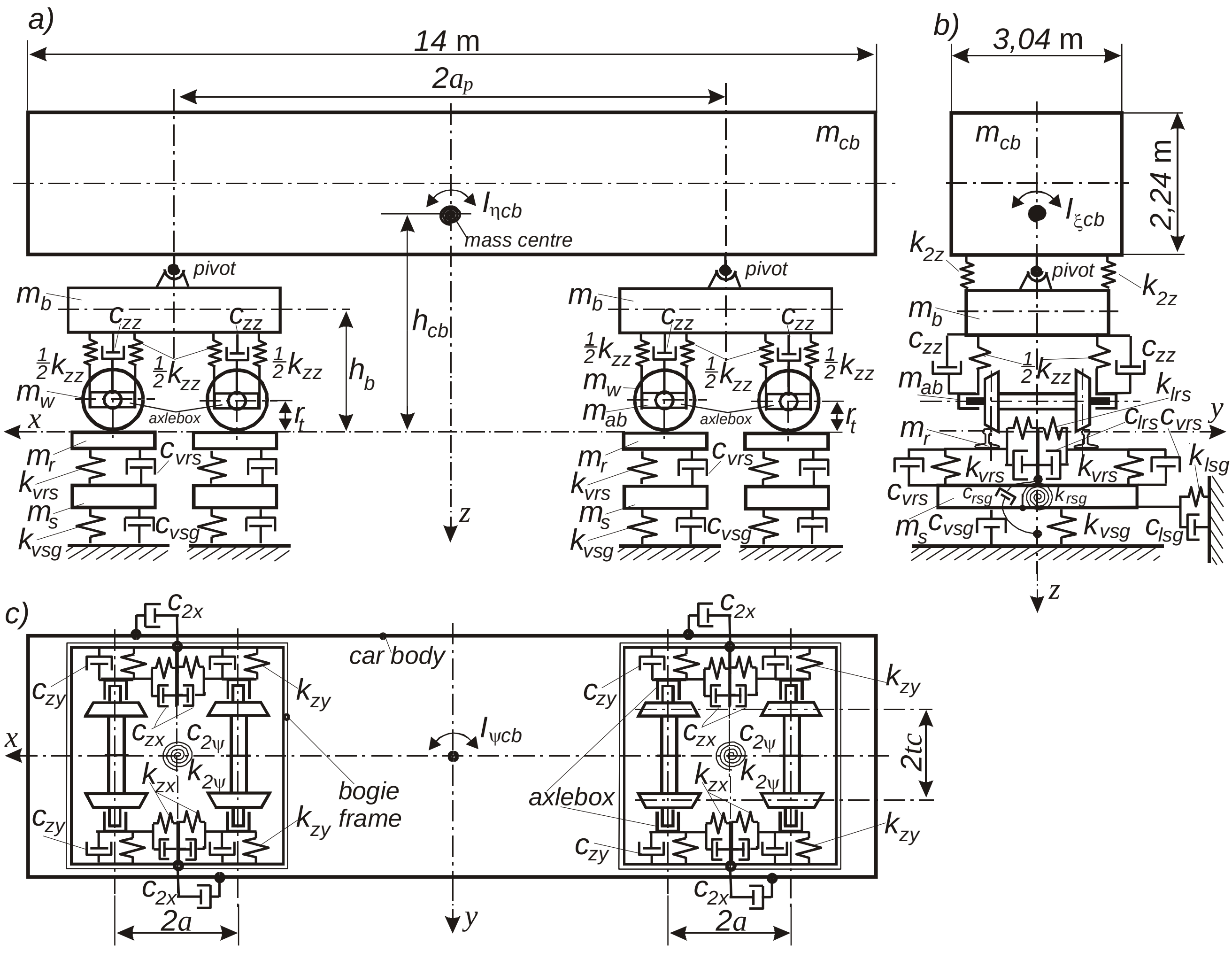

Fig. 2. Diagram of the freight vehicle – track system: a) side view, b) front view, c) top view

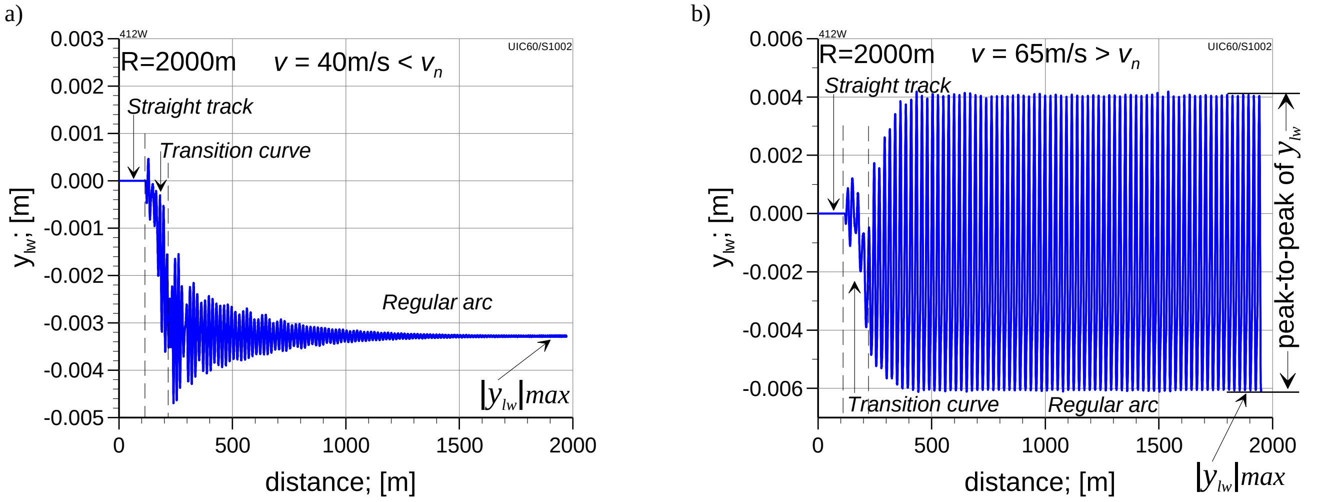

Fig. 3. The first wheelset lateral displacements y_{lw} versus distance: a) stable stationary solutions (40 m/s < v_n), b) stable periodic solutions (limit cycle, 65 m/s > v_n)

