# EXPERIMENTAL VERIFICATION OF THE LATERAL ACCELERATIONS MEASURING METHOD FOR GUIDING FORCES SUM DETERMINATION OF A TWO AXLE WAGON

Daniel-Marius MIHAI[1],  Nicolae ENESCU[2]

*Lucrarea constă în analiza rezultatelor obținute pentru suma forțelor de ghidare prin două metode: măsurare directă și determinare prin calcul aplicată valorilor măsurate ale accelerațiilor orizontal-transversale de la nivelul cutiilor de osie. Comparația celor două metode evidențiază faptul că, după aplicarea unei corecții masice introduse datorită efectelor dinamice ale întregului ansamblu, metodele sunt echivalente. Această comparație este un element de originalitate în domeniul feroviar în România. Măsurătorile au fost efectuate pe un vagon de marfă pe două osii, în condiții de testare, la rularea acestuia în curbe cu raze mici.*

*This work consists in analyzing the results obtained for the guiding forces sum by two methods: direct measurement and indirect determination by applying the theoretical calculation to the lateral accelerations measured in the axle boxes. A comparison is made between the results obtained by the two methods, highlighting that, after applying a mass correction introduced due to the dynamic behavior of the system, the methods are equivalent. This comparison, made for the first time in Romania, represents an original procedure. The measurements were performed on a two axle freight wagon, in test conditions, on a track with small radii curves.*

**Keywords:** forces, accelerations, experimental determinations, railway freight vehicle, equivalent methods verification

## 1. Introduction

The sum of the guiding forces represents one of the most important criteria of determining the safety of the railway vehicle. Besides, the guiding forces sum is a measure of the track shifting and of the vehicle stability. An increased value of the guiding forces sum may lead to a dangerous track shifting, producing a serious deterioration of the track geometry.

The measuring process of the guiding forces, as a preliminary step of the guiding forces sum determination, becomes in this context an important procedure, not only for homologation process of new vehicles. There are two types of measuring methods applied for the guiding forces measuring: onboard

---

[1] Eng., EXCO TRANSPORT SYSTEMS, Bucharest, Romania , e-mail  daniel.mihai@excosys.ro
[2] Prof., Mechanics Department, University POLITEHNICA of Bucharest, Romania

measuring of forces in the wheel and offboard measuring of the forces determined in the rail while the train passes by.

The onboard measuring implies a measuring run along the whole route where the measuring sectors are located. Both methods use strain gauges applied either on the webs of both wheels of a special designed wheelset or on rails, along the measuring sector. The strain of the wheel web or of the rail is collected by the strain gauges and converted into signal by suitable measuring equipment. The results are interpreted in values of force via a previously made calibration. These methods represent the classic manner and are accepted, on a large scale, as the most appropriate ones.

Regarding the measuring wheelset, either a specially designed wheelset (with web or with spokes) can be used or the strain gauges mounting can be made on the spot, on web of the existing wheel of the vehicle. In this last case, the advantage is that the wheelset must not be dismounted and it doesn't need to have a large number of wheelsets to cover the whole range of wheel diameters used for the railway vehicles. The main disadvantage is that, for each vehicle, a separate mounting has to be done, a new calibration has to be performed and the strain gauges become disposable.

A Japanese study proposes an alternative method: instead of using a measuring wheelset equipped with strain gauges, destined to measure the web deformation, non-contact gap sensors are used to measure the relative wheel displacement in relation with the bogie. The gap sensors are placed on the non-rolling parts of the bogie, behind the wheel hub and on the inner part of the wheel rim. The non-contact gap sensors are supposed to simultaneously measure the web distortion in the hub area and in the wheel rim area. Based on a previously performed calibration the wheel displacement is interpreted as force variation. The method is innovative but still not very convenient, because compensation is needed due to lack of high accuracy. Another inconvenient is the installation of delicate equipment in an extremely exposed area, at the clearance limit of the vehicle. [1]

Nowadays, due to high costs of equipment and measuring runs, determination of the guiding forces and, as a consequence, of the guiding forces sum by simulation is widely spread. For instance, a study performed for the track shift and bogie stability of the Swedish train Regina 250, for straight track and large radius curves run at high speeds but also for running in tight curves, has approached the guiding forces determination by simulation, using a specially designed program. [2]

