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The method of determining design assumptions for bogie platform

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This article is devoted to the methods of designing running gear in compliance with recently changed requirements, regulations of rail authorities, technical specification for interoperability, and safety certification. Changes are focused on the certification process. According to ERA these changes are recommended for bogies holding IC certificate issued according to the former TSI WAG, without assessment of the strength of the bogie frame.

The presented methods allow designing a bogie platform with a wide range of applications in regional, and long-distance trains. Issues related to calculations, and studies of bogie frames are described. Particular attention is paid to methods for determining loads applied to the bogie frame based on assumed axle loads, determining reference forces for the projected shock absorbers, and compiling computational cases for strength analysis of traction units.

1. Introduction

The running gear, as one of the most significant elements of a rail vehicle, having a crucial impact on its functionality, such as safety, ride comfort, etc. must meet a series of requirements. These come from the customers side, railway transport authorities, standards, including the Technical Specification for Interoperability. Despite years of research, experience, and improvements in running gear design, potential directions for further development continue to be identified [12, 13, 16]. In recent years, three revisions to the standard specifying design requirements for bogies have been issued [24–27]. Therefore, the design process is extremely complex. Any changes in requirements that occur during construction carry a significant risk of deteriorating the vehicle's technical parameters. A good example is the thorough analysis of parameters essential for the running gear design, which has been addressed in a publication describing the influence of bogie frame stiffness on the running properties of rail vehicles [4].

To meet all requirements, taking into account the complexity of running gear design issues, it would be worth considering a method of designing running gear that ensures high product quality while providing high resistance to changes, allowing the designer the greatest possible freedom in choosing technical solutions when configuring the vehicle, especially the running gear. Examples of analyses focusing on the optimization of selected parts of the rail vehicle were studied [3, 5, 7].

Designing new constructions that are not adaptations of existing, proven solutions requires numerous approval procedures, constituting the homologation process. Mass-produced bogie frames undergo static and fatigue strength assessments at multiple stages of the design process, starting from calculations, through destructive testing (static and dynamic) on dedicated test benches, to field tests conducted on prototype vehicles. All these stages must be carried out in accordance with the EN 13749:2021 standard. The methods used during the research and calculations are described in detail in several works [9, 15, 17].

The aforementioned standard defines exemplary, simplified conditions that the running gear must meet to obtain approval for operation. However, due to the general nature of the standard, the described require-

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ments must be adapted to the construction of the specific vehicle in which the tested running gear would be used.

2. Normative requirements

The standard provides a general description of data for load cases directed at specific, commonly used solutions. The base vehicle is a wagon with evenly distributed load on both bogies. However, it does not take into account structural differences in the bogie suspension or active systems used in running gear. Many articles [9, 19] state that standard requirements are insufficient.

The loads acting on the bogie described in the standard [26] for category BI/BII vehicles are divided into exceptional loads occurring in operation (subject to static strength testing), and normally occurring loads in operation, i.e., those assumed in dynamic fatigue testing. Both groups of loads depend on the following vehicle mass components:

– M_V – vehicle mass in working order
– P_1 – exceptional payload
– P_2 – normal payload
– m^+ – bogie mass.

The standard divides the loads into groups of forces acting on the frame resulting from the bogie's free movement, acceleration, braking, action of the anti roll bar, bogie yaw dampers, as well as inertia effects of masses attached to the frame.

The analysis aims to determine the forces resulting from the bogie's movement in such a way as to ensure the bogie's versatility in various configurations, conditions, and even types (EMU, DMU, BEMU, etc.). It is obvious that the technical solutions of DMU and EMU vehicles differ significantly in construction, but they may still share bogie equipment elements, such as the sideframe with brackets.

2.1. The loads resulting from the operation of viscous dampers

A very useful and valuable part of the standard [24] edition from 2005 was the provision of typical velocities for shock absorbers to estimate the forces acting on the bogie brackets, which were respectively:

– 0.1 m/s for lateral shock absorbers
– 0.15 m/s for vertical shock absorbers
– 0.3 m/s for primary vertical shock absorbers
– 0.0026 m/s for yaw dampers.

The recommendation at that time was to use double reference force for exceptional cases occurring in operation. However, for fatigue strength analysis, the reference value was to be directly adopted. An example cycle of vertical displacement of secondary suspension with an amplitude of 10 mm was provided on the Fig. 1, corresponding to the yaw damper stroke reaching a maximum velocity of 0.0025 m/s.

