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Determination of Stability Conditions for High-Speed Electric Trains with Carbody Tilting When Negotiating Curved Track Sections

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1. Introduction

The increase in the efficiency of passenger transportation on existing railway infrastructure remains a critical challenge for modern railways, particularly under conditions of limited financial resources and the high cost of constructing dedicated high-speed lines. In this context, the use of rolling stock equipped with carbody tilting systems represents an effective alternative, allowing higher operating speeds on conventional tracks without substantial infrastructure upgrades. Such systems improve passenger comfort by compensating for lateral acceleration in curves while maintaining acceptable levels of dynamic interaction between the vehicle and the track. At the same time, the implementation of tilting technologies introduces additional complexity into the mechanical system of the vehicle, which requires careful analysis of its influence on safety-related parameters.

Recent studies have addressed various aspects of tilting train technology, including control strategies, dynamic behavior, and infrastructure interaction. The authors of [1] investigated the nonlinear stability of railway tracks under the influence of unbalanced cant and train speed, demonstrating that changes in dynamic loading conditions can significantly affect track stability and, consequently, the safety of train operation.

The development of tilting control systems has also been widely studied. The authors of [2] proposed multi-objective optimization approaches for tilt control design, emphasizing the need to balance passenger comfort and dynamic performance. Reference [3] explored the application of fractional-order control methods, demonstrating their potential to improve the robustness and efficiency of railway systems. However, these studies primarily focus on control algorithms rather than on the mechanical configuration itself.

In the field of simulation modelling, the authors of [4] developed a model of a hybrid electropneumatic tilting system, highlighting the advantages of combining different actuation principles. Earlier, the authors of [5] proposed a simulation approach for combined tilting systems of high-speed rolling stock, demonstrating the feasibility of analyzing such systems using modern computational tools. Despite these contributions, the influence of the geometric parameters of the tilting mechanism – particularly the coordinates of lever attachment points – on safety indicators has not been sufficiently investigated.

At the same time, the interaction between rolling stock and track infrastructure continues to be an important research area. The authors of [6] reviewed methods for monitoring railway track resilience, emphasizing the importance of accounting for dynamic loading conditions. In [7], the authors discussed the adaptation of railway systems to future requirements, noting that innovations in rolling stock design must be accompanied by a thorough assessment of their impact on infrastructure and safety.

Despite the significant body of research on tilting train systems, there remains a lack of comprehensive studies addressing the relationship between the structural design of the tilting mechanism and derailment stability. In particular, the redistribution of vertical loads between the wheels of a wheelset due to changes in the geometry of the mechanism can directly influence the derailment coefficient and overall running safety. Modern simulation environments, such as MATLAB/Simulink with the Simscape library, provide advanced capabilities for modelling complex mechanical systems as interconnected physical components, enabling efficient parametric analysis without the need for repeated reformulation of differential equations.

Unlike previous studies, including our earlier work devoted to the development of simulation models and the determination of actuator force characteristics, the present study focuses on the influence of the geometric configuration of the carbody tilting mechanism on running safety. The scientific novelty of this research lies in establishing quantitative relationships between the coordinates of lever attachment points, the derailment stability coefficient, and the carbody rotation moment by combining a MATLAB/Simscape multibody model with a fractional factorial experimental design and third-order regression modelling. The proposed approach not only extends the existing methodology to account for the redistribution of vertical wheel loads caused by variations in mechanism geometry but also provides an efficient engineering tool for identifying feasible design regions while substantially reducing the number of required simulation runs.

The aim of this study is to investigate the influence of the geometric parameters of a carbody tilting mechanism,

specifically the coordinates of lever attachment points, on the stability conditions and safety indicators of high-speed electric trains operating on curved track sections.

2. Analysis of Recent Research and Problem Statement

Previous studies have demonstrated the effectiveness of carbody tilting systems in improving the performance of high-speed railway transport [2, 4]. Other researchers have proposed advanced actuation principles, including electromagnetic and hybrid tilting systems [5]. However, insufficient attention has been paid to the redistribution of vertical wheel loads caused by carbody tilting, despite its significant influence on wheel–rail interaction, infrastructure loading, and traffic safety [1, 6-8].