Other determinations by simulation of the guiding forces were applied for stability prediction of bogies, in engineering stage of railway vehicles. [3]

Several simulation programs have been developed, generally focused on the whole dynamic behavior of the vehicles, but with direct application for

determination of the guiding forces and, therefore, of the guiding forces sum. One of these programs is the German program pack SIMPACK, a program that showed an impressive accuracy which made it suitable for efficient acceptance analyses. [4]

The values of the guidance forces obtained by simulation were also used in studies regarding the interaction between trains and turnouts (crossings and switches). [5]

Although the simulation method has the advantage of reduced costs and fast evaluation, it leaves place for an unexpected behavior of such complex systems like the railway vehicles. An experimental determination is, therefore, absolutely necessary, especially in the prototype phase, when hidden designing faults may occur.

A simplified method is the measuring of the lateral accelerations, on both ends of the wheelset and the determination by calculation of the guiding forces sum value, according to the theory. This method avoids the difficulties implied by the guiding forces sum measurement by the mean of a specially designed wheelset or by strain gauges applied on the rail in special measuring sectors. Although the testing regulations mention this simplified method, a practical direct comparison was never performed. This comparison represents an original procedure in the railway domain in Romania.

The measurements were performed on a two axle freight wagon rolling on track sectors located in curves with constant radii and superelevation, with a constant speed corresponding to a certain range of cant deficiencies. The recorded signals for guiding forces and accelerations were processed by computer with dedicated software.

In regard with the theoretical considerations in this domain it was investigated if the values of the guiding forces sum obtained by direct measurement with the dedicated wheelset are similar to the values obtained by calculations based on the simultaneously measured axle box lateral accelerations. The accelerations were measured by the mean of accelerometers mounted on both axleboxes of the measuring wheelset.

## 2. Theoretical considerations

### 2.1. Wheelset forces

A dynamic balance can be established between the wheel-rail contact forces and the external forces that act upon the wheelset. On each of the wheels acts a nominal load $Q_o$ which represents half of the total axle load on a plane horizontal surface. During the run, the wheel load is being modified due to quasistatic forces which introduce transfers of load on the wheels of the same

wheelset or between the wagons' wheels. This happens also due to dynamic forces resulting mainly from the track irregularities. The transfer of load on a single wheelset is half the difference between the loads on the wheels of the same wheelset:

$$\Delta Q = \frac{Q_1 - Q_2}{2}. \tag{1}$$

Thus, on the wheels of the same wheelset we can find the momentary loads $Q_o + \Delta Q$ and $Q_o - \Delta Q$ respectively, while the wheelset load $2Q_o$ remains constant. By convention, it is considered that the transfer of load $\Delta Q$ has a positive sign if it loads the outer wheel. The ratio $\frac{\Delta Q}{Q_0}$ is called *transfer coefficient*. [6]

These transfers of load can be determined mainly by the external forces that act against the vehicle and are transmitted through the suspension to the wheels (such as the inertial forces caused by vehicle's change of direction and by twisted track). [7] Simultaneously, along each axle, a lateral force $H$ is present and it is transmitted through the axleboxes to the frame of the vehicle. This force combines all the rotational resistances, including the inertial forces caused by vehicle's change of direction. The $H$ force, acting above the wheel-rail contact area, creates a tilting torque of the axle. This fact determines a supplementary load and a resulting supplementary transfer of load depending on the wheel radius $r$:

$$\Delta Q_H = \frac{Hr}{2e}. \tag{2}$$

The effect of this load transfer is diminished by the elastic behavior of the rail and of the suspension. As consequence, the real transfer of load $\Delta Q_H$ is

$$\Delta Q_H = \frac{H \cdot \lambda \cdot r}{2e}, \tag{3}$$

where the diminishing coefficient $\lambda$ is considered to be 0.85 for regular wheels, with outer shafts. $2e$ represents the gauge (distance between the two rails).