It was demonstrated in [14] and in these research that wheelset's oscillations can be described as a limit cycle of hard excitation. It means some minimum values of initial conditions are necessary to initiate the periodic solutions (wheelset vibrations). The transition curve negotiation causes the wheelsets lateral shift and fall out of equilibrium (central position in track). In model studies, this is a way of impose initial conditions to the curved track solutions. In case of straight track analysis, a single track lateral irregularity is located 100 m from the track beginning. All wheelsets are shifted laterally about 0.005…0.006 m as a result of the irregularity negotiation. The first bogie's leading wheelset lateral displacements maximum value y_{lw} max is read from the final section of each simulation diagram (example Fig. 3).

Depending on curve direction and adopted orientation of the coordinate systems the y_{lw}max may be positive or negative (in Fig. 3). The negative value correspond to curves turning to left hand side. The solutions for curves turning to left- and right-hand side are antisymmetric to each other for the same conditions of motion. So to avoid sign problem the maximum absolute value of wheelset lateral displacements $|y_{lw}|$max is accepted as the result of solutions for a given vehicle velocity. If the solution is stationary (Fig. 3a), $|y_{lw}|$max is the only solution value read. If the solution is periodic (Fig. 3b), peak-to-peak of y_{lw} values are also read off and record. Peak-to-peak of y_{lw} = 0 for stationary solutions. Thus, $|y_{lw}|$max and peak-to-peak of y_{lw} are read from each simulation of motion performed for a constant velocity and presented in form of the so-called „stability maps". Stability maps are a pair of diagrams showing $|y_{lw}|$max and peak-to-peak of y_{lw} (p-t-p of y_{lw} in short) as a function of vehicle velocity for curved tracks of different radius R value (example Fig. 4). The diagrams enable to observe on them: nature of the solutions, values of solutions and critical velocity v_n value in the range of velocities, for which stable solutions exists. If there are stationary solutions only in the range of velocity below the critical value v_n, the presentation of results starts with velocity slightly below the critical v_n (30 m/s in these research usually) and covers the over critical velocity range. Lines on the diagrams refer to particular specific route – it is curved track with a constant radius R (denoted by different line colors). The last point of each line corresponds to the velocity, for which a stable solution (stationary or periodic) was obtained. Detailed description the method of research vehicle stability can be found in publications [4, 14–16]. It should be mentioned that stable stationary solutions (Fig. 3a) are identify with constant wheelset position in track and expected in normal vehicle operation conditions. The periodic solutions are identify with wheelset self-exciting vibrations appearance. These vibrations accompanying the main vehicle motion (along track), increase derailment risk [10] and are strictly undesirable in normal vehicle operation conditions.

4. Results of the research

The first stage of research consisted in preparing the so-called stability maps. As it turned out, the empty wagon model did not show any features requiring changes (Fig. 4). The lowest value of critical velocity occurred on a straight track and was 37.2 m/s (Table 2). It is therefore sufficiently greater than the maximum operating velocity of the bogie 120 km/h ≈ 33.3 m/s. Critical velocities are higher on curves. Moreover, the model shows homogeneous properties. This means, that for velocities lower than the critical one there are stable stationary solutions only, and for higher velocities there are stable periodic solutions of a limit cycle nature only. The solutions lose stability at velocities much higher than the critical value on each tested curve.

The loaded wagon model has significantly different properties (Fig. 5). The research shows that the critical velocity on a straight track is 30.8 m/s. It is therefore lower than the maximum bogie operating velocity (33.3 m/s).

a) b)

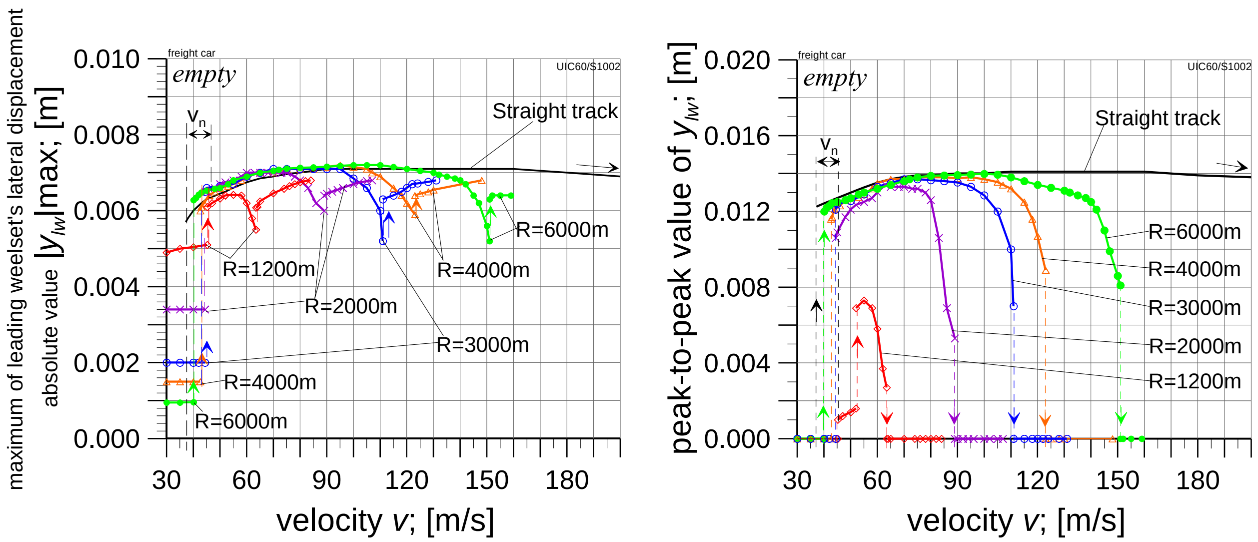

Fig. 4. The empty vehicle model stability map: a) maximum of leading wheelset lateral displacements absolute value $|y_{lw}|$max and b) peak-to-peak value of y_{lw} versus vehicle velocity