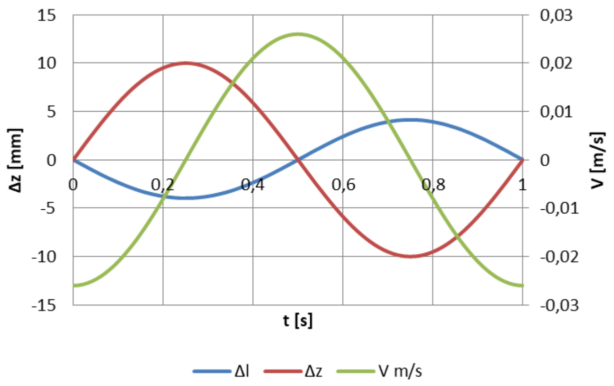

Fig. 1. The displacement and velocity of the yaw damper due to the vertical displacement of the secondary suspension

Yaw dampers are sensitive to such displacements, therefore, the value given in the EN 13749:2005 standard was certainly underestimated and can be replaced with a value corresponding to the maximum displacement velocity of the damper achieved, for example, when passing through a curve with a radius of R = 150 m. In this case, a vehicle with a body base of 19,000 mm and a bogie base of 2,500 mm, passing through a curve with a radius of R = 150 m with a maximum unbalanced lateral acceleration of about 0.7 m/s^2, the yaw damper will reach a maximum velocity of 0.2 m/s, thereby significantly exceeding the reference force. An example characteristic of the yaw damper is presented in Fig. 2.

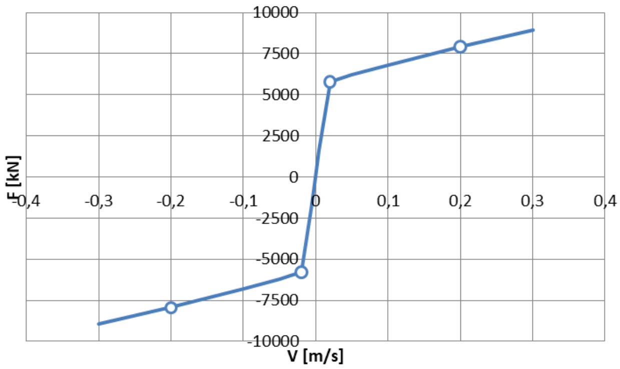

Fig. 2. An example yaw damper characteristic

The sample forces provided in the 2005 standard for strength analysis can be estimated based on the reference force, but they must always be multiples of it. The reference force of the shock absorber depicted in Fig. 2 is ±6 kN, whereas in the discussed case, the shock absorber will reach a damping force of ±7.95 kN. For this particular example, the coefficient would be 1.3. However, it is important to remember that the determined coefficient applies to a specific case.

A coefficient encompassing all shock absorber configurations may reach a higher value.

2.2. Formation of computational cases

The generalized approach presented in the standard does not specify a clear procedure for motor bogies. Annex F of the standard [25] only suggests simultaneous application of vertical and longitudinal forces. Such a method of applying loads is the most common approach in formulating computational cases, i.e., separately applying forces to the frame resulting from acceleration, braking, or operation of the lateral stability system, etc. However, it should be noted that longitudinal, twist, and lateral loads can occur simultaneously during operation. This results in the superimposition of stresses in the area of load-bearing brackets, along with stresses resulting from the twist of a bogie. In the case of structures sensitive to such phenomena, especially those containing multiple stress concentration areas, distributing computational or test loads into separate components may lead to a situation where the stress level in the bogie is underestimated. Example of this kind of design is described in [20]. In such a case, it is worth considering the simultaneous action of forces caused by ARB, acceleration, twist, and curve negotiation. An example set of loads is included in Table 1. In reference [8] to establish superimpose of load cases during on track tests, a couple of bogie components were transformed into load sensors.

