Proof tests of geometric-kinematic calculations of railway vehicles

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Geometric-kinematic calculations and experimental proof tests of the mutual position of assemblies and sub-assemblies of wagons that are in relative displacement are significant in developing new types of railway vehicles and especially in the application of non-standard solutions. In the case of standard types of wagons and standard solutions, compliance with the recommended geometric dimensions and parameters of the vehicle should guarantee that even in extreme positions there will be no irregular contact between moving parts of the rail vehicle. This is especially important if it is necessary to place some elements or parts of the wagon's equipment in the zone of wheelsets and in the zone of the connection between the bogies and the car body. This check can be carried out using graphic-analytical methods, according to the methodology given in the relevant EN standards, ORE reports, and AVV publications, and the limit values of the angular and translational coordinates of the moving parts. Verification of these calculations and verification of the derived state can be done by simulating these movements on the vehicle prototype itself or using a mock-up. This paper presents limit values and proof tests, for checking the mutual position of the running gear and the vehicle car body of a tank wagon car type Zacns. The test was performed using a transfer table for simulating the rotation of bogies relative to the car body in the horizontal plane and by using pads placed under wheels for simulating rotation in the vertical plane. The check was performed by visual inspection and by distance measurements.

Keywords: Geometric-kinematic calculations, Proof tests, Mutual position of the running gear and the railway vehicle car body

1. INTRODUCTION

Geometric-kinematic calculations serve for checking the limitations of the adopted dimensions of the wagons, which are a consequence of possible movements of the wagons in relation to the track, relative movements of two wagons in the composition, and relative displacements of unsprung, primarily suspended, and secondary suspended parts of the structure. These calculations include:

- verification of the profile (dimensions) of the vehicle,
- checking the mutual position of the running gear and the car body (wheel vs.car body, car body and bogie frame),
- determination of the required dimensions of buffers plates and checking the possibility of passing through the curves,
- checking the mutual position of the draw-buff gears and the car body (hook or automatic coupler head space in sharp curves and vertical track curves),
- checking the mutual position of the two wagons and their gangways in the curve (for wagons with gangway systems).

For the unhindered movement of rail vehicles and avoiding contact with stable structures (tunnels, cuts, poles, etc.), space around the track should be provided. This space can be completely defined by some boundary contour - the reference profile (or dimensions) in the transverse plane in relation to the track and the rules, by which, starting from the reference profile, the permitted profile of the rolling stock is determined on the one hand, and on the other hand the free profile of the stable structure.

In the world, there are many different profiles in different countries and even multiple profiles on different railways in the same country. For the European standard track gauge, the EN [1] has defined a reference profile, as well as a procedure by which the vehicle profile is calculated from. This profile was obtained by analyzing the railways involved in international traffic and is an internal envelope of all profiles on these lines. Thus, the unhindered exchange of railway vehicles between all member states of the UIC was ensured.

Figure 1 shows how starting from the reference profile, by introducing the required movements of the wagon in the vertical plane for the values of Δh and narrowing E, which are the result of the movement of the wagon in the track and passing through the curve (Figure 2), the permissible profile of the vehicle is reached.

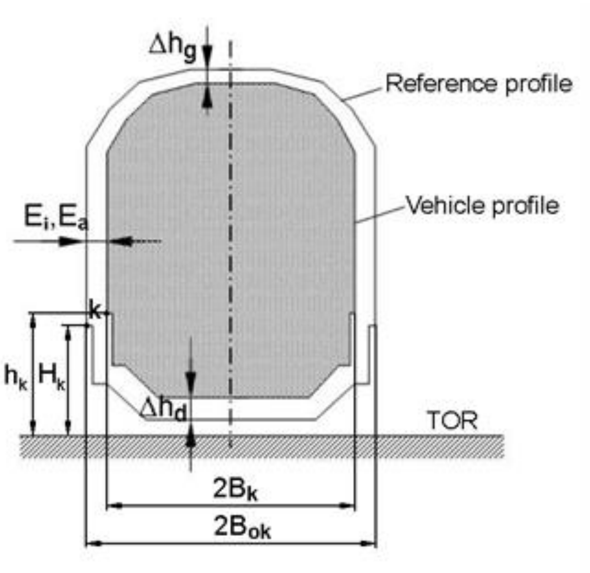

Figure 1: Reference profile and vehicle profile [1, 2]