As a consequence, taking into account the transfers of load $\Delta Q_o$ and $\Delta Q_H$, the wheel load becomes:

$$Q_1 = Q_o + \Delta Q_o + \Delta Q_H; \tag{4}$$

$$Q_2 = Q_o - \Delta Q_o - \Delta Q_H. \tag{5}$$

Satisfying the equilibrium condition we obtain:

$$Q_1 + Q_2 = 2Q_o. \tag{6}$$

The above expressions are valid for the free wheelset. The transfers of load between the axles can appear also due to vehicle's stiffness. [6]

### 2.2. The adopted model

We consider that the following non-inertial reference system is attached to the vehicle's wheelset:
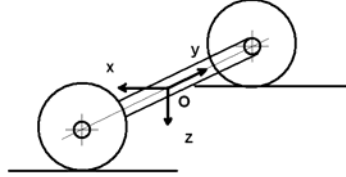


Fig.1. The non-inertial reference system chosen for the wheelset

In order to determine the dynamic force that appears between wheel and rail on lateral direction during the vehicle's run over a railway track with geometry defects we consider:

- $m_c$ = mass of the vehicle's chassis (sprung mass);
- $m_w$ = mass of the wheelset (unsprung mass);

The model's relative displacement is

$$y(t) = y_w(t) + y_c(t),$$ (7)

where $y_w(t)$ is the relative displacement of the wheelset and $y_c(t)$ is the relative displacement of the vehicle's chassis.

In these conditions, it might be presumed that the dynamic force on lateral direction that is acting on the wheelset is

$$Fd(t) = m_c \ddot{y}_c(t) + m_w \ddot{y}_w(t).$$ (8)

The dynamic force that appears in the wheelset while the vehicle is rolling in the above mentioned conditions is

$$Fd = m_c \ddot{y}_c + m_w \ddot{y}_w.$$ (9)

The equation that characterizes the wheelset's movement is:

$$\ddot{y}_w(t) = Y_w \sin 2\pi f t,$$ (10)

where $f$ is the frequency of the system's oscillation, comprised in the $[f_{min}, f_{max}]$ frequency band and $Y_w$ is the amplitude of the acceleration, ($Y_w$ = ct.).

As long as the system is a linear one, the dynamic force will follow the equation:

$$F_{Dw}(t) = Y_{Dw}(f) \sin[2\pi f t + \varphi(f)],$$ (11)

where $Y_{Dw}$ is the amplitude ($Y_{Dw}$ = ct.) and $\varphi(f)$ is the phase which does not affect the value of the force, introducing only a certain phase difference.

## 2.3. Vehicle's guiding forces in circular curves

Starting from the equilibrium conditions on lateral direction it comes out that the *H* force, named also axle lateral force or conducting force, in quasi-static conditions equals to the vector sum of the guiding forces $Y_1$ and $Y_2$ corresponding to both wheels of the same wheelset. In dynamic conditions, the *H* force value has to be added to the inertial lateral resistance of the wheelset *$(-m_w\ddot{y})$*, where $m_w$ is the cumulated mass of the wheelset and of all the items strapped on the wheelset and *$\ddot{y}$* is the wheelset lateral acceleration.

The sum of the guiding forces $Y_1$ and $Y_2$ is equal and of opposite sign with the force acting on the lower surface of the sleepers (between the sleepers and the ballast). This force is of major importance regarding safety against track lateral shifting. The *H* force, as well as the sum of the guiding forces, has a significant effect in rolling parts stress.

The dependency between the *H* force and the sum of the guiding forces is

$$\sum Y_{\max(2m)} = \left(H + m_w\ddot{y}\right)_{\max(2m)} \text{ [kN]}. \tag{12}$$

The expression proves that the axle load increases track resistance against track shifting but also determines an increment of the track lateral stress.

A vehicle that runs in quasi-static regime, (characterized by constant velocity $v$ [m/s], constant track radius $R$ [m]) is subject to a centrifugal acceleration:

$$a = \frac{v}{R} = \frac{V^2}{3.6^2 R} \text{ [m/s}^2], \tag{13}$$

where $V$ is the velocity in km/h.

The vehicle's body suspended on springs leans out the curve with an angle related to the track's plane $\phi_c$ named *quasistatic rolling angle.* [6]

The resultant acceleration projected on a direction parallel with the vehicle's uncompensated residual lateral acceleration is

$$\gamma_T = a\cos\left(\phi_0 - \phi_c\right) - g\sin\left(\phi_0 - \phi_c\right), \tag{14}$$

where $\phi_0$ is the track superelevation angle.