When a train negotiates a curve, centrifugal force redistributes vertical loads between the wheels of each wheelset. This effect is further influenced by the displacement of the carbody center of mass during tilting, which changes the position of the resultant gravitational force relative to the track plane and consequently affects wheel–rail contact forces. In passive tilting systems, the centre of mass shifts toward the outer rail, which may increase the load on the outer wheels. In contrast, active tilting systems provide controlled displacement of the carbody toward the inner rail, improving centrifugal force compensation, reducing wheel load imbalance, and enhancing both passenger comfort and wheel–rail interaction.

Traffic safety depends on several dynamic criteria, including derailment stability, track stability, vehicle overturning resistance, and the strength of rails and running gear components [9, 10]. Among these, wheel derailment stability is one of the most critical indicators, as the interaction between wheel flanges and rail heads governs reliable wheel guidance, particularly on curved track sections [11].

3. Materials and Methods of Research

The leading wheels of bogies, when running on straight or curved track sections, contact the side faces of the rail heads with their flanges. For the flange to slide downward along the rail head, that is, for the wheel not to climb onto the rail head, the following condition must be satisfied:

$$K_c = \frac{tg\beta - \mu}{1 + \mu\, tg\beta} \cdot \left(\frac{P_1}{Y_b} \right) > 1, \tag{1}$$

where: K_c is the safety factor of the wheel against derailment; β is the inclination angle of the wheel flange to the horizontal, deg.; μ is the coefficient of friction between the wheel and the rail head; P_1, Y_b are the vertical and horizontal forces acting on the wheel, respectively, N.

By rearranging Eq. (1), the expression for the derailment stability coefficient is obtained as

$$K_{c1} = \left(\frac{P_1}{Y_b} \right) > 1.4...1.5\,. \tag{2}$$

In [5], the authors analyzed the limit configurations of the carbody tilting mechanism for a typical electric train bogie and determined the trajectory of the vehicle's center of mass during carbody tilting up to an angle of 8°. These results were used to calculate the force required from the linear actuator; however, the resulting redistribution of vertical wheel loads was not considered.

In the present study, the proposed methodology is extended to evaluate the center-of-mass displacement for the limit configurations of the mechanism and its effect on the redistribution of vertical wheel loads.

During carbody tilting, the distance between the vehicle's center of mass and the rotation center varies owing to both passengers loading and the geometry of the tilting mechanism. The lateral displacement d' of the center of mass is expressed as

$$d' = h \cdot tg\theta\,, \tag{3}$$

where: h is the vertical distance from the rail head to the vehicle's center of mass, and θ is the carbody tilting angle.

The total change in loads on the outer ΔA and inner ΔB rails, caused by the centrifugal forces and the displacement of the center of mass of the "carbody–passengers" system, is equal to

$$\begin{aligned} \Delta A &= -\frac{MV^2}{R}\frac{h}{2S} + Mg\frac{S \mp d'}{2S}, \\ \Delta B &= \frac{MV^2}{R}\frac{h}{2S} + Mg\frac{S \pm d'}{2S}, \end{aligned} \tag{4}$$

where: M is the vehicle mass, including passengers, assigned to one wheelset; V is the train speed; R is the curve radius; $2S$ is the track gauge; and g is the acceleration due to gravity.

The simulations were performed for a representative high-speed electric train operating on a conventional railway line. The analysis was carried out for a constant carbody tilt angle of 8°, corresponding to the maximum operating tilt considered in the mechanical design of the proposed system. The vehicle mass, including passenger load assigned to one wheelset, curve radius, train speed, track gauge, rail cant, and the center-of-mass height were specified according to the model parameters described in [4, 5, 12]. During the parametric study, only the coordinates of the lever attachment points were varied, whereas all remaining vehicle, track, and operating parameters were kept constant to isolate the influence of mechanism geometry on derailment stability and carbody rotation moment. The simulation model developed in [4, 5, 12] was adopted as the basis for this study. However, its functionality was unnecessarily extensive for the present analysis, resulting in excessive computational time. Therefore, the model was simplified and redesigned, while the model parameters were transferred to an external data file, enabling rapid parametric modification.