Table 1. Compilation of computational cases considering simultaneous action of vertical, lateral, longitudinal and twisting forces

Loadcase	Fz1	Fz2	Fy1	Fy2	Fs1	Fs2	Fx1	Fx2	Twist
1	1.0	1.0	1.0	1.0	1.0	–1.0	1.0	1.0	1.0
2	1.0	1.0	1.0	1.0	1.0	–1.0	1.0	1.0	–1.0
3	1.0	1.0	–1.0	–1.0	–1.0	1.0	–1.0	–1.0	1.0
4	1.0	1.0	–1.0	–1.0	–1.0	1.0	–1.0	–1.0	–1.0
5	1.0	1.0	1.0	1.0	1.0	–1.0	1.0	1.0	1.0
6	1.0	1.0	1.0	1.0	1.0	–1.0	1.0	1.0	–1.0
7	1.0	1.0	–1.0	–1.0	–1.0	1.0	–1.0	–1.0	1.0
8	1.0	1.0	–1.0	–1.0	–1.0	1.0	–1.0	–1.0	v1.0

where: Fz1, Fz2 – vertical forces applied to the secondary suspension; Fy1, Fy2 – lateral forces applied to the wheelset axis; Fs1, Fs2 – forces applied to the anti-roll bar links; Fx1, Fx2 – forces applied to the drive brackets; Twist – displacement applied support of the frame.

The disadvantage of the compilation above is simulating the use of full traction force for fatigue strength analysis, which does not reflect operational conditions. Mostly the Palmgren-Miner method for estimation of cumulative damage is used in that field [17]. For general purpose, specially selected sample coefficients are used to reduce the traction force, as shown in Table 2. On the other hand in publication [21] various simulations were conducted which describe more accurately real conditions than those defined in [27].

Table 2. Compilation of example computational cases which provide a representation of the working conditions of a motor bogie

Loadcase	Fz1	Fz2	Fy1	Fy2	Fs1	Fs2	Fx1	Fx2	Twist
1	1.0	1.0	1.0	1.0	1.0	–1.0	0.5	0.5	1.0
2	1.0	1.0	1.0	1.0	1.0	–1.0	0.5	0.5	–1.0
3	1.0	1.0	–1.0	–1.0	–1.0	1.0	–0.5	–0.5	1.0
4	1.0	1.0	–1.0	–1.0	–1.0	1.0	–0.5	–0.5	–1.0
5	1.0	1.0	1.0	1.0	1.0	–1.0	0.5	0.5	1.0
6	1.0	1.0	1.0	1.0	1.0	–1.0	0.5	0.5	–1.0
7	1.0	1.0	–1.0	–1.0	–1.0	1.0	–0.5	–0.5	1.0
8	1.0	1.0	–1.0	–1.0	–1.0	1.0	–0.5	–0.5	–1.0

3. Course of examination

The values of forces applied to the frame are of paramount importance in conducting strength tests. Equally significant is the method of frame loading. Below are commonly used methods developed based on standard [25]. Among the factors determining the success of validation of design assumptions in field tests, several mutually complementary aspects should certainly be highlighted:

- Preservation of all degrees of freedom of the bogie frame is crucial. Particularly important in this regard is replicating the twist behavior of the bogie frame.
- Precise placement of force application points is essential to avoid over-stiffening the frame in any loading scenario.
- The course of forces applied to the frame.

3.1. Dynamic tests

The example course of dynamic loads applied to the bogie frame is clearly described in Annex F of the standard [25].

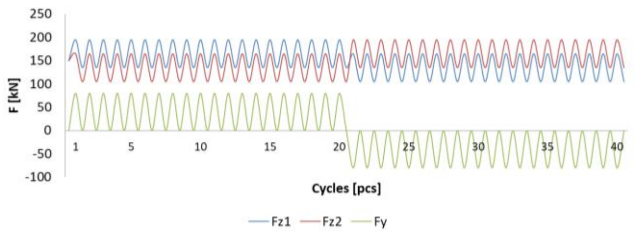

Fig. 3. An example of the course of dynamic forces during fatigue testing

While the matter is straightforward for vertical and lateral loads, the cycles of bogie twist are described in a way that leaves room for interpretation. Examples of bogie twist cycles are illustrated in Fig. 4–6.

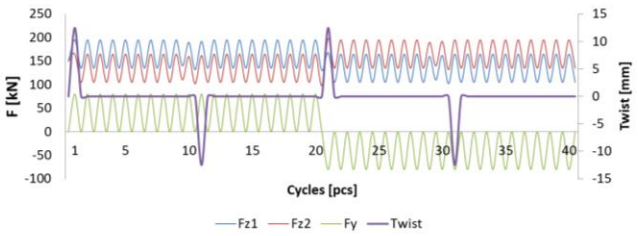

Fig. 4. The triangular cycle of bogie twist in the form of a peak coinciding with the frequency of vertical force cycles