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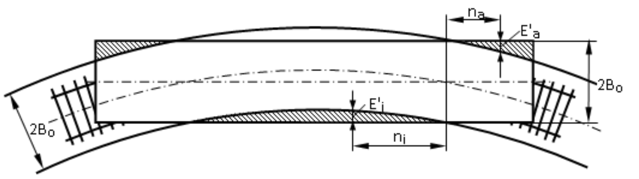


Figure 2: Geometric deviation of the vehicle and the track axes during passing through a curve [1, 2]

Calculations related to checking the undisturbed movement of draw-buff gear elements and gangways of the vehicles during vehicle movement are not in the scope of this paper.

Significant displacements of wagon sub-assemblies occur when crossing the marshalling humps, passing through sharp horizontal curves in combination with possible railway twisting, and when loading onto the ferry boats, i.e., when crossing the loading ramp. This paper presents an analytic procedure for calculating possible movements of the running assembly in relation to the car body, as well as a presentation of an experimental check with a tank wagon Zacns for the transportation of petroleum products at the most unfavourable combination of maximum deflections with rotation of the bogies in relation to the car body in both, horizontal and vertical, plane. Verification of these calculations and verification of the derived state can be done by simulating these movements on the vehicle prototype itself or using a mock-up.

2. CHECKING THE MUTUAL POSITION OF THE RUNNING GEAR AND THE CAR BODY

2.1. Criteria for an estimation of wagon capability to use on a ferry boat

The angles of rotation of the bogies in the vertical and horizontal plane, which are decisive for the use of wagons on the ferries, are given in Table 1 and calculated for the tank wagon, using its dimensions and designed gaps and plays. These angles serve for checking the possibility that some moving elements get into the irregular contact.

For calculation and testing, it is necessary first to determine the maximum values of possible displacements that can occur during exploitation, which are defined in the relevant EN standards [1], ORE B12/DT 135 [3], and General Contract for Use of wagons (AVV), Appendix 14 [4]. These values are defined for all ferry lines involved in the Trans-European Transport Rail network.

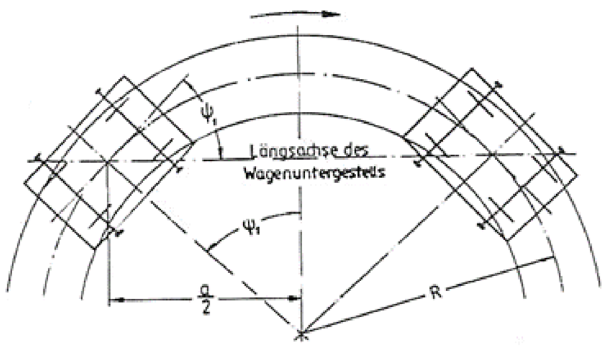


Figure 3: Rotation of the bogie vs. car body in the horizontal plane during passing through the curve [3, 5]

Figure 3 shows the curve negotiation of the 4-axle wagon with bogies and the resulting bogie vs. car body rotation.

Figure 4 presents the rotation of the bogies relative to the car body in a vertical plane during entering the ferry boat in the two critical positions. First, when entering the ramp slope, and the second position, when the first bogie enters the horizontal boat floor, with the second bogie still on the ramp.

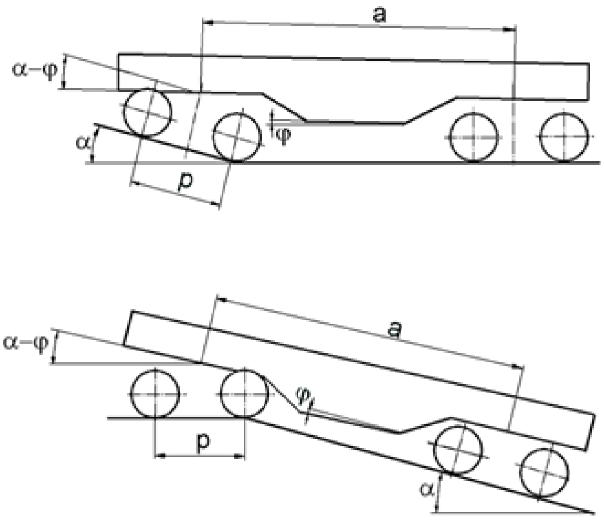


Figure 4: Rotation of the bogie vs. car body in the vertical plane during entering the ferry boat [3, 5]

Figure 5 presents the possible rotation of the bogie frame as a result of the existing wheel set vs. track play.