The modified model was implemented in MATLAB/Simulink [13]. The mechanical subsystem was built using the Simscape library, whereas model execution and parameter transfer were controlled through a MATLAB script [14, 15].

The simulation model generates load characteristic graphs for different configurations of the studied mechanism [14].

A numerical experiment was conducted to evaluate the influence of the coordinates of the carbody tilting mechanism lever attachment points on the derailment stability

coefficient and the moment about the instantaneous centre of rotation [2, 16].

To investigate the effects of the selected design parameters, a fractional factorial design 5^{4-1} was employed to identify a cubic regression model. This approach provides an efficient compromise between the number of simulations and the accuracy of the resulting mathematical model [17, 18].

The regularity of the design means that the set of combinations of factor levels can be obtained using a defining relation and a generating relation [17, 19, 20].

Algorithm for constructing the design and the regression model. The block diagram of the regression model calculation algorithm is shown in Fig. 1.

A full factorial design would require a total of 625 experiments, making it impractical for engineering applications because of the considerable computational time and resources involved.

A fractional factorial design (FFD) was therefore adopted, allowing the system to be analysed using only a subset of all possible factor-level combinations while preserving sufficient information to estimate the main effects and selected interaction effects [17].

Cubic regression model. In engineering modeling problems, third-order polynomial models are often used to adequately approximate nonlinear relationships [17, 19].

To further solve the optimization problem, the ranges of possible positions of the suspension arms in the proposed coordinate system were determined. The upper right arm is defined in absolute coordinates as $X_1 = X'_1$ ranging from 0.2 to 0.65 m, and $X_2 = Y'$ ranging from 0.2 to 0.5 m. In contrast, X_3 and X_4 are defined in a local coordinate system relative to the upper hinge position. Their absolute positions are therefore obtained from the corresponding transformation equations. Specifically, the lower right arm is described by $X_3 = X'_2 – X_1$, ranging from 0.1 to 0.5 m, and $X_4 = X_2 – Y'_2$, ranging from 0.2 to 0.65 m. Consequently, positive values of X_3 and X_4, they are lower than X_1 and X_2 correspond to the lower attachment point being located below the upper attachment point in the actual mechanism. The variation ranges are indicated in Fig. 2 by cross-hatching.

The coordinates X'_1, X'_2, Y'_1, and Y'_2 describe the positions of the upper and lower arms, as shown in Fig. 2. Such a geometric range of parameter variation is justified by the integration of the body tilt mechanism into the overall structure of the train running gear with a tilting body.

The rational design of the body tilt mechanism assumes that the reaction force generated by the car body tilt system, relative to the moment of rotation of the car body about its instantaneous center of rotation, corresponds to the tractive force of the tilt drive motor, while simultaneously satisfying the requirements of the stability coefficient (2), which determines system operability.

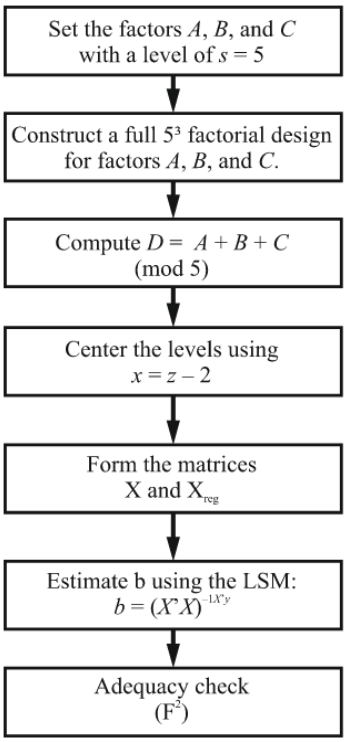


Fig. 1 Block diagram of the regression model calculation algorithm: *A, B, C* – independent factors; *s* – level of coordinate space; 5^3 – factorial design; *D* – factor determined by the experimental conditions; *x* – centered levels; *z* – initial levels; *X* – experimental matrix; X_{reg} – matrix of cubic regressors; *b* – regression coefficients; F^2 – model adequacy check