Table 2. The modelled vehicle critical velocity values v_n

Curve radius R [m]	1200	2000	3000	4000	6000	∞
Empty; v_n [m/s]	45.2	44.4	44.4	42.6	40.1	37.2
Loaded; v_n [m/s]	55.8	43.7	33	45.6	45	30.8

a) b)

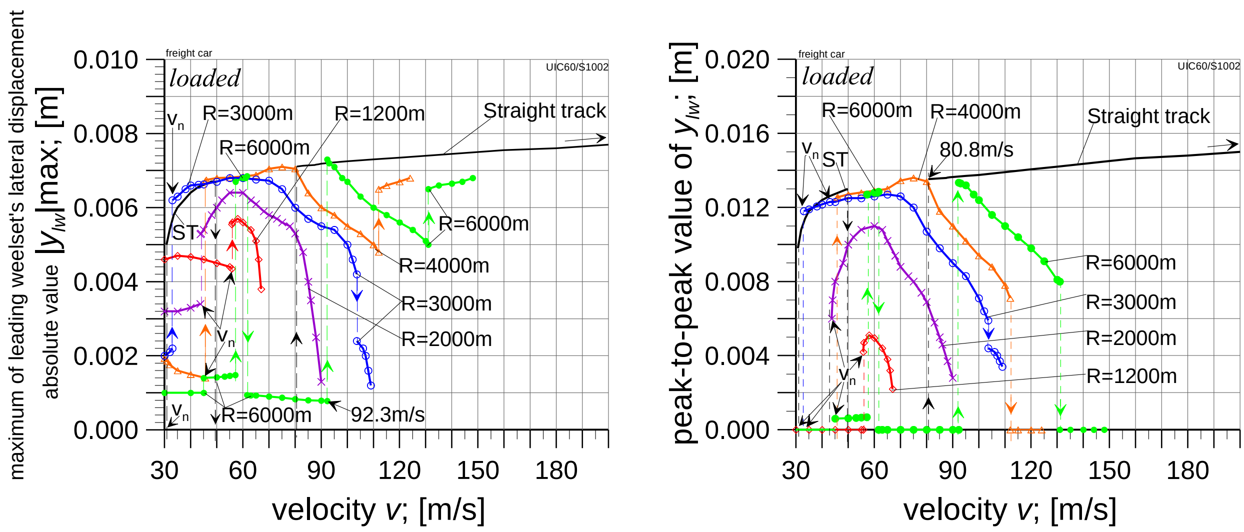

Fig. 5. The loaded vehicle model stability map: a) maximum of leading wheelset lateral displacements absolute value $|y_{lw}|$max and b) peak-to-peak value of y_{lw} versus vehicle velocity

In the aim of critical velocity increase, suspension parameters influence on the nature of model solutions was analysed. Result of this research is selection of the lateral stiffness k_{zy} as a parameter, which reduction in relation to the nominal value brings beneficial effects from the observed parameters point of view. As it was mentioned, the Y25 bogies does not have separate elements ensuring the lateral stiffness between wheelsets and bogie frame. It is executed by vertical springs that bend in the lateral direction (flexi coil effect). Nominal lateral stiffness value between the wheelsets and bogie frame is assumed $55.6{\cdot}10^5$ N/m. The conducted research shows that reducing this value leads to an increase the critical velocity on each tested routes. Straight track motion simulation results are presented below (Fig. 6). The lateral stiffness was reduced to 90% ($k_{zy} = 50{\cdot}10^5$ N/m) and to 80% ($k_{zy} = 44.5{\cdot}10^5$ N/m) of the nominal value.

As can be seen, the critical velocity increased from 30.8 m/s for the nominal lateral stiffness to 36.8 m/s for the lateral stiffness reduced to 90% of the nominal value. Reducing the k_{zy} to 80% of the nominal value causes the critical velocity significant increase to 136.1 m/s. The values of periodic solutions ($|y_{lw}|$max) remain at a similar level in each case and persist up to velocity exceeding 200 m/s.

Effect of reducing the lateral stiffness on first wheelset-track lateral forces Y_{max} was also examined. The results are presented in Fig. 7. As can be seen, the lateral forces are very small for stationary solutions. Periodic solutions appearance, causes a sudden increase of the lateral forces acting on the track to va-