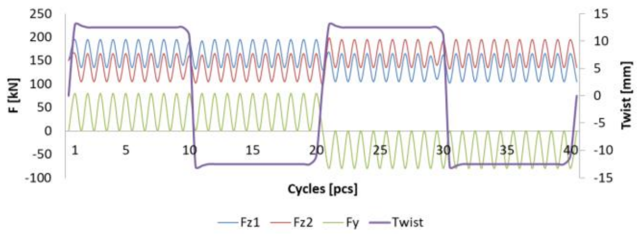

Fig. 5. The trapezoidal cycle of bogie twist

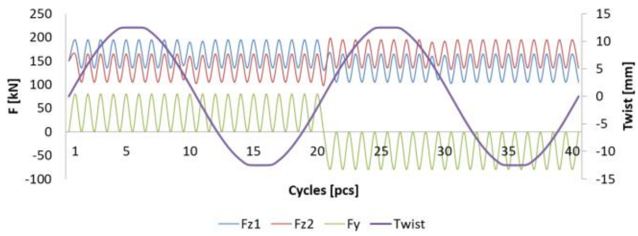

Fig. 6. The sinusoidal cycle of bogie twist

All of the above waveforms are successfully used in fatigue testing. A factor in favor of the displacement waveform from Fig. 6 is the low influence of the twisting load on other forces. In dynamic testing conditions, introducing twisting displacement in one support can disrupt the control system of the remaining actuators, potentially leading to increased amplitudes of vertical forces in the initial cycles of direction change of the twisting bogie support.

3.2. Static tests

In the case of static tests, differences in measured stress values are observed depending on the method of applying load to the bogie frame. Figures 7–10 illustrate examples of methods used in testing.

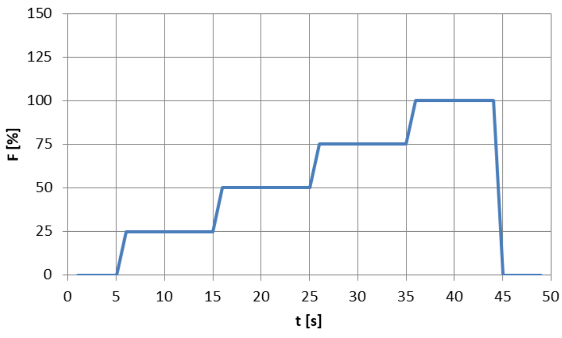

Fig. 7. The course of force application expressed as a percentage of force in time

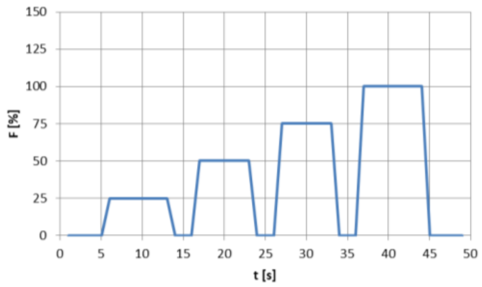

Fig. 8. The course of force application expressed as a percentage of force in time

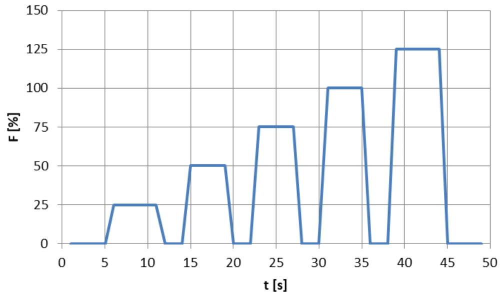

Fig. 9. The force application profile expressed as a percentage of force in time

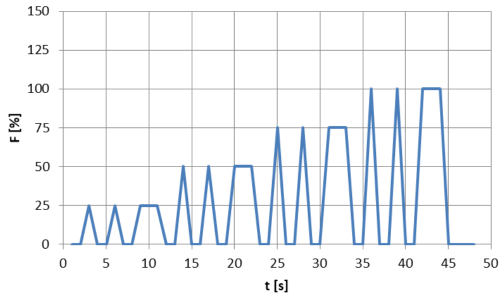

Fig. 10. The force application curve expressed as a percentage of force in time

Figures 7–10 illustrate sample load profiles on the bogie frame in several steps: 25%, 50%, 75%, 100%. The most justified profile is shown in Fig. 8. It stands out for its simplicity. Such a profile can be applied for any force exerted on the frame. The load profile in Figure 10 is shaped to overcome the resistance of the fixtures and the associated hysteresis. However, this is a time-consuming and complicated method when applying load in multiple axes simultaneously. The method presented in Fig. 8, on the other hand, refers to the process of stress relief [14]. Stress relief is applied to reduce residual stresses in the bogie frame. The application of this method should not be limited only to the examined piece. It should be noted that the entire series of produced bogies is covered by a quality plan, and the method of stress relieving in the bogie frame is a very important part of it. It is also recommended to have breaks in applying loads to verify whether the bogie frame has undergone plastic deformation under the assumed load.