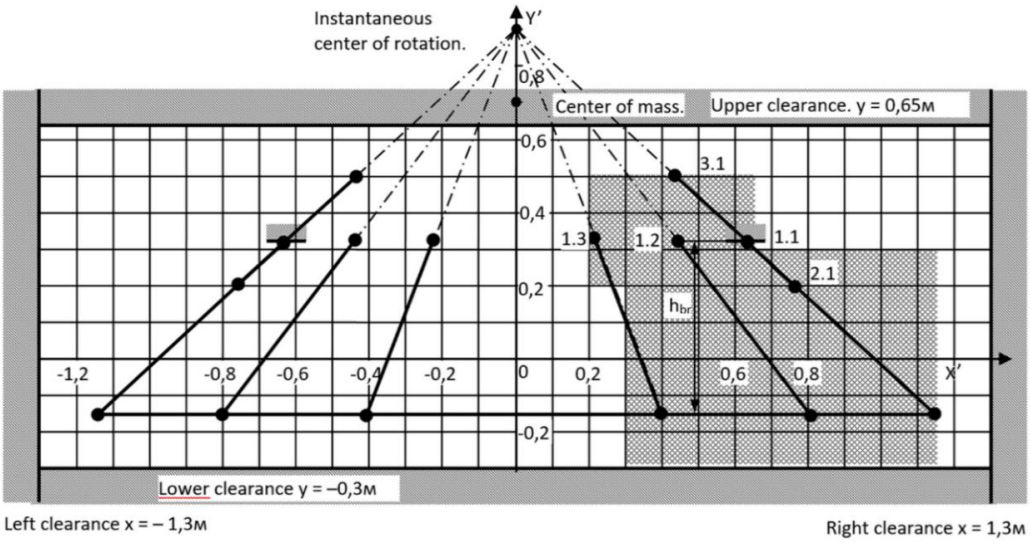


Fig. 2 Variation of the car body tilting mechanism configuration

The coordinates were transformed into coded form

$$x_i = \left(X_i / 1000 + Z_i\right) / K_i, \quad (5)$$

where Z_i and K_i are the scale coefficient and the shift coefficient of the parameter that determines the position of the body tilting levers (Fig. 2), measured in millimeters. The variation ranges are indicated in Fig. 2 by cross-hatching.