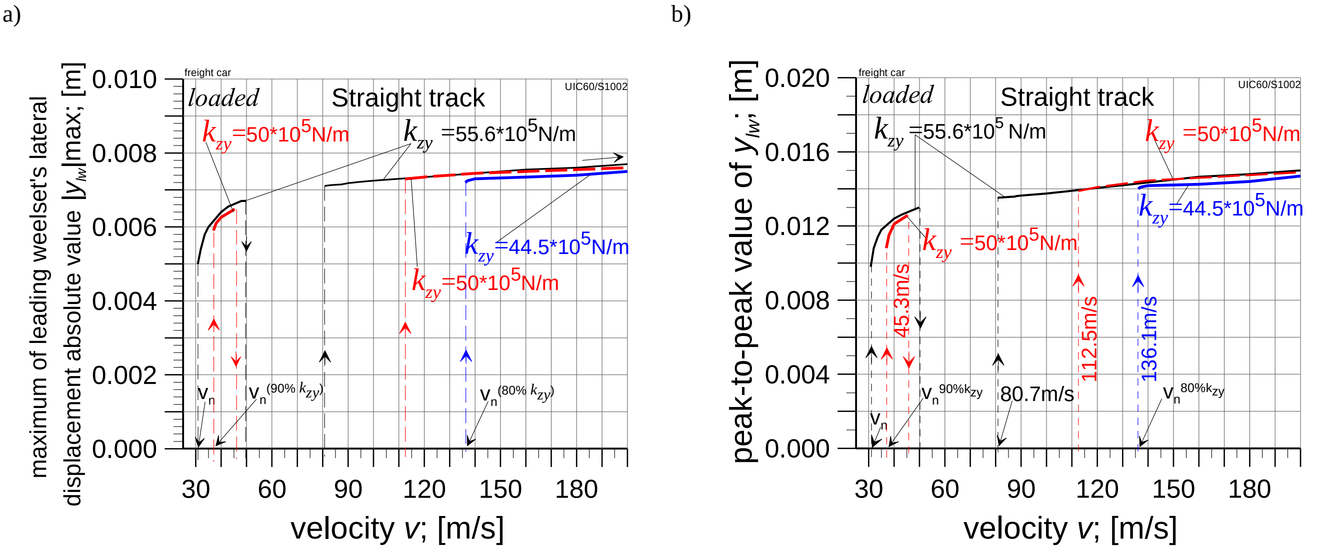

Fig. 6. Maximum of the first wheelset lateral displacements absolute value $|y_{lw}|$max – a and peak-to-peak value of y_{lw} – b versus vehicle velocity for different k_{zy} values: nominal one ($55.6{\cdot}10^5$ N/m), 90% of nominal value ($50{\cdot}10^5$ N/m) and 80% of nominal value ($44.5{\cdot}10^5$ N/m). Straight track motion

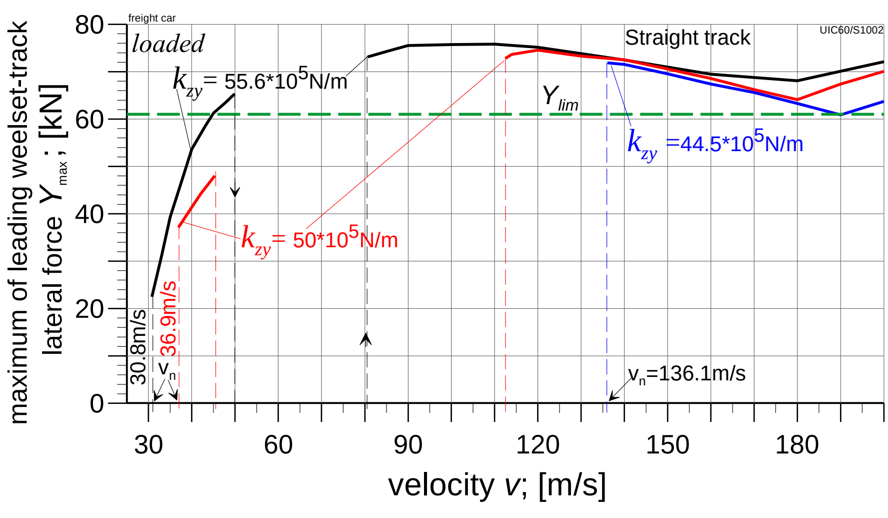

Fig. 7. The first wheelset-track lateral forces for different k_{zy} values: $55.6{\cdot}10^5$ N/m (nominal value), $50{\cdot}10^5$ N/m (90% of nominal value) and $44.5{\cdot}10^5$ N/m (80% of nominal value). Straight track motion

lues over 20 kN. For the model periodic solutions existence (self-excited vibrations of the wheelset), the lateral forces Y also change periodically at constant vehicle velocity. The diagram shows maximum (Y_{max}) lateral force values read from the simulation results for individual vehicle velocities. The limit value of vehicle-track lateral force (lateral impact on the track) Y_{lim} is determined by the possibility of track damage. Rails and sleepers permanent lateral displacements relative to the ballast may appear when Y_{lim} is exceeded. This is the so-called track criterion empirically defined as [5]:

$$Y_{lim} \le K \cdot \left(10 + \frac{2Q}{3}\right) \text{ [kN]} \tag{4.1}$$

where: Y_{lim} – sum of lateral forces acting on the 2 m long track section, K – factor depending on the type of sleepers, ballast type and its compaction, 2Q – static pressure of the wheelset on the track in [kN].

Assuming the coefficient K = 0.8 (concrete sleepers and compacted ballast) and 2Q = 200 kN (loaded coach conditions), the value of Y_{lim} is:

$$Y_{lim} \le 0.8 \cdot \left(10 + \frac{200}{3}\right) \approx 61 \text{ [kN]} \tag{4.2}$$

Therefore, the lateral force impact of this vehicle on the track from one wheelset should not exceed 61 kN. For the tested wagon, this means that reaching the critical velocity (appearance of self-excited vibrations) does not necessarily mean the occurrence of track damage for the nominal k_{zy} value and 90% of nominal k_{zy}. However, for 80% of k_{zy}, when the critical velocity reaches high values (136.1 m/s), the appearance of self-excited vibrations generates wheelset-track lateral forces, which may exceed Y_{lim} value. The impact of reducing lateral stiffness on the so-called safety factor against derailment Y/Q value is presented in Fig. 8. According to the recommendations [5] the Y/Q value should not exceed 0.8. As can be seen, for each k_{zy} value applied, the limited Y/Q value is not exceeded.

Reducing the lateral stiffness effect on vehicle properties in curved track was examined next. Results for two curves with radius R = 6000 m and R = 3000 m were selected to present. The stability map for curved track of curve radius R = 6000 m is presented in Fig. 9.

For the nominal lateral stiffness k_{zy}, self-excited vibrations with very small peak-to-peak values appear at velocity above 30 m/s. The critical velocity v_n = 45 m/s was assumed arbitrarily. Peak-to-peak values reach approximately 0.001 m at this velocity. Increasing the velocity causes a slight increase in peak-to-peak and at 57.1 m/s periodic solutions with peak-to-peak values of approximately 0.013 m appear. They persist up to velocity of 61.6 m/s and then bifurcate to stationary solutions. The next bifurcation of stationary to periodic solutions occurs at a velocity of 92.3 m/s. Periodic solutions persist up to velocity of 131 m/s. There is a transition to stationary solutions at the final stage of stable solutions existence, due to very high centrifugal force action. The stable solutions last up to approx. 148 m/s. Stable stationary solutions exist up to a velocity of 102.1 m/s for the lateral stiffness k_{zy} reducing to 90% of the nominal value. So, the wheelset vibrations with small peak-to-peak values may not occur in this case. Thus the critical velocity increase is significant. Next the critical velocity increases to 107.1 m/s for k_{zy} = 80% of the nominal value. In this case, also for velocity lower than v_n there are stationary solutions only, and for velocity higher than v_n there are periodic solutions, which exist up to 133 m/s.