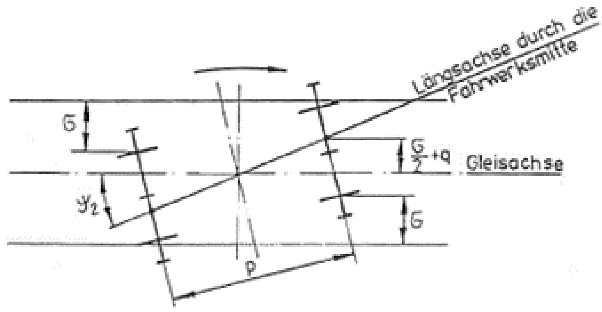


Figure 5: Rotation of the bogie due to the wheelset vs. track play [3, 5]

In the event of irregular contact, parts may be damaged, as well as functionality may be disrupted, and traffic safety may be compromised.

Because of all this, some Notifying Bodies require that, in addition to the calculation, an experimental check of the mutual position is also carried out.

The angle of the ferry boat ramp may vary from 2.5 to 3.0° depending on the boat type.

The tank wagon is equipped with a standard Y25Lsd bogie with known dimensions and clearances of the running gear parts.

Table 1: Required limit values depending on the wagon dimensions [3]

Parameter	Abbreviation	Equation	Value
The angle of the ferry boat ramp	α		2.5°
Tangent of the vertical kink angle of the ferry boat ramp	$tg\ \alpha$		0.0437
Distance of the pivots	a		11.36 m
Bogie base	p		1.8 m
Vertical bogie inclination	φ	$\sin\varphi = \frac{p \cdot \sin\alpha}{2a}$	0.0035 rad=0.20°
Bogie deflection vertical	α- φ		2.3°
Track width (according to EN 15273-4)	$2b_o$		1.465 m
The track dimension of the wheel sets 10 mm under the running rolling radii	2b		1.41 m
Wheelset vs. track play	$2b_o – 2b$		0.055 m
Lateral play of the wheel sets vs. bogie frame	q		0.011 m
Curve radius	R		120 m
The horizontal turning angle of the bogies in the curve	ψ_1	$\sin\psi_1 = \frac{a}{2R}$	2.71°
Horizontal turning angle by adjusting the wheel sets in the track and possible lateral displacement of the wheel sets in relation to the wheel set guides and stops	ψ_2	$\sin\psi_2 = \frac{2(\frac{\sigma}{2}+q)}{p}$	2.45°
Yaw angle of the bogie relative to the wagon underframe in the curved track R	ψ_3	$\psi_1 + \psi_2$	5.16°

2.2. Testing on the transfer table

These conditions were achieved using a transfer table for simulating the rotation of bogies relative to the car body in the horizontal plane and by using appropriate pads placed under wheels for simulating rotation in the vertical plane. Table 2 presents calculated values of pads height and transfer table lateral displacement, that correspond to possible bogie and car body relative movement, according to [1, 2 and 3]. The bogie suspension springs were removed, and the bogie frames supported at the spring stops, to simulate the maximum possible vertical spring travel.

Figure 6: Rotation in the vertical plane bogie vs. car body for an angle α-φ= +2.3° [5]

The vehicle was placed on the transfer table with the first bogie.