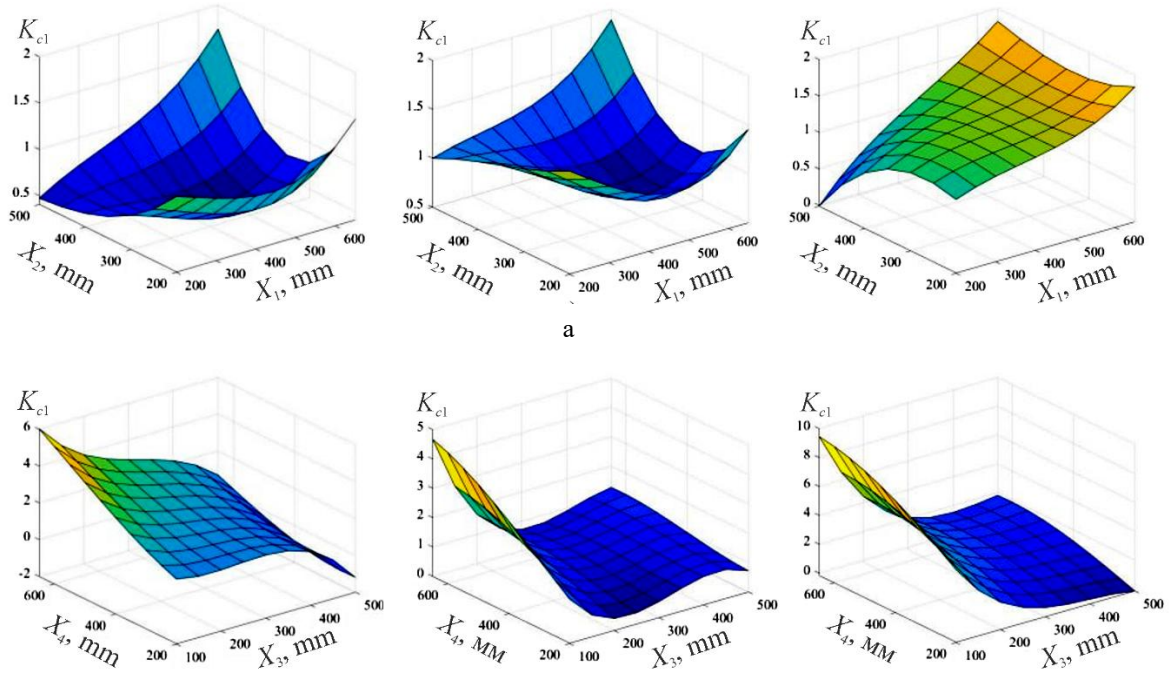


Fig. 3 Regression model of the stability coefficient: K_{c1} derailment stability coefficient; X_1 & X_2 – coordinates of the lower attachment points of the carbody suspension levers; X_3 & X_4 coordinates of the upper attachment points. a – minimum, average, and maximum values of the coordinates of the lower attachment points of the carbody suspension levers; b – minimum, average, and maximum values of the coordinates of the upper attachment points of the carbody suspension levers

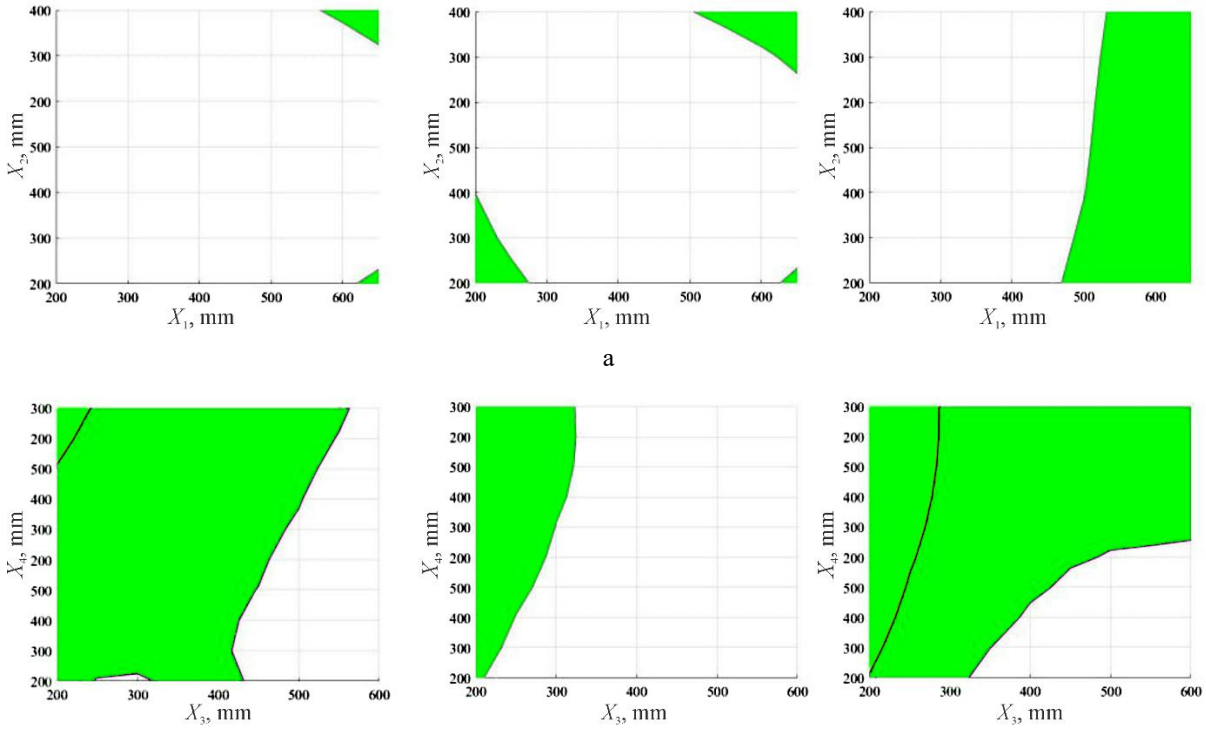


Fig. 4 Operability zones of the carbody tilting mechanism with respect to stability coefficient limits: a – minimum, average, and maximum values of the coordinates of the lower attachment points of the carbody suspension levers; b – minimum, average, and maximum values of the coordinates of the upper attachment points of the carbody suspension levers