Because the difference *$(\phi_0 - \phi_c)$* has a small value, it can be considered that

$$\sin\left(\phi_0 - \phi_c\right) = \sin\phi_0 - \sin\phi_c, \tag{15}$$

and

$$\cos\left(\phi_0 - \phi_c\right) = 1 . \tag{16}$$

Considering that $\sin\phi_0 = \dfrac{h}{2e}$, where $2e$ [mm] is the gauge we obtain:

$$\gamma_T = \frac{V^2}{3.6^2 R} - \frac{gh}{2e} + g\sin\varphi_c, \tag{17}$$

which acts in vehicle's body against the freight. The expression highlights both the acceleration compensation introduced by track superelevation $h$ in curve and the increment of the lateral acceleration determined by the vehicle chassis leaning out in curve.

The quasi-static chassis lean out angle $\phi_c$ is in close connection with the vehicle's suspension. From the equilibrium equation of the forces that are acting upon vehicle's body in the axis of rotation it is deducted that:

$$\sin\varphi_c = S\left(\frac{V^2}{3.6^2 gR} - \frac{h}{2e}\right),\tag{18}$$

where $S$ is the stiffness coefficient of the vehicle which depends exclusively on the elastic and mass characteristics, specific to each type of vehicle.

Therefore, the uncompensated lateral acceleration can be expressed as:

$$\gamma_T = \left(\frac{V^2}{3.6^2 R} - \frac{gh}{2e}\right)(1+S) \ [\text{m/s}^2].\tag{19}$$

For $\phi_c = 0$ and $S = 0$, the transversal acceleration is

$$\gamma_{T0} = \frac{V^2}{3.6^2 R} - \frac{gh}{2e}\tag{20}$$

and, therefore,

$$\gamma_T = \gamma_{T0}(1+S),\tag{21}$$

relation that proves that the vehicle feels a greater value for acceleration in the chassis comparing to the one existing in track. [6]

Examining the expression (19), it can be noticed that to a real superelevation $h$ corresponds a velocity $V_0$ called *equilibrium speed (nominal speed)*. While $\gamma_T = 0$ the value of $V_0$ becomes:

$$V_0 = 3.6\sqrt{\frac{g}{2e}}\sqrt{Rh}\tag{22}$$

For the nominal gauge $2e = 1500$ mm, $V_0$ is:

$$V_0 = 0.291\sqrt{Rh} \ [\text{km/h}].\tag{23}$$

Introducing in the relation the value of the nominal velocity $V_0$, another expression of the transversal acceleration is obtained:

$$\gamma_T = \frac{V^2 - V_0^2}{3.6^2 R}(1+S) \ [\text{m/s}^2].\tag{24}$$

The theoretical superelevation $h_t$, corresponding to the zero value of the uncompensated transversal acceleration, at a velocity value $V$ different from $V_0$, is

$$h_t = \frac{2e}{3.6^2 g}\frac{V^2}{R},\tag{25}$$

which, for $2e = 1500$ mm, becomes:

$$h_t = \frac{11.8V^2}{R} \quad \text{[mm]}. \tag{26}$$

The difference between the theoretical superelevation $h_t$ and the real superelevation $h$ is called cant deficiency and is $I = h_t - h$. For a normal gauge it is

$$I = 11.8 \frac{V^2 - V_0^2}{R} \quad \text{[mm]}. \tag{27}$$

The cant deficiency considered for the general case is $I = 11.8 \frac{V^2}{R} - h$. The uncompensated centrifugal acceleration is $\gamma_{T0} = \frac{I}{153}$ and it is directed towards the outer side of the curve. [6]

Theoretically it is difficult to establish a general mathematical model for the calculation of the $H$ force, taking into consideration the multiple factors that occur, both from vehicle side and track side. According to Prud'homme, the dynamic force acting against the track is:

$$H_{qst} = \frac{2Q_o}{1500} I = \frac{2Q_o}{1500}(11,8 \frac{V^2}{R} - h) \quad \text{[kN]}, \tag{28}$$

where $H_{qst}$ is the quasi-static component which is generated by the uncompensated centrifugal acceleration $\gamma_{TO}$, proportional with the cant deficiency $I$ [mm] and with the axle load $2Q_o$ [kN].