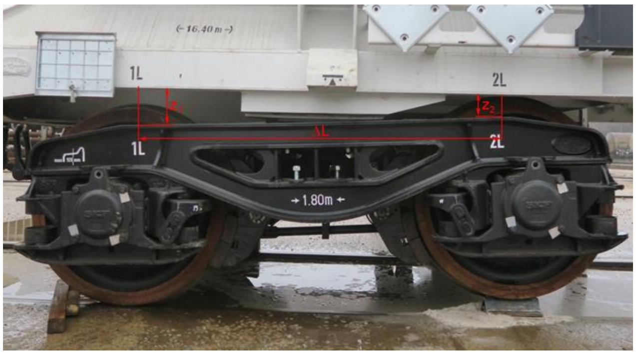


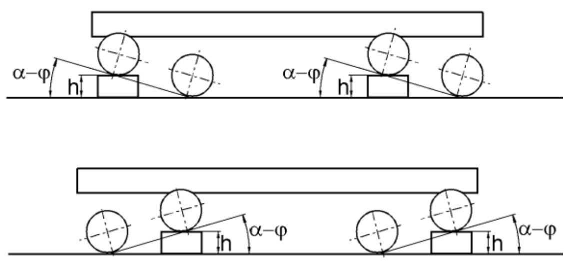
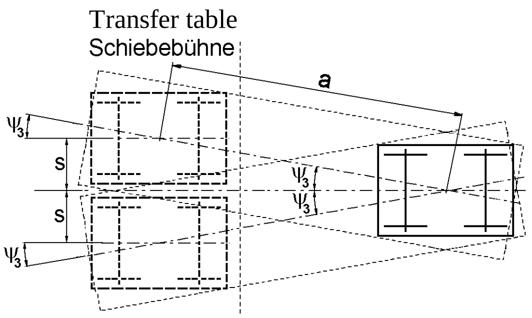
Figure 7: Rotation in the vertical plane bogie vs. car body for an angle α-φ= -2.3°[5]

Underneath the wheels of the first and the third wheel sets, the pads of height h = 72 mm were placed, as presented in Table 2, to achieve the angle α-φ= 2.3° between the bogies and the wagon underframe (Figures 6 and 7). In addition, the influence of the deviation of the wheel-rail contact points from a plane was checked by measuring the height differences between two points on the plane surface of each bogie frame longitudinal beam and the longitudinal beam of the car body (z_1 and z_2), as shown in

Figure 7. If necessary, the correct α-φ angle was achieved with additional shims of 1 mm thick sheets.

After that, the transfer table was carefully moved by distance s=1150 mm, slightly more than shown in Table 2, until an angle slightly larger than $+\psi_3$ is reached, Figure 8.

Table 2: Height of the pads and transfer table displacement, chosen to simulate possible displacements [5]

Case	Parameter	Abbreviation	Equation	Value
	Height of the pads for vertical displacement	h	$h=p\cdot \sin(\alpha-\varphi)$	72 mm
	Transfer table lateral displacement	s	$s=a\cdot \sin\psi_3$	1022 mm

After that, the transfer table was carefully moved by distance s=1150 mm, slightly more than shown in Table 2, until an angle slightly larger than $+\psi_3$ is reached, Figure 8. The process was repeated by moving the transfer table in a different direction until the angle $-\psi_3$ was slightly exceeded, Figure 9.

Figure 8: Transfer table displacement to simulate rotation angle ψ>5.16° [5]

The pads are then placed under the wheels of the second and fourth wheel sets to achieve the angle -(α-φ) between the bogies and the wagon underframe.

The process was repeated by moving the transfer table in both directions.

Figure 9: Transfer table displacement to simulate rotation angle ψ<-5.16°[5]

2.3. Test results

It has been observed that there is no contact between the bogie frame and wheels on the one hand and the underframe of the car body on the other.

During the tests, no clearance of less than 20 mm was found between the bogie frame and the superstructure.

The distance between any wheel and wagon superstructure was at least 30 mm.

Figures 10 to 14 show some examples of the clearance between bogie parts and the wagon superstructure in different critical positions. The closest to the wheel is the brake pipe.

Figure 10: Running gear vs car body relative position 1

Figure 11: Running gear vs car body relative position 2

Figure 12: Running gear vs car body relative position 3

Figure 13: Running gear vs car body relative position 4

Figure 14: Running gear vs car body relative position 5

3. CONCLUSIONS

The tests carried out, with the aim to prove the relative position of the wagon moving parts, have shown that, under prescribed conditions, there is no contact between the wheels, parts of the bogie, and the car body superstructure. The distance between any wheel and superstructure was at least 30 mm. The tested tank wagon meets the requirements for use on the ferries according to ORE B12/DT135 [3]. This check using calculations and experiments is significant in the design phase of a new type of railway vehicle and especially in the application of non-standard solutions of the running gear and car body superstructure.

ACKNOWLEDGEMENTS

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