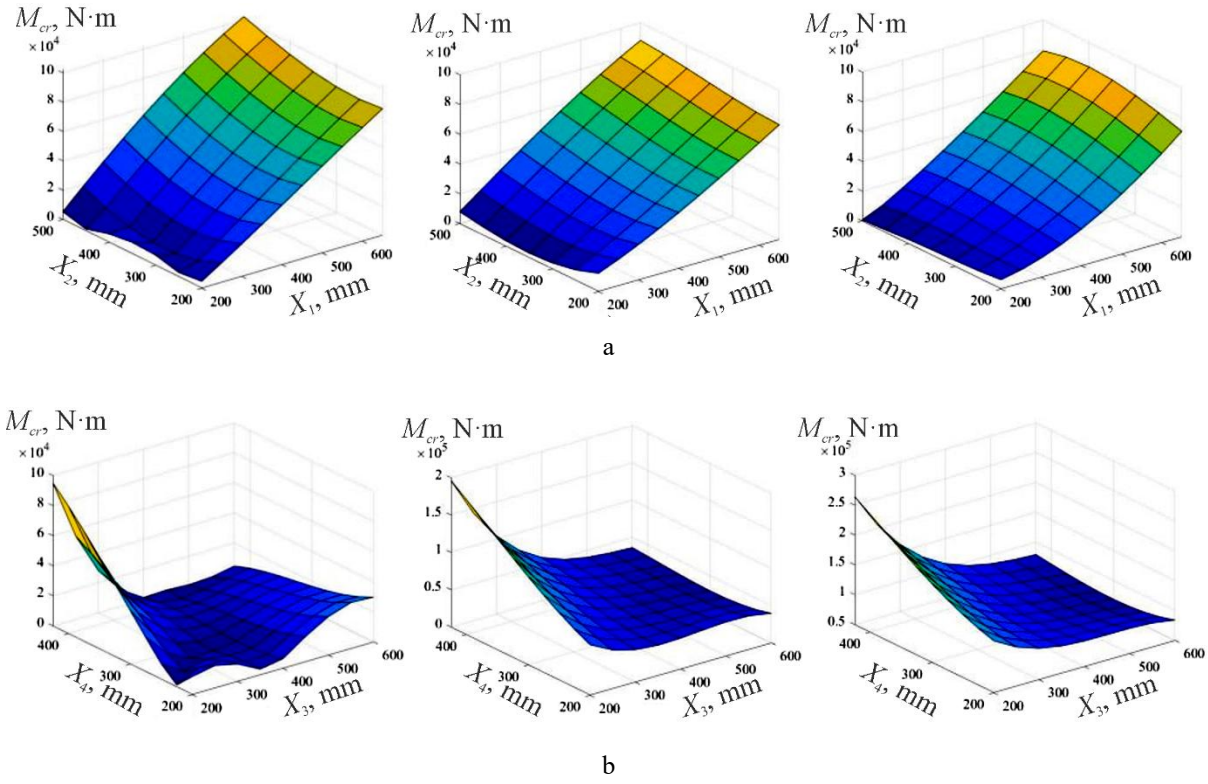


Fig. 5 Regression model of the carbody rotation moment relative to the instantaneous center of rotation: M_{cr} – carbody rotation moment relative to the instantaneous center of rotation: a – minimum, average, and maximum values of the coordinates of the lower attachment points of the carbody suspension levers; b – minimum, average, and maximum values of the coordinates of the upper attachment points of the carbody suspension levers

Model adequacy check according to the coefficient of determination. R^2 is a measure of the quality of the regression model, indicating the proportion of the response variation y explained by the model.

The coefficients of determination for the stability coefficient are at the level of $R^2_{Kc1} = 0.905$, and for the carbody rotation moment relative to the instantaneous center of rotation, they are at the level of $R^2_{Mcr} = 0.987$, which confirms the adequacy of the regression mathematical model.

4. Results

Analysis of the results of the numerical experiment for determining the stability coefficient and the carbody rotation moment relative to the instantaneous center of rotation. Fig. 3 shows the graph of the regression model of the stability coefficient for the minimum, average, and maximum values of the coordinates of the upper Fig. 3, b and lower Fig. 3, a; attachment points of the carbody suspension levers.