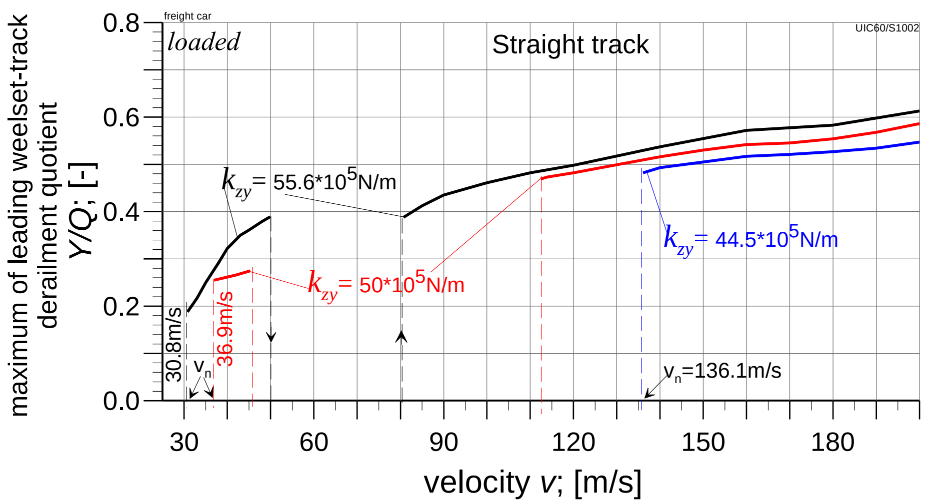

Fig. 8. The first wheelset derailment quotient Y/Q for different k_{zy} values: $55.6 \cdot 10^5$ N/m (nominal value), $50 \cdot 10^5$ N/m (90% of nominal value) and $44.5 \cdot 10^5$ N/m (80% of nominal value)

a) b)

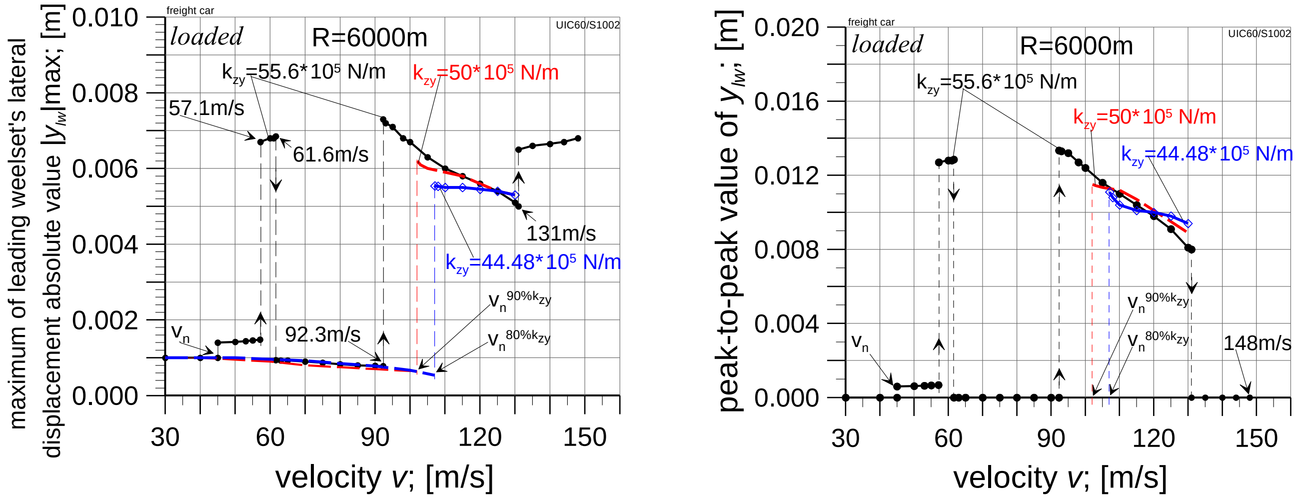

Fig. 9. Maximum of the first wheelset lateral displacements absolute value $|y_{lw}|max$ – a and peak-to-peak value of y_{lw} – b versus vehicle velocity for different k_{zy} values: nominal one ($55.6\cdot10^5$ N/m), 90% of nominal value ($50\cdot10^5$ N/m) and 80% of nominal value ($44.5\cdot10^5$ N/m). Curved track motion of curve value R = 6000 m

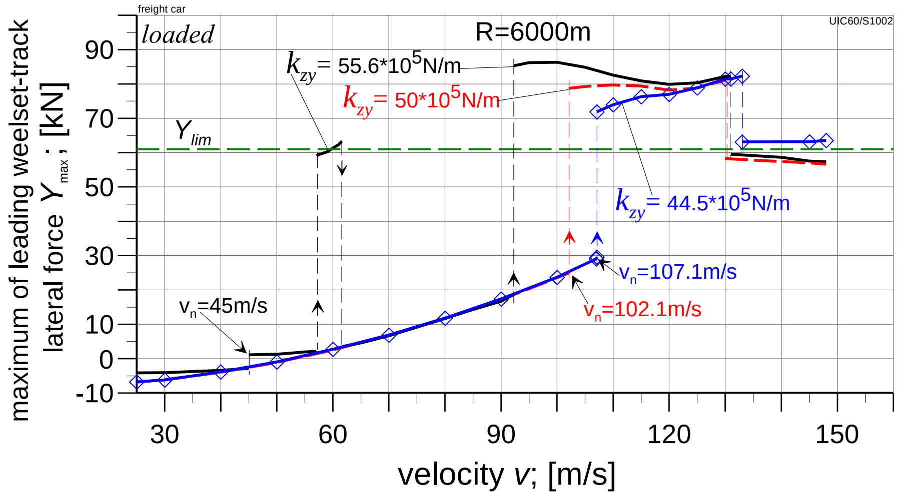

Fig. 10. The first wheelset-track lateral forces for different k_{zy} values: $55.6\cdot10^5$ N/m (nominal value), $50\cdot10^5$ N/m (90% of nominal value) and $44.5\cdot10^5$ N/m (80% of nominal value). Curve track of curve radius R = 6000 m