4. Standardization of requirements

In the case of analyzing the strength of a bogie designed for a BI/BII category two-car vehicle with the Bo'2'2'Bo' or Bo'Bo'2'2' axle arrangement, there is a possibility of designing the same frame for the motor and trailer bogies. This entails a number of benefits while simultaneously oversizing the frame of one of the bogies. A similar situation occurs when considering a design based on Jakobs bogies, where shared elements such as axle bearing housings and wheelset guiding components can be common for both bogies. The necessity of standardizing force values also arises from the possibility of freely adapting the running gear between individual tenders in the domestic or foreign market.

The motivation for undertaking work to seek standardized loading conditions also stems from the desire to break the dependence between the mass management sheet and the assumptions for the running gear project. The mentioned breaking of dependence is possible only when the load assumptions are applied over a wider mass range than that indicated in the mass management sheet. Such an approach allows for parallel design of the running gear while simultaneously developing the body equipment concept. Standardizing the loads applied to the elements of the running gear covered by the standard [25] will allow for easy servicing of the vehicle construction and expand the scope of subsequent modernizations without the need for costly strength analyses of the bogie. The search for standardized loads acting on the bogie is based on the analysis of axle loads, due to the wide range of applications of this parameter. The axle load is a crucial piece of information, a basic vehicle property mentioned at the stage of ordering the trainset. Directly relating strength calculations to axle loads provides unambiguous information for adapting selected solutions within the running gear project to analogous calls or tenders and orders with a defined maximum axle load. Additionally, the axle load serves third parties while exchanging information between suppliers of individual elements and subassemblies of the running gear. The axle load also serves as a criterion for permitting access to infrastructure and is the basis for calculating fees charged for access to infrastructure.

To properly assume the maximum force values necessary for analyzing the strength of the running gear, the application of which is not fully determined, the following formulas can be used. This also guarantees a stable design that does not require additional analyses, calculations, or even tests when the loads defined in the design phase prove to be insufficient. To avoid a situation where the modernization of a railway vehicle requires simultaneous modernization of the bogie frame, it is worthwhile to relate the loads acting on the bogie to the maximum axle loads provided for the vehicle. In this way, a design is created that is resistant to any changes resulting from the installation of components with a mass that exceeds the assumptions provided in the project. For example, when modernizing the vehicle or changing its type. In other words, it is possible to determine basic loads that allow for the use of universal components that can be used in different vehicles.

5. Calculating the forces acting on the bogie based on defined axle loads

In the formulas provided below, the position of the wagon's center of gravity along the x-axis in the vehicle or the percentage of the load on individual bogies, as was the case in the UIC [21, 23] cards, are irrelevant. When applying the proposed formulas, only the axle loads for each bogie in the composition and the mass of the bogie undergoing analysis are taken into account.

5.1. Exceptional loads occurring during operation

To determine vertical force applied to secondary suspension, using axle loads as a reference following formula should be used:

$$F_{Z1max} = \frac{1.4g(2N_{MXD} - m^{+}_{min})}{2} \tag{1}$$

where: N_{MXD} – axle loads under exceptional payload, m^{+}_{min} – This is the bogie configuration distinguished by the minimum mass among all available options, g – acceleration due to gravity.

The same value can be expressed as Prud'homme criterion:

$$F_{Y1max} = F_{Y2max} = 10^4 + \frac{gN_{MXD}}{3} \tag{2}$$

Lozenging forces:

$$F_{X1} = F_{X2} = 0.2 \cdot gN_{MXD} \tag{3}$$

5.2. Normal loads occurring during operation

To determine the reference force F_z under normal conditions, the following formula should be used:

$$F_{Z1} = F_{Z2} = \frac{F_Z}{2} = \frac{(6N_{MND} - N_{MVD} - 2.5m^{+}_{min})g}{5} \tag{4}$$

where: N_{MVD} – axle loads in running order, N_{MND} – axle loads under normal payload, m^{+}_{min} – This is the bogie configuration distinguished by the minimum mass among all available options, g – acceleration due to gravity.

