Method of Static Determination of the Safety against Overturning of the Road – Rail Machines

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Abstract: - In the present work, it is shown an overturning safety statically procedure for a road – rail machine, with a dual road and rail rolling system. It is analyses the rail rolling variant of the machine, when it will be deemed as an railway motor vehicle

Key-Words: - Road-rail machine, overturning, wheel load, super-elevation

1 Introduction

Most of the studies performed up to the present time, refer to safety running of rolling stock designed for passengers and freight trains, in standard operating conditions, under a strict schedule.

This rolling stock usually runs with high speed, the risk of appearing running safety disorders increasing when the speed increases.

Road-rail machines which are designed for planned or unplanned maintenance interventions over railway infrastructure. These interventions are always made when the traffic is closed. Generally, road-rail machines are running at low speed. However, running safety disorders still could occur.

The aim of the present study is to experimentally establish a method of statically determining the safety against overturning of road-rail machines, based on the analysis of the load distribution on the wheel in rail configuration of road-rail machines. Thus, an important factor in order to ensure road-rail machines safety against overturning is to ensure the load per wheels in certain limits.

2 Load distribution on pair of wheels

As generally known, a pair of wheel of a railway vehicle, usually known as independent rotating wheels [1], is charged in static condition with a load acting on vertical direction, which is partially due to own weight, known as load per wheels $2Q_o$. In case of a uniform distribution of the load per wheels, it results a load per wheel defined Q_o. In fact, a railway vehicle wheel in running condition is charged both with the load per wheel Q_o and with the loads caused by the rolling dynamical forces.

The dynamical loads are produced by the external forces transmitted through the suspension on the wheels mainly resulting from wheel/rail surface irregularity as well as from continuous change of the attack angle in case of the driving pair of wheels. These dynamical loads are also produced by the longitudinally forces acting on the wheels, caused by the inertial forces of the chassis during the hunting movement. Additionally, the super-elevation of the outer rail in curves generates different loads on the outer/inner wheel. Thus, actually, the loads on the outer/inner wheels will be defined with Q_L and Q_R, caused by theirs loading and unloading.

Half-difference between the loads acting on the pair of wheels is named "load transfer":

$$\Delta Q = \frac{(Q_L - Q_R)}{2} \tag{1}$$

Conventionally, ΔQ is considered as having positive value (+) if the wheel is loaded, and negative value (–) if the wheel is unloaded. [2]

The wheel unloading factor is defined as the wheel ability to unloading, without altering the vehicle safety against overturning:

$$\Delta q_{LR} = \frac{\Delta Q}{Q_o} = \frac{(Q_L - Q_R)}{2Q_o} \qquad (2)$$

This ratio must be less or equal with 0,6 so as to insure the minimum safety requirements against railway motor vehicles overturning when running through small radius curves, over the switches or at the level track crossings, according to last European railway regulations, referring to the testing of the running behavior of the railway vehicles. [3]
Regarding the running through curves, because of the un-compensated centrifugal forces, the road – rail machines have an additional load transfer ΔF_o on the loading wheels, which depends on the road – rail machine running speed.
The road – rail machines could be assimilated with freight wagons having a high center of mass of the primary vertical suspension. Similarly with other railway vehicles, the road – rail machine passing through the curve inclines with regards to the track, the angle of inclination being φ_b, the following counteraction forces appearing in the primary suspension springs:

$$\Delta F = 2c_z^+ \times b^+ \times \varphi_b \qquad (3)$$

where:

c_z^+ = primary suspension spring stiffness [kN/mm];

b^+ = transversal distance between the axis of the primary spring suspension

Most of freight wagons on bogies or two axle wagons are not equipped with divergent swing links in the suspension. However, there are cases of rolling stocks with vertical divergent swing links having the length λ supporting the vehicle body carrier, the counteraction forces into the primary suspension will consequently be:

$$\Delta F = \frac{G_c}{2b^+}(h_c + \lambda)(1+S)\frac{\gamma_{TO}}{g} \qquad (4)$$

where:

G_c = body weight;

h_c = center of mass height;

γ_{To} = acceleration in the transversal direction of the truck;

g = 9,81 m/s^2;

S = coefficient of flexibility

In this case, the coefficient of flexibility will be:

$$S = \frac{1}{\left[\frac{4c_z^+ (b^+)^2}{G_c(h_c + \lambda)} - 1\right]} \qquad (5)$$

There are situations when the road – rail machines have no primary suspension. Then:

$$\Delta F = \frac{G_c}{2b^+} h_c (1+S)\frac{\gamma_{TO}}{g} \qquad (6)$$

and:

$$S = \frac{1}{\left[\frac{4c_z^+ (b^+)^2}{G_c h_c} - 1\right]} \qquad (7)$$

3 Determination of stability against overturning of the road-rail machines by static simulation

Referring to the all types of road-rail machines, these must to be verified against overturning. In case of a crane placed on rails in working condition, the movement of the arm involves the crane's load centre modification, which consequence could be the overturning of the crane. The overturning could happen still in absence of any hook load, namely if no proper calculation and stability tests against overturning were made.
Hence, checking the stability of the road-rail machines is a necessary condition to ensure their safety against overturning [4], [5].
In the Laboratory of Rolling Stock of the Romanian Railway Authority was developed a method for statically verification of overturning, so as to determine the stability against overturning of road-rail machines [6].
This method consists from the simulation of the static wheels unloading conditions of a road-rail machine standing in a curve where the super-elevation of the outer rail is 100 mm. The tests were carried out at a temperature of about 25 - 30 ° C, on dry rail.
The road-rail machine has two bogies hybrid sets, each bogie having four wheels symmetrically arranged. The wheel load for the front bogie, near the driver's cab, was Q_o = 6 t and the wheel of the

second bogie, behind the road-rail machine, was Q_o' = 3 t. Also, the corresponding wheels on the two sides of the machine have been independently connected to the bogie frame and actuated by a low-power hydraulic motor, instead of being joined to each other by means of a connecting shaft.
In order to determine the wheel loads, there have been used rail coupons of 120 mm, containing strain gauges and a data acquisition system making possible to process the obtained values in a computer.
At the same time, the simulation of the elevation of the outer rail was achieved by means of rectangular steel prisms which dimensions were 10 mm x 150 mm x 20 mm, 25 mm x 150 mm x 20 mm and 50 mm x 150 mm x 20 mm, and these coupons were interposed between rail and wheel. The wheel-rail contact was made in the point corresponding to the strain gauge rail lateral position. The road-rail machine was raised on one side with winches. To simulate various stages of cant rail, under the wheels was interposed the steel prisms 10, 25 and 50 mm in a successive positioning. The tilting of the road-rail machine was made using winches in order to lift the left side of the machine.
Considering the odd numbering to the left of the vehicle wheels, from wheel number 1 in the left front of the cab driver and the even numbering to the right of the road-rail machine from wheel number 2 in the right front of the cab driver, the load transfer distribution and the wheel unloading factor between the wheels according to cant.
The experimental results were obtained:

Cant [mm]	Load per wheel nr. 1 Q_1[daN]	Load per wheel nr. 2 Q_2[daN]	ΔQ_{21} [daN]	$\Delta Q_{21}/Q_o$ [%]
0	5878,60	6207,18	164,29	2,73
10	5591,35	6299,88	354,26	5,90
25	5670,64	6322,67	326,01	5,43
35	5491,78	6447,24	477,72	7,96
50	5574,54	6432,13	428,79	7,14
60	4999,78	6859,33	929,77	15,49
75	4720,02	7218,93	1249,4	20,82
100	4691,57	7188,11	1248,27	20,80

Table 1

These experimental values obtained are shown in the following graphs:

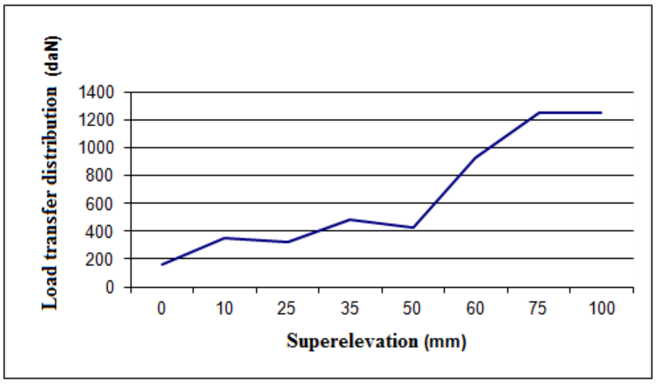

Fig.1: The load transfer distribution between wheels number 1 and number 2

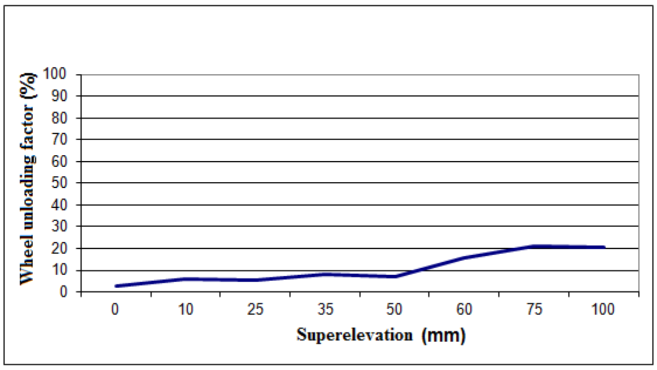

Fig.2: The wheel unloading factor for the wheel number 1

It is to be noted that for a maximum of the rail super-elevation of 100 mm, the discharge of the static load between the two wheels is relatively symmetric. The load on the wheel number 2 increases with about 20%, the load on the wheel number 1 decreases at a rate of 22%. At the same time, the wheel unloading factor is about 1/3 from its maximum of 60%.
Similarly, it can be analyzed the load transfer distribution and the wheel unloading factor between the wheels number 3 and number 4, following the experimental results obtained.
In Table 2 are presented the experimental values obtained for the pair of wheels consisting of wheels number 3 and number 4.

Cant [mm]	Load per wheel nr. 3 Q_3[daN]	Load per wheel nr. 4 Q_4[daN]	ΔQ_{43} [daN]	$\Delta Q_{43}/Q_o$ [%]
0	6486,96	6205,95	-140,50	-2,34
10	6334,93	6279,86	-27,53	-0,46
25	6205,07	6421,66	108,29	1,80
35	6238,10	6426,55	94,22	1,57
50	5747,58	6908,41	580,41	9,67
60	5681,90	7051,23	684,66	11,41
75	5382,56	7283,71	950,57	15,84
100	5077,06	7684,42	1303,67	21,73

Table 2

Below are shown the corresponding graphs for values obtained.