The random component $H_a$ is in interdependence with the vehicle characteristics, the track geometry quality and its mechanical characteristics having the value

$$H_a = \frac{2Q_o V}{1000} \quad \text{[kN]}. \tag{29}$$

In these circumstances, the $H$ force is

$$H = H_{qst} + H_a = \frac{2Q_o}{1500} I + \frac{2Q_o V}{1000} = \frac{2Q_o}{1500}(11.8 \frac{V^2}{R} - h) + \frac{2Q_o V}{1000} \quad \text{[kN]}. \tag{30}$$

While the lateral acceleration in the axlebox is experimentally determined, the guiding forces sum can be determined based on the relation:

$$\sum Y_{\max(2m)} = (H + m_o \ddot{y})_{\max(2m)} = \left[ \frac{2Q_o}{1500}(11.8 \frac{V^2}{R} - h) + \frac{2Q_o V}{1000} + m_o \ddot{y} \right]_{\max(2m)} \quad \text{[kN]}$$

$$. \tag{31}$$

## 3. Experimental measurements

### 3.1. Measuring procedure and testing conditions

In order to experimentally determine the dynamic force between wheel and rail, the measurements are being performed on a two axles freight wagon rolling with constant speed on track sectors located in circular curves, with constant radii and superelevation. The run corresponds to certain cant deficiencies determined by the rolling speed and by the track radius. For simultaneous determination of the guiding forces and the axlebox lateral accelerations, two separate measuring chains were used. A proper calibration was previously performed. The lateral forces were measured by the means of a specially constructed measuring axle, with strain gauges applied on the web of the wheel. The accelerations on lateral direction were measured with accelerometers mounted on the axleboxes.  The signals corresponding to forces, accelerations and speed were digitally converted, recorded and processed by dedicated software for parameters determination.

The tests consisted in rolling at speeds up to 60 km/h on track sectors located in full curves with radii of 180 m, 250 m, 400 m and 800 m, with a nominal  superelevation of 70 mm, so that the cant deficiencies are comprised between 91 mm and 150 mm. The track sectors located in full curve (excluding spirals) were marked in the recorded signals, eliminating the file selection errors.

### 3.2. Processing of the acquired signals

In order to eliminate the external influences and the noise within the recorded signals and to keep only the frequencies corresponding to the parameters to be analyzed it was necessary to filter the signal. [8] To choose the right filter, a spectral analysis of each recorded signal was performed, highlighting the frequencies of the processed signals. [9]

The dominant frequencies for the acquired signals were:

*Table 1*

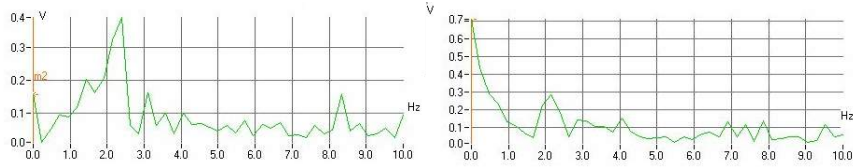| Loading state | $f_{(\Sigma Y)axle\ 1}$ [Hz] | $f_{(\Sigma Y)axle\ 2}$ [Hz] | $f\ \ddot{y}_{axle\ 1}$ [Hz] | $f\ \ddot{y}_{axle\ 2}$ [Hz] |
|---|---|---|---|---|
| Unloaded state | 2.2 | 2.3 | 2.2 | 2.3 |
| 22.5 t/axle loaded state | 1.9 | 1.8 | 1.7 | 2.1 |

Fig.2. Spectral analysis - acceleration signal - axle 1 and 2, unloaded state



Fig.3. Spectral analysis - force signal - axle 1 (outer and inner wheel), unloaded state



Fig.4. Spectral analysis - force signal - axle 2 (outer and inner wheel), unloaded state