An important criterion for the operability of the carbody tilting mechanism is that the stability coefficient satisfies the condition expressed in Eq. 2. Geometrically, the operability zone can be represented in Fig. 4, showing the zones of operability shaded according to the variations in the coordinates of the lever attachment points for the minimum, average, and maximum values of the upper Fig. 4, b and lower Fig. 4, a carbody lever coordinates.

Fig. 5 shows the graph of the regression model of the carbody rotation moment relative to the instantaneous center of rotation for the minimum, average, and maximum values of the upper Fig. 5, b and lower Fig. 5, a carbody lever attachment point coordinates.

An analysis of the stability of high-speed trains with tilting carbodies on curved track sections was performed. The results show that reduced contact forces significantly decrease derailment safety margins and require stricter constraints on suspension and tilting system parameters.

Unlike the approaches in [14], the proposed model enables variation of the spatial coordinates of lever attachment points on both the car body and the bogie, allowing a comprehensive parametric study of the tilt mechanism geometry. This capability makes it possible to identify the key geometric factors influencing system stability and force distribution in the suspension.

5. Conclusions

To investigate the relationships between the stability coefficient, carbody rotation moment, and lever attachment point coordinates, a fractional factorial design (5^{4-1}) was applied. This approach reduced the number of simulations while preserving the ability to assess main and interaction effects. Based on the simulation data, a regression analysis was performed, resulting in a third-order polynomial model with 31 coefficients.

The regression model demonstrated high adequacy and revealed nonlinear dependencies and interaction effects among geometric parameters of the tilting mechanism. The vertical and longitudinal positions of the lever attachment points were found to have the greatest influence on the stability coefficient, while transverse positioning plays a secondary role through interactions with other factors. The analysis identified optimal parameter regions where maximum stability is achieved without exceeding acceptable

rotation moments, whereas deviations from these regions lead to reduced stability or increased actuation effort and potential system overload.

It was found that for initial lever angles below 52°, an operability zone exists for upper attachment points located more than 530 mm from the axis, ensuring reliable operation without loss of kinematic consistency. At higher tilt angles, this zone decreases due to increased geometric constraints and sensitivity to parameter variations. The operability region for lower attachment points expands when the upper coordinates (X_1, X_2) approach their extreme values, indicating reduced sensitivity and improved design robustness in these boundary regions.

It was determined that the carbody rotation moment relative to the instantaneous center of rotation generally increases monotonically with increasing upper attachment point coordinates both horizontally (X_1) and vertically (X_2). Physically, this is associated with an increase in the effective lever arm and, consequently, a higher torque required to achieve the same tilt angle. The variation is nearly constant for most positions of the lower attachment point, indicating a weaker influence of these parameters compared to the upper attachment configuration. The minimum rotation moment is 706 Nm, and the maximum is 70,829 Nm, demonstrating a wide range of possible loading conditions. Increasing the lower attachment point coordinates reduces the moment from 137,080 Nm to 6,815 Nm, highlighting their role as an effective means of reducing actuation effort.

This study investigated the influence of geometric parameters of a carbody tilting mechanism on running safety indicators of high-speed trains operating on conventional railway lines. The developed simulation model in MATLAB Simscape, combined with a fractional factorial design and regression analysis, proved effective for evaluating the stability coefficient and carbody rotation moment at a tilt angle of 8°.

The constructed third-order regression models demonstrated high adequacy (0.905 for stability and 0.987 for rotation moment), confirming their suitability for engineering applications. The results revealed pronounced nonlinear dependencies between the studied parameters and the coordinates of the lever attachment points. A geometric interpretation of the operability region showed that initial lever angles below 52° ensure a significantly wider range of feasible configurations, while larger angles reduce system operability.

It was established that the upper attachment point coordinates have the greatest influence on both stability and actuation effort, whereas appropriate positioning of lower points can effectively reduce the required rotation moment.

Further research should address dynamic vehicle–track interaction, include suspension and control system parameters, and provide experimental validation, with a focus on multi-objective optimization.

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S u m m a r y

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Keywords: rolling stock, tilting system, stability coefficient, attachment coordinates, fractional factorial design, regression analysis.

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