The formula for transverse loads normally occurring in operation remains unchanged, due to its connection with the force F_z

$$F_{Y1} = F_{Y2} = \frac{F_Z + gm_{min}^{+}}{8} \quad (5)$$

provided by [25].

Lozenging forces:

$$F_{X1} = F_{X2} = 0.1 \cdot gN_{MND} \quad (6)$$

Assuming a fixed level of axle loads, the above transformation of formulas implies that the determining factor for the load applied to the bogie frame is its mass. To illustrate the idea more clearly, load calculations were performed for various input data, as presented in Table 3.

Table 3. The input data for analyzing the unified forces acting on the frame of the bogie

Parameter	Max.	Min.	Step
	[t]		
MV	36	60	2
P1	10	22	2
P2	2	22	2
M+	6	11	0.5

The analysis results in 22,000 combinations of input parameters, indicating that, from the perspective of assumed axle loads, the determining factor influencing the load is the bogie's mass. Figure 11 illustrates the vertical force applied to the bogie relative to the axle load. Bogies subjected to the same force can be used over a wide range of axle loads. The factor determining the width of this range is the difference in mass between the bogies (motor and trailer). The trailer bogie may reach axle loads of around 11.5 tons per axle, whereas the motor bogie may reach up to 14 tons per axle.

The conclusion drawn from the above analysis is the necessity of shaping the bogie systems in such a way that components shared between the motor and trailer bogies achieve the lowest possible mass while maximizing their load capacity. This places the trailer bogie in the role of dimensioning the project for system solutions in the vehicle family.

The best example of applying such an approach is seen in the Stadler Flirt 3 family of vehicles. The trailer and motor bogies share a number of suspension components, however, they differ significantly in mass due to the use of larger diameter wheels on the motor bogies (920 mm) compared to the trailer bogies (760 mm) [2]. The heavy, approximately 3-ton drive system is implemented through a fully suspended gearbox. As a result, the bogie frames differ structurally, with the trailer bogie frame being open and the motor bogie frame being enclosed with a front headbeam.

6. Summary

Typically, scientific papers focus on the issue of fatigue strength [1, 6, 11]. On the other hand, the remaining ones describe the procedure for conducting calculations or research [9, 15, 17]. The methods presented in this article facilitate the process of designing and testing new structural solutions. The aim of the described methods is to design the drive system based on defined fundamental parameters independently of the mass management sheet.

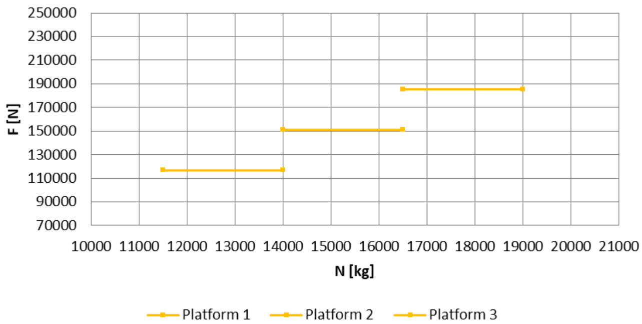

Fig. 11. Sample distribution of axle loads for platform bogies of various applications with a set load acting on the bogie

The analyses conducted indicate that to assess the fatigue of yaw damper brackets, the reference force value for the presented yaw damper bracket should be increased by 33% to accurately reflect its operation without the need for additional testing or simulation.

The standard specifying requirements for designing bogies often requires clarification of the loads acting on the bogie. A method has been proposed to replicate the loading during acceleration or braking of the train in a curve. This allows the stresses calculated or examined in the frame to be subjected to the superimposition of cases, thereby increasing the level of safety and reliability of the running gear system.

Various methods of applying loads to the bogie frame are presented in this article, with preferred approaches indicated based on their time efficiency and influence on other forces applied to the structure subjected to multi-axial loading conditions.

The design of bogies for broad application in long-distance and regional vehicles is complemented by the definition of forces applied to the bogie. It is helpful to determine forces based on axle loads, and under these assumptions, it has been noticed that the design of platform solutions for running gear systems aimed at component sharing should primarily focus on the analysis of the trailer bogie.

The topic of all issues related to the assumptions for projects with wide applications has not been completely covered and will be the subject of interest for authors in subsequent publications.

Nomenclature

ARB anti roll bar
MND mass of a vehicle under normal design payload
MVD design mass in working order
MXD mass of a vehicle under exceptional design payload

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