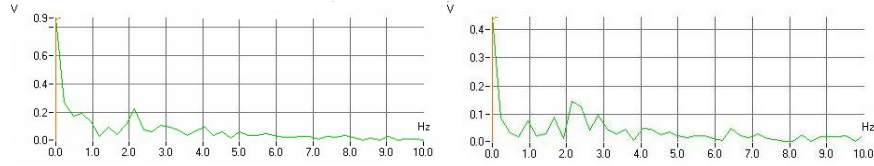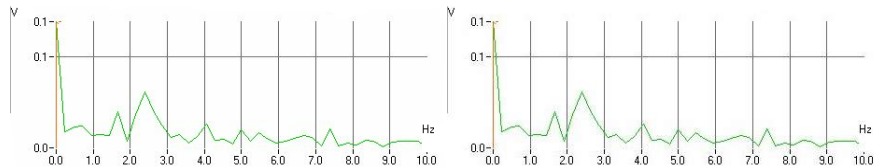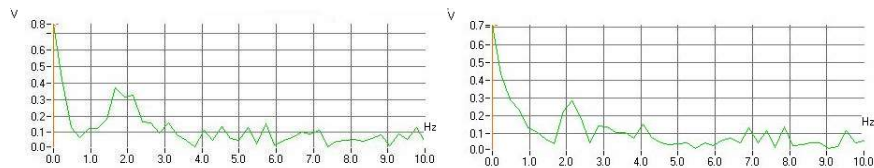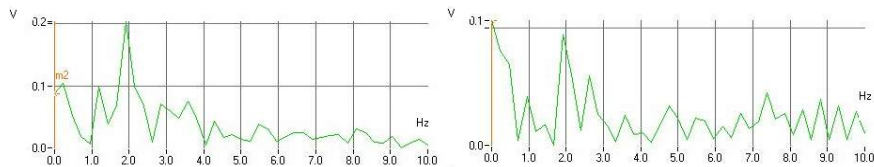Fig.5. Spectral analysis - acceleration signal - axle 1 and 2, loaded state



Fig.6. Spectral analysis - force signal - axle 1 (outer and inner wheel), loaded state
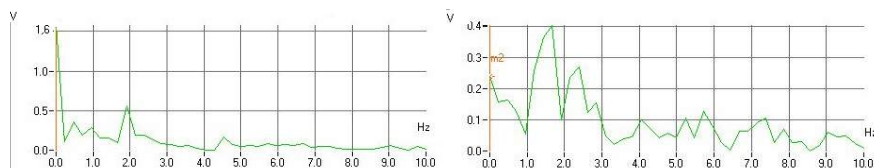


Fig.7 Spectral analysis - force signal - axle 2 (outer and inner wheel), loaded state

All recorded signals for the measured parameters were filtered with a lowpass filter with the cut-off frequency value at 10 Hz. [10] After signal filtering, the sectors on which the recordings were performed were selected. The small radii curves were split in groups as follows: the 1st group of curves with 180 m radius, the 2nd group of curves with 250 m, the 3rd group of curves with 400 m radius and the 4th group of curves with 800 m radius. For the curves with radii of 180 m and 250 m the length of the sectors was of 70 m and for curves with radii of 400 m and 800 m the length of the sectors was of 100 m.

After applying the filters, the force signals corresponding to the wheels of the same wheelset were algebraically summed. Signal processing was performed using the one-dimensional statistic processing method described by the international regulation in force referring to railway vehicles homologation from the safety point of view, Code UIC 518. 4th edition, 2009. [11]

The maximum estimated values, both for forces and accelerations, were determined at the statistic distribution of 0.15 % and 99.85 % frequencies, using the expression:

$$x_{\max} = \bar{x} + ks \tag{32}$$

where $\bar{x}$ is the mean value, $s$ the standard deviation and $k$ is a factor depending on the chosen level of trust. For safety related parameters $k = 3$. [11]

Taking into consideration that the mass of the wagon was 17.200 kg in the unloaded state and 45.000 kg in the loaded state corresponding to 22.5 t axle load, the maximum allowed limit for the sum of the guiding forces $(\Sigma Y)_{2m}$ is :

$$\left(\sum Y\right)_{2m} = \alpha\left(10 + P_0 / 3\right) \tag{33}$$

where $P_0$ is the axle load expressed in kN and $\alpha$ is the coefficient that takes into account the minimum characteristics imposed for the track stability under lateral forces generated by the vehicle. For freight wagons $\alpha = 0.85$.[11]

Therefore, the calculated maximum allowed limit for the sum of the guiding forces $(\Sigma Y)_{2m}$ is 32,4 kN for the wagon in the unloaded state and 71,04 kN for the wagon in the loaded state corresponding to 22.5 t axle load. The calculated average dynamical mass of a wheelset is 1853 kg for the unloaded wagon and 2426.5 kg for the wagon in loaded state.