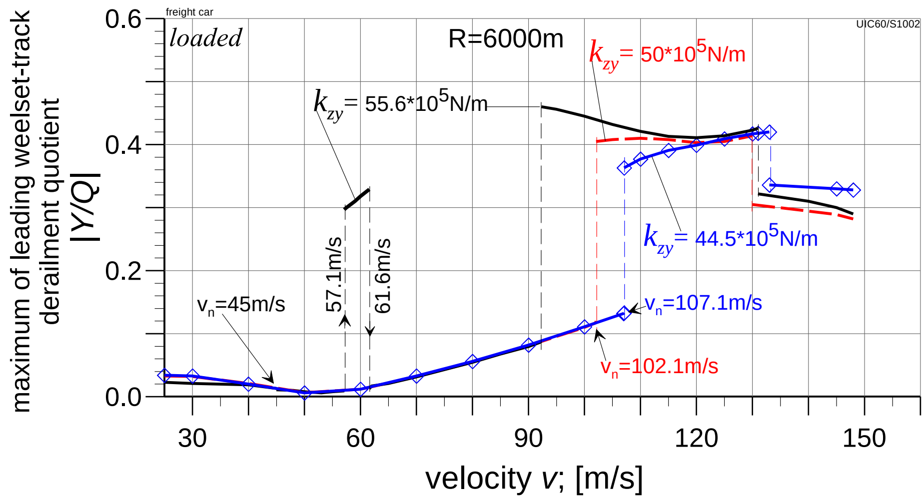

Fig. 11. The first wheelset derailment quotient absolute value |Y/Q| for different k_{zy} values: $55.6\cdot10^5$ N/m (nominal value), $50\cdot10^5$ N/m (90% of nominal value) and $44.5\cdot10^5$ N/m (80% of nominal value). Curve track motion of curve radius R = 6000 m

Maximum values of the first wheelset lateral forces impact on the track Y_{max} is shown in Fig. 10. The lateral wheelset-track forces change sign (direction) with the velocity increase in the under critical velocity range. This is an effect of the centrifugal force appearance and track superelevation h = 0.051 m application. Balance of the centrifugal force and acting parallel in the track plane component of the gravitational force occurs at velocity of approximately 50 m/s.

The lateral forces are significantly smaller than Y_{lim} for stationary model solutions. Reaching the critical velocity means exceeding the Y_{lim} value for each k_{zy} values. Lateral stiffness changes influence on the safety factor against derailment Y/Q is shown in Fig. 11. As can be seen, in the entire range of stable solutions existence Y/Q does not exceed 0.5.

Curved track motion with radius R = 3000 m is analysed next. Stability map is shown in Fig. 12. Critical velocity is 33 m/s for the nominal lateral stiffness value. It is therefore slightly lower than the maximum operating vehicle velocity of 33.3 m/s (120 km/h).

Reducing the k_{zy} to 90% of the nominal value increases critical velocity to 34.6 m/s. Reducing the k_{zy} to 80% of the nominal value results in a significant increase in the critical velocity to 73.3 m/s. The solution values are similar for each k_{zy} value. Stable stationary solutions are only in the velocity range lower than the critical value. There is a bifurcation of stable periodic solutions to also stable periodic ones with smaller peak-to-peak values at velocity of approximately 105 m/s. The stable solutions end at velocity of 108 m/s for each k_{zy} value. The first wheelset-track lateral forces Y_{max} are much lower than the limited value Y_{lim} in the range of stable stationary solutions existence (Fig. 13).

a) b)

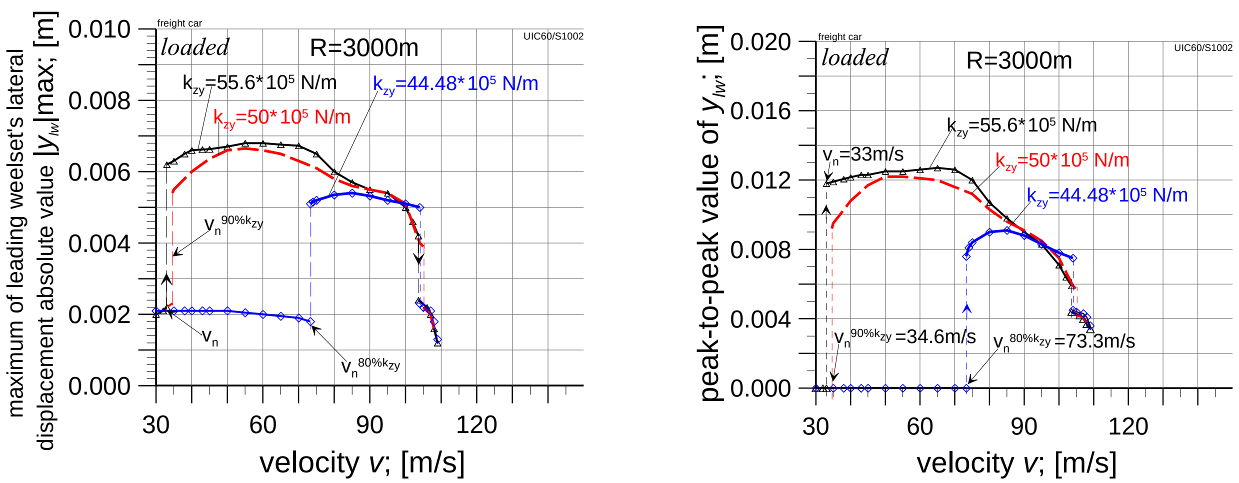

Fig.12. Maximum of the first wheelset lateral displacements absolute value $|y_{lw}|max$ – a and peak-to-peak value of y_{lw} – b versus vehicle velocity for different k_{zy} values: nominal one ($55.6 \cdot 10^5$ N/m), 90% of nominal value ($50 \cdot 10^5$ N/m) and 80% of nominal value ($44.5 \cdot 10^5$ N/m). Curved track motion of curve value R = 3000 m