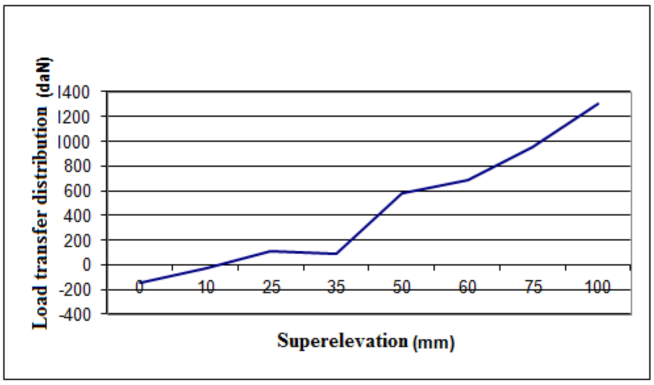

Fig.3: The load transfer distribution between wheels number 3 and number 4

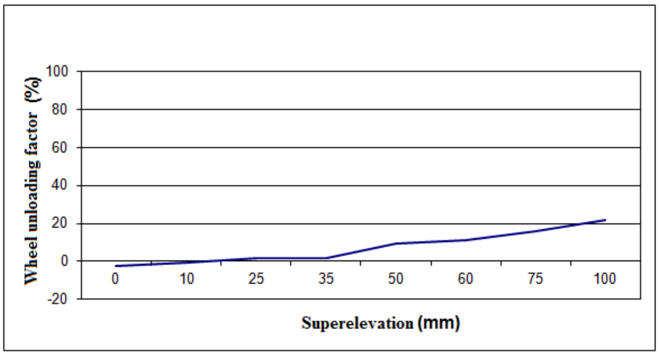

Fig.4: The wheel unloading factor for the wheel number 3

As shown, for a rail super-elevation of 100 mm, the wheel number 4 is loaded with 30% more than the theoretical average wheel load, while the wheel number 3 is being unloaded only with 15% from the theoretical average wheel load.

At the same time, a rail super-elevation between 10 to 25 mm, in particular about 15 mm, the load transfer distribution and the wheel unloading factor have negative values, indicating that an inertia exists in the bogie frame, possibly caused by the constructive torsion of the first bogie.

Forward, are presented the experimental results obtained for the pair of wheels number 5 and number 6 and graphs corresponding to these experimental results.

Cant [mm]	Load per wheel nr. 5 Q_5[daN]	Load per wheel nr. 6 Q_6[daN]	ΔQ_{65} [daN]	$\Delta Q_{65}/Q_o$’ [%]
0	2466,74	2812,81	173,03	5,77
10	2482,40	2922,40	220,00	7,33
25	2348,88	3044,84	347,98	11,60
35	2284,15	3113,24	414,54	13,82
50	2254,75	3090,21	417,73	13,92
60	2366,26	3033,44	333,58	11,12
75	2507,87	2930,16	211,14	7,04
100	2353,53	2988,23	317,35	10,58

Table 3

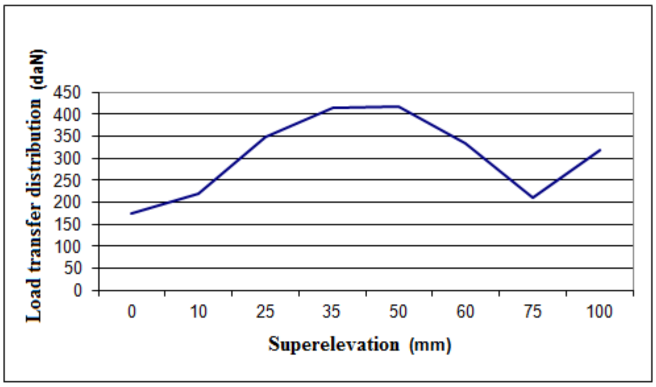

Fig.5: The load transfer distribution between wheels number 5 and number 6

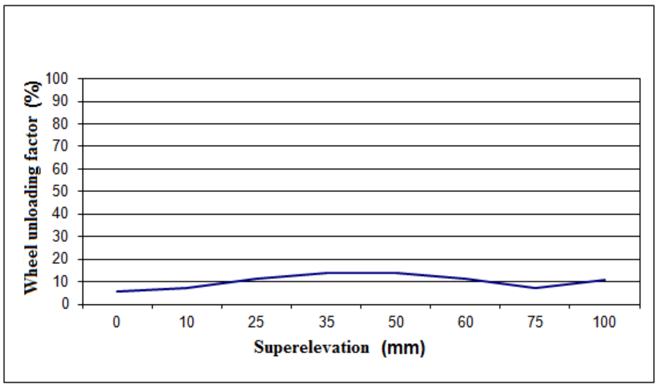

Fig.6: The wheel unloading factor for the wheel number 5

Finally, the experimental values obtained for the last pair of wheels, wheels number 7 and number 8 are shown in Table 4.