### 4. Results

The measured values of the lateral acceleration in the axlebox $\ddot{y}$, the maximum values of the guiding forces sum $(\Sigma Y)_{2m}$ determined by calculation starting from the measured acceleration values and the measured values of the guiding forces sum $(\Sigma Y)_{2m}$ obtained after recorded signals processing are presented below. The maximum allowed limit for $(\Sigma Y)_{2m}$ is also presented. The

results are separated for each axle and are corresponding to each of the two loading states of the wagon (empty and loaded to 22.5 t axle load), as follows:

*Table 2*

**Wagon in unloaded state, axle 1**

| Curves | $\ddot{y}$ [m/s²] | Calc. $(\Sigma Y)_{2m}$,[kN] | Meas. $(\Sigma Y)_{2m}$, [kN] | Lim. $(\Sigma Y)_{2m}$,[kN] |
|--------|-------------------|------------------------------|-------------------------------|------------------------------|
| group 1 | 8.55 | **30.24** | **27.5** | |
| group 2 | 3.68 | **17.50** | **16.91** | 32.4 |
| group 3 | 2.61 | **11.93** | **12.28** | |
| group 4 | 1.46 | **6.82** | **6.77** | |

*Table 3*

**Wagon in unloaded state, axle 2**

| Curves | $\ddot{y}$ [m/s²] | Calc. $(\Sigma Y)_{2m}$,[kN] | Meas. $(\Sigma Y)_{2m}$, [kN] | Lim. $(\Sigma Y)_{2m}$, [kN] |
|--------|-------------------|------------------------------|-------------------------------|------------------------------|
| group 1 | 6.97 | **27.31** | **27.53** | |
| group 2 | 4.30 | **18.65** | **18.46** | 32.4 |
| group 3 | 2.12 | **11.03** | **11.49** | |
| group 4 | 1.43 | **6.76** | **7.01** | |

*Table 4*

**Wagon in loaded state, corresponding to a 22,5t axle load, axle 1**

| Curves | $\ddot{y}$, [m/s²] | Calc. $(\Sigma Y)_{2m}$, [kN] | Meas. $(\Sigma Y)_{2m}$, [kN] | Lim. $(\Sigma Y)_{2m}$, [kN] |
|--------|--------------------|-------------------------------|-------------------------------|------------------------------|
| group 1 | 7.09 | **54.87** | **53.81** | |
| group 2 | 3.25 | **35.83** | **36.94** | 71.04 |
| group 3 | 2.60 | **24.88** | **24.60** | |
| group 4 | 1.62 | **14.69** | **14.71** | |

*Table 5*

**Wagon in loaded state, corresponding to a 22,5t axle load, axle 2**

| Curves | $\ddot{y}$, [m/s²] | Calc. $(\Sigma Y)_{2m}$, [kN] | Meas. $(\Sigma Y)_{2m}$, [kN] | Lim. $(\Sigma Y)_{2m}$, [kN] |
|--------|--------------------|-------------------------------|-------------------------------|------------------------------|
| group 1 | 5.15 | **50.17** | **51.18** | |
| group 2 | 3.61 | **36.71** | **34.29** | |
| group 3 | 2.20 | **23.91** | **24.36** | 71.04 |
| group 4 | 0.89 | **12.92** | **13.07** | |

## 5. Conclusions

Taking into consideration that the guiding forces and the lateral accelerations were simultaneously measured, on identical track sectors, having identical recording settings and that the analysis procedure was identical as well, we have the conditions to establish a correspondence between the two methods applied for the determination of the guidance forces sum.

In order to determine if the two methods are equivalent we have to compare the values obtained and see if significant differences exist. This can be

done by calculating the relative average deviations between the values obtained by calculation from the measured values of the accelerations and the direct measured values for $(\Sigma Y)_{2m}$.