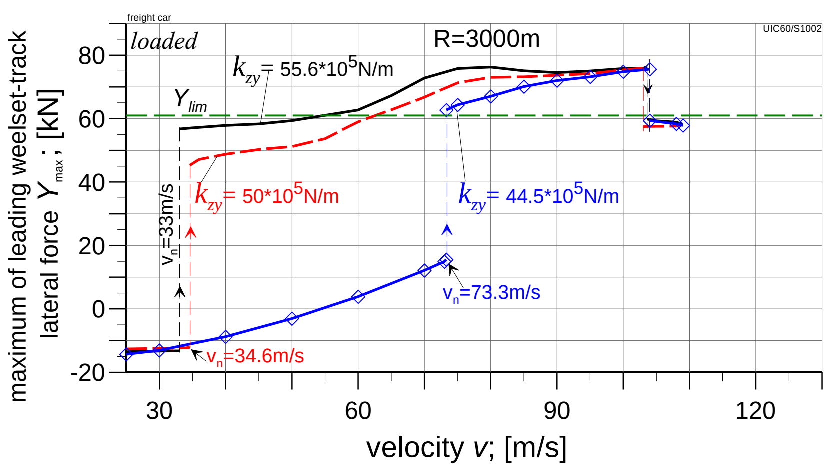

Fig. 13. The first wheelset-track lateral forces for different k_{zy} values: $55.6 \cdot 10^5$ N/m (nominal value), $50 \cdot 10^5$ N/m (90% of nominal value) and $44.5 \cdot 10^5$ N/m (80% of nominal value). Curve track of curve radius R = 3000 m

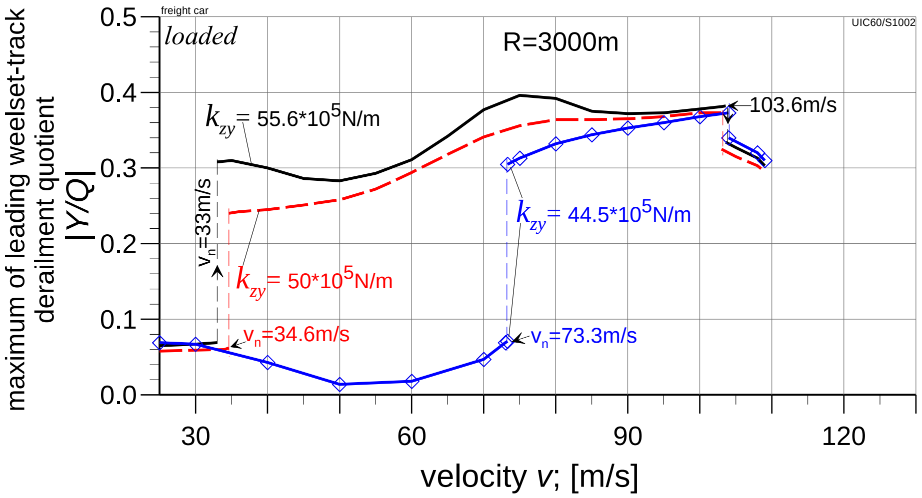

Fig. 14. The first wheelset derailment quotient absolute value |Y/Q| for different k_{zy} values: 55.6·10^5 N/m (nominal value), 50·10^5 N/m (90% of nominal value) and 44.5·10^5 N/m (80% of nominal value). Curve track motion of curve radius R = 3000 m

Reaching the critical velocity causes a sudden increase in the Y_{max} value, however, for the nominal k_{zy} applied and 90% of k_{zy}, the limit of lateral force Y_{lim} is not exceeded. Lateral forces increase in overcritical velocity range and exceed the Y_{lim} value for velocity greater than 60 m/s.

The safety factor against derailment Y/Q does not exceed 0.1 in the range of stable stationary solutions existence Fig. 14. Reaching the critical velocity causes a sudden increase in Y/Q, but the maximum value does not exceed 0.4, for all k_{zy} values applied.

5. Conclusions

The numerical research shows that the modeled wagon with Y25 bogies has different dynamic properties when empty and when loaded. Empty wagon meets the requirements resulting from operating conditions. The requirements are not met at velocity greater than 100 km/h, when loaded. The critical velocity v_n may be lower than the maximum operating vehicle velocity 120 km/h, for the adopted nominal suspension parameters. This means that the wheelset-track self-excited vibrations may associate vehicle motion. Reducing the first stage suspension lateral stiffness k_{zy} by dozen percent in relation to nominal value is the possible way of eliminating the unfavorable characteristics of the modelled wagon in loaded condition at velocity bigger than 100 km/h. The lateral stiffness k_{zy} and vertical stiffness k_{zz} are executed by the same vertical springs. So, the method of implementing the k_{zy} reduction without a k_{zz} change remains an open question.

Accuracy of the obtained results is subject to some error, resulting from the simplification of the model by describing elements with dry friction in the real system, with elements with viscous friction in the model. Eliminating this inaccuracy will be the subject of further author works.

Acknowledgements

This work was supported by the National Center for Research and Development (NCBiR), Poland, under the TANGO program no. – TANGO-IV-A/0027/2019-00.

Nomenclature

h	track superelevation
k_{zy}	the Y25 first stage suspension lateral stiffness
R	track curve radius
ST	straight track
v_n	nonlinear critical velocity
y_{lw}	leading wheelset lateral displacements
Y_{max}	maximum of leading wheelset-track lateral force
Y/Q	derailment quotient

Bibliography

[1] Bogusz W. Technical stability (in Polish: Stateczność techniczna). PWN, Warsaw 1972.