Cant [mm]	Load per wheel nr. 7 Q_7[daN]	Load per wheel nr. 8 Q_8[daN]	ΔQ_{87} [daN]	$\Delta Q_{87}/Q_o$' [%]
0	2520.26	2742.56	111.15	3.70
10	2452.85	2862.90	205.02	6.83
25	2413.11	2912.37	249.63	8.32
35	2349.78	2985.17	317.69	10.59
50	2374.10	3000.76	313.32	10.44
60	2555.668	2799.16	121.75	4.06
75	2587.19	2756.37	84.59	2.82
100	2330.34	3102.81	386.23	12.87

Table 4

Also, the load transfer distribution and the wheel unloading factor between the wheels 7 and 8, based on the results of the testing of the road-rail machine, could be observed in the diagrams:

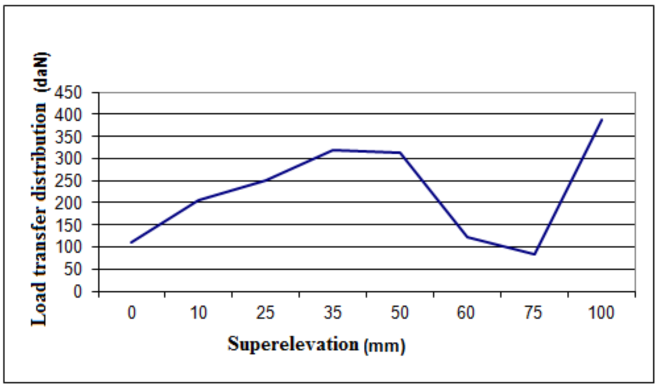

Fig.7: The load transfer distribution between wheels number 7 and number 8

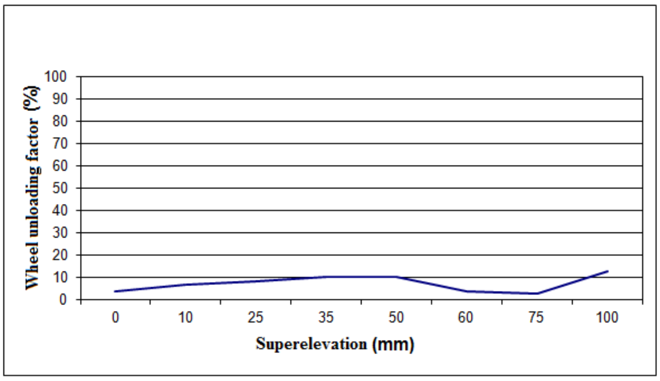

Fig.8: The wheel unloading factor for the wheel number 7

As it can be seen in the graphs corresponding to unloading of the rear bogie wheels, it appears that for both pairs of wheels, a normal unloading of the wheels on the left side of the vehicle occurs, simultaneous with the loading of the wheels on the right side vehicle up to a rail super-elevation of 50 mm. An abnormal behavior of the wheels load distribution appears for rail super-elevation between 50 mm and 75 mm, namely the loading of the odd wheels of the vehicle, simultaneous with unloading of the even wheels of the vehicle. This fact is due to a certain moment of inertia that opposes to the torsion of the vehicle bogie. Above 75 mm, when the simulated rail super-elevation increases, the loading of the rear bogie even wheels and the unloading of its odd wheels appear.

4 Conclusion

To summarize, the determination of the safety against overturning of the railway machines with dual road-rail system, is a method of overall verification of safety running of road-rail machines. Moreover, extending the applicability of the concerned method to all railway motor or trailer vehicles, a better assessment of the aspects of the safety running of the rolling stock that runs on Romanian railways can be obtained.

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