*Table 6*

**Relative average deviations between the guiding forces values determined by the two different methods**

| Curves | Relative average deviation between $(\Sigma Y)_{2m\ calc.}$ and $(\Sigma Y)_{2m\ meas}$ [%] | | | |
|---|---|---|---|---|
| | *Unloaded state* | | *Loaded corr. to 22,5t axleload* | |
| | Axle1 | Axle 2 | Axle 1 | Axle 2 |
| R = 180m | 4.75 | 0.40 | 0.97 | 0.99 |
| R = 250m | 1.72 | 0.51 | 1.52 | 3.41 |
| R = 400m | 1.45 | 2.04 | 0.57 | 0.93 |
| R = 800m | 0.37 | 1.82 | 0.07 | 0.58 |
| Average deviation between $(\Sigma Y)_{2m\ calc.}$ and $(\Sigma Y)_{2m\ meas}$ | | | | |
| 1.63 % | | | 1.13 % | |
| 1.38 % | | | | |

It is noticed that the obtained values are approximately equal. The values obtained by calculation based on accelerations determination have the tendency to be lower than the ones determined by experimental forces measuring. For each separate axle and corresponding to each loading state, the deviations between the values obtained by the two methods are calculated.

The eventual differences are caused mainly by the different evaluation of the conducting force *H*, particularly by its random component which could not take into consideration the real geometrical and mechanical characteristics of the railway track and their corresponding influences. The calculation of the force *H* could not include the real transfers of load which occur during the run. The unsprung mass introduces supplementary effects which could not be determined, as well.

The two methods lead to significantly close results and it can be considered that, while proper corrections are applied, both are relevant for the evaluation of the guiding forces sum $(\Sigma Y)_{2m.}$ The differences between the values obtained by calculation after acceleration determination and the experimentally determined values are indeed low.

The importance of applying this method of the guiding forces sum $(\Sigma Y)_{2m}$ determination is that it offers a low-cost and easy solution for evaluating the safety parameters, without the need of a specially constructed measuring wheelset or other expensive equipment. The inconvenience generated by the mandatory use of a device that has to transmit the measured signal from bodies in rotation to stationary bodies, at high rotational speeds, is also eliminated. This leads to the lowest possible errors or signal loss. The main advantage of the alternative

method is that it does not require the use of high tech standard equipment with an extremely high acquisition price and it does not presume long times for preparations of the specially designed wheelset, offering comparably reliable results.

R E F E R E N C E S

[1] *Matsumoto, Akira; Sato, Yasuhiro; Ohno, Hioroyuki; Tomeoka, Masao; Matsumoto, Kosuke; Kurihara, Jun; Ogino, Tomohisa; Tanimoto, Masuhisa; Kishimoto, Yasushi; Sato, Yoshi; Nakai, Takuji,* "A new measuring method of wheel-rail contact forces and related considerations", in Wear no.265 , Feb.2008, pp. 1518-1525.

[2] *Orvnäs, Anneli; Andersson, Evert; Persson, Rickard,* "Development of track-friendly bogies for high speed – a simulation study", KTH Engineering and Sciences, Royal Institute of Technology, Aeronautical and Vehicle Engineering, Rail Vehicles, Stockholm, May 2007, ISBN 978-91-7178-726-2.

[3] *Polach, Oldrich; Vetter, Adrian,* "Methods for running stability prediction and their sensitivity to wheel/rail contact geometry", 6th International Conference on Railway Bogies and Running Gears, Budapest, 13-16 Sept. 2004.

[4]***SIMPACK AG – SIMPACK Rail – web site of SIMPACK AG, www.simpack.com

[5] *Kassa, Elias; Nielsen, Jens C.O.,* "Dynamic train-turnout interaction in an extended frequency range using a detailed model of track dynamics", in Journal of Sound and Vibrations no.320 , Aug.2008, pp. 893-914.

[6] *I. Sebeşan* "Dinamica vehiculelor de cale ferată (Railway Vehicles Dynamics)"**,** Ed.Tehnică, Bucureşti, 1995. (in Romanian)

[7] ERRI B55 RP 8, "Prevention of derailment of goods wagons on distorted tracks", Utrecht, April 1983.

[8]***National Instruments LabVIEW 8.6 – Users Guide, 2008.

[9] *I. Magheti, Mariana Savu,* "Vibratii mecanice: teorie şi practica (Mechanical vibrations: theory and praxis)", Ed. BREN, Bucureşti, 2004. (in Romanian)

[10] *Al. Mihalcea,* "Prelucrarea optimă a semnalelor în sistemele informaţionale (Optimal processing of signals in informational systems)", Ed. Militară, Bucureşti, 1987.

[11] Code UIC 518, "Essais et homologation de vehicules ferroviaires du point de vue du comportement dynamique – Securite –Fatigue de la voie – Qualite de marche", 4th edition, 2009.