[2] Bruni S, Vinolas J, Berg M, Polach O, Stichel S. Modeling of suspension components in a rail vehicle dynamics context. Vehicle Syst Dyn. 2011;49(7): 1021-1072. https://doi.org/10.1080/00423114.2011.586430

[3] Csiba J. Bogie type anniversary: the bogie type Y25 is over 50 years. Proceedings of the 10th International Conference on Railway Bogies and Running Gears. Budapest, 12-15 September 2016, 253-262.

[4] Dusza M. Rail vehicle model possibility of safe motion analysis in the overcritical velocity range. Proceedings of the 11th International Conference on Railway Bogies and Running Gears, Edited by Zobory I. Department of Rolling Stock, SSME/GTE, Budapest 2020, 159-168.

[5] European Standard PN-EN 14363+A1:2019-02.

[6] Iwnicki S. (ed.). Handbook of railway vehicle dynamics. CRC Press Inc. 2006. https://doi.org/10.1201/9780429469398

[7] Kaiser AB, Cusumano JP, Gardner JF. Modelling and dynamics of friction wedge dampers in railroad trucks. Vehicle Syst Dyn. 2002;38(1):55-82. https://doi.org/10.1076/vesd.38.1.55.3519

[8] Kalker JJ. A fast algorithm for the simplified theory of rolling contact. Vehicle Syst Dyn. 1982;11:1-13. https://doi.org/10.1080/00423118208968684

[9] Knothe K, Böhm F. History of stability of railway and road vehicles. Vehicle Syst Dyn. 1999;31:283-323. https://doi.org/10.1076/vesd.31.5.283.8362

[10] Pagaimo J, Magalheas H, Costa JN, Ambrosio J. Derailment study of railway cargo vehicle using a response surface methodology. Vehicle Syst Dyn. 2020; 60(1):309-334. https://doi.org/10.1080/00423114.2020.1815810

[11] Piotrowski J, Pazdzierniak P, Adamczewski T. Curving dynamics of freight wagon with one- and two-dimensional friction damping. Proc. of the 10th Mini Conference on Vehicle System Dynamics. Identification and Anomalies. Edited by Zobory I. Budapest 6-8 November 2006, 215-222.

[12] Piotrowski J. Kalker's algorithm Fastsim solves tangential contact problems with slip-dependent friction and friction anisotropy. Vehicle Syst Dyn. 2010; 48(7):869-889. https://doi.org/10.1080/00423110903178495

[13] Shabana AA, Zaazaa KE, Sugiyama H. Railroad vehicle dynamics: a computational approach. Taylor & Francis LLC and the CRC, 2008. https://doi.org/10.1201/9781420045857

[14] Zboiński K, Dusza M. Self-exciting vibrations and Hopf's bifurcation in non-linear stability analysis of rail vehicles in curved track. Eur J Mech A-Solid. 2010;29(2):190-203. https://doi.org/10.1016/j.euromechsol.2009.10.001

[15] Zboiński K, Dusza M. Extended study of rail vehicle lateral stability in a curved track. Vehicle Syst Dyn. 2011;49(5):789-810. https://doi.org/10.1080/00423111003770447

[16] Zboiński K, Dusza M. Bifurcation analysis of 4-axle rail vehicle models in a curved track. Nonlinear Dynam. 2017;89(2):863-885. https://doi.org/10.1007/s11071-017-3489-y

Appendix

Parameters of the freight car accepted to research

Notation	Description	Measurement unit	Value	
			empty	loaded
m_{cb}	Vehicle body mass	kg	11,000	72,000
m_b	Bogie frame mass	kg	1600	
m_w	Wheelset mass	kg	1400	
m_{ab}	Axlebox mass	kg	100	
$I_{\xi cb}$	Body moment inertia; longitudinal axis	kg·m^2	17,300	90,055
$I_{\eta cb}$	Body moment inertia; lateral axis	kg·m^2	188,500	1210,606
$I_{\psi cb}$	Body moment inertia; vertical axis	kg·m^2	188,140	1231,450
$I_{\xi b}$	Bogie frame moment inertia; longitudinal axis	kg·m^2	790	
$I_{\eta b}$	Bogie frame moment inertia; lateral axis	kg·m^2	1000	
$I_{\psi b}$	Bogie frame moment inertia; vertical axis	kg·m^2	1090	
$I_{\xi w}$	Wheelset moment inertia; longitudinal axis	kg·m^2	747	
$I_{\eta w}$	Wheelset moment inertia; lateral axis	kg·m^2	131	
$I_{\psi w}$	Wheelset moment inertia; vertical axis	kg·m^2	747	
k_{zz}	Vertical stiffness of the 1st level of suspension	kN/m	1017	2280
k_{zy}	Lateral stiffness of the 1st level of suspension	kN/m	3890	5560
k_{zx}	Longitudinal stiffness of the 1st level of suspension	kN/m	12,000	12,000
c_{zz}	Vertical damping of the 1st level of suspension	kN·s/m	7	123.3
c_{zy}	Lateral damping of the 1st level of suspension	kN·s/m	42	138
c_{zx}	Longitudinal damping of the 1st level of suspension	kN·s/m	100	
k_{2z}	Vertical stiffness of the bogie frame – car body side bearer	kN/m	22,500	
c_{2x}	Longitudinal damping on the bogie frame – car body side bearer	kN·s/m	6	10
$k_{2\psi}$	Torsional stiffness between bogie frame and car body	kN·m/rad	20	
$c_{2\psi}$	Torsional damping between bogie frame and car body	kN·m·s/rad	0.5	
a_p	Half of bogie's pivot-pivot distance	m	4.5	
a	Semi-wheel base	m	0.9	
tc	Semi-tape circle distance	m	0.75	
h_b	Vertical distance between bogie frame centre mass and track plane	m	0.69	
h_{cb}	Vertical distance between car body centre mass and track plane	m	1.5	1.87
r_t	Wheel radius	m	0.46	
μ	Wheel-rail coefficient of friction	–	0.4	