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INFLUENCE OF VARIOUS ANOMALIES IN THE VEHICLE AND TRACK PARAMETERS ON THE POSSIBILITY OF DERAILMENT

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Abstract

A detailed nonlinear model of the four-axle real vehicle, considering both vertical and lateral vibrations has been proposed. Inertial and elastic-viscous properties of the track have been taken into account. Comparison of the theoretical investigations and on-track tests has confirmed the rather good authenticity of the model. The calculations for anomalies either in track or in vehicle parameters as well as for their combinations have been performed.

1. Introduction

Despite safety measures, strictly observed on railways, the cases of derailment take place sometimes. These cases are becoming more frequent with increasing of operational speeds, axle loads, train length and mass. Thus, the necessity of detailed modelling of such accidents arises in order to determine the causes of them and, that is most important, to prevent the recurrence of such situations in future.

Such an analysis becomes possible on condition that the reliable information about the track and the vehicle is available and that the analytical model detailed sufficiently of the vehicle–track system, allowing to use this information, exists.

The analytical model of the four-axle freight wagon, the software package, designed on the base of this model and some results, obtained with their help, are described in this paper.

2. Freight Wagon Vertical and Lateral Dynamics Model

The four-axle open goods wagon was chosen by the authors as an object for modelling. Such a choice has made it possible to investigate a lot of various types of freight rolling stock, operated on the Ukrainian railways. The model of the wagon is shown in *Fig. 1*.

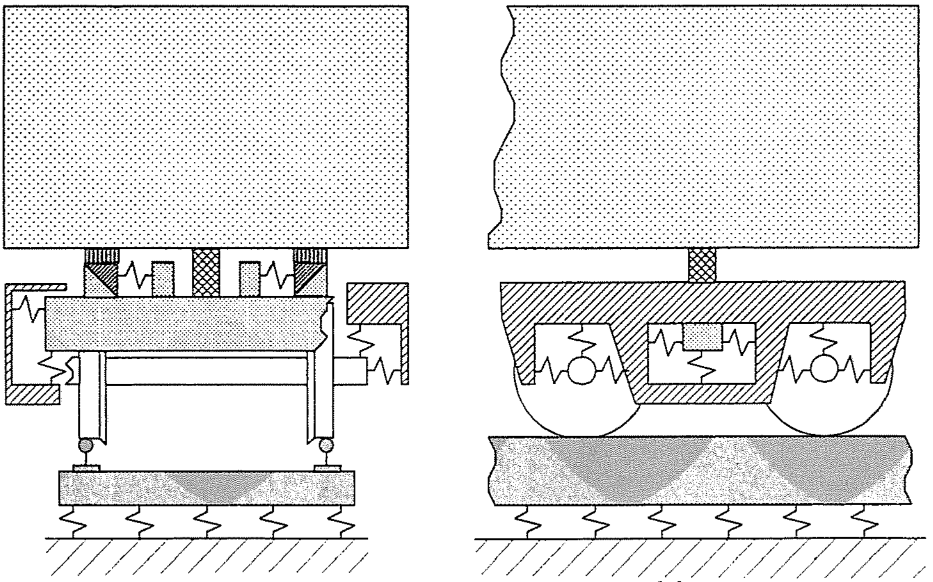

Fig. 1. Four-axle freight wagon model

A brief description of the model is given below (more details one can find in [1]).

The investigated system consists of eleven solid bodies (a car body, two bolsters, four side frames, four wheelsets). If necessary, it is possible to take into account car body flexibility. The following specific features of the vehicle have been reflected in the model as well.

Car body - bolster. One neglects with a clearance between the pivot and the centerplane, thus the mutual longitudinal displacements of these bodies are absent and their pitching is identical. As regards rolling and yawing of the bolster – they may occur independently from rolling and yawing of the car body. The action of the slippers of various types can be modelled.

Bolster - side frame. Vertical, lateral and longitudinal springs allow all possible mutual linear displacements and mutual yawing, In addition to springs, dry friction dampers with asymmetric characteristic are installed in this unit.

Side frame - wheelset. All possible mutual displacements between these bodies are allowable. Installation of the rubber spacers can be simulated by adding the Coulomb friction elements.

Thus, the following non-linearities are modelled: rolling and pitching of the car body on the pivot, kicks after taking up the clearances in the slippers, geometric and physical non-linearities in the wheel–rail contact points (the creepage forces are modelled by the CARTER's theory [2]).

The VLASOV's hypothesis [3] is accepted for the description of elastic-viscous and inertial properties of the track in vertical and lateral directions.

After taking into account all the above-mentioned features and restraints one obtains a 59-degree-of-freedom model. The corresponding ordinary differential equations system has been composed. The Runge–Kutta (for the first steps) and Adams–Bashforth methods are used for its solving.

The nominal values for some parameters of the standard freight open goods wagon without anomalies are listed in *Table 1.* They have been taken from [4] and [5].

3. A Brief Description of the Software Package

The described model has become the base for the program package for IBM PC computers with MS DOS. The computational part has been written in FORTRAN and the interface modules have been written in C language. The package provides a user-friendly interface, allowing to change the input parameters in easy way. The output of the program contains values for more than one hundred dynamic characteristics.

In addition some other, stand-alone programs have been created. They allow:

- to generate the file with the track irregularities, necessary for the main program in different ways:
- to draw plots of the computed displacements, accelerations and forces;
- to compute the power spectrum densities of the various processes;
- to determine the increment of the rolling radius depending on mutual lateral wheel–rail displacement for various real profiles of the wheel and rail.

The authors would like to thank Mr. R. Granovsky and Mr. A. Reidemeister for their help in creation of these programs.

4. Modelling of the Track Irregularities

Four types of the track irregularities are used as an input for the simulation program, described in Section 3. Those are:

- deviation of the left rail from a uniform profile in a vertical plane;
- deviation of the right rail from a uniform profile in a vertical plane;
- deviation of the left rail from a straight line in a horizontal plane;
- deviation of the right rail from a straight line in a horizontal plane;

For some reason such a set of irregularities was chosen as more convenient in comparison with a traditional vertical profile – cross level – alignment – gage set.

The file with necessary track irregularities may be created with the separate program in several different ways, namely:

Table 1.
Data for the standard wagon

Mass of the car body	76500 kg
Rolling moment of inertia for the car body	7.5E4 kg·m^2
Pitching moment of inertia for the car body	1.05E6 kg·m^2
Yawing moment of inertia for the car body	1.1E6 kg·m^2
Bogie centre distance	8.66 m
Mass of the bolster	450 kg
Rolling moment of inertia for the bolster	300 kg·m^2
Pitching moment of inertia for the bolster	50 kg·m^2
Yawing moment of inertia for the bolster	300 kg·m^2
Vertical stiffness of the secondary suspension springs	4000 kN/m
Shearing stiffness along the lateral axis between the bolster and the side frame	6000 kN/m
Torsional stiffness between the bolster and the side frame	200 kN m/rad
Clearance in the slippers	0.005 m
Mass of the side frame	680 kg
Pitching moment of inertia for the side frame	220 kg·m^2
Yawing moment of inertia for the side frame	220 kg·m^2
Bogie wheelbase	1.85 m
Mass of the wheelset	1370 kg
Rolling and yawing moment of inertia for the wheelset	1000 kg·m^2
Nominal wheel radius	0.45 m
Distance between planes of rolling radii	1.580 m
Flange clearance	0.007 m
Reduced vertical track stiffness	43830 kN/m
Reduced horizontal track stiffness	16570 kN/m
Reduced vertical track mass	659 kg
Reduced horizontal track mass	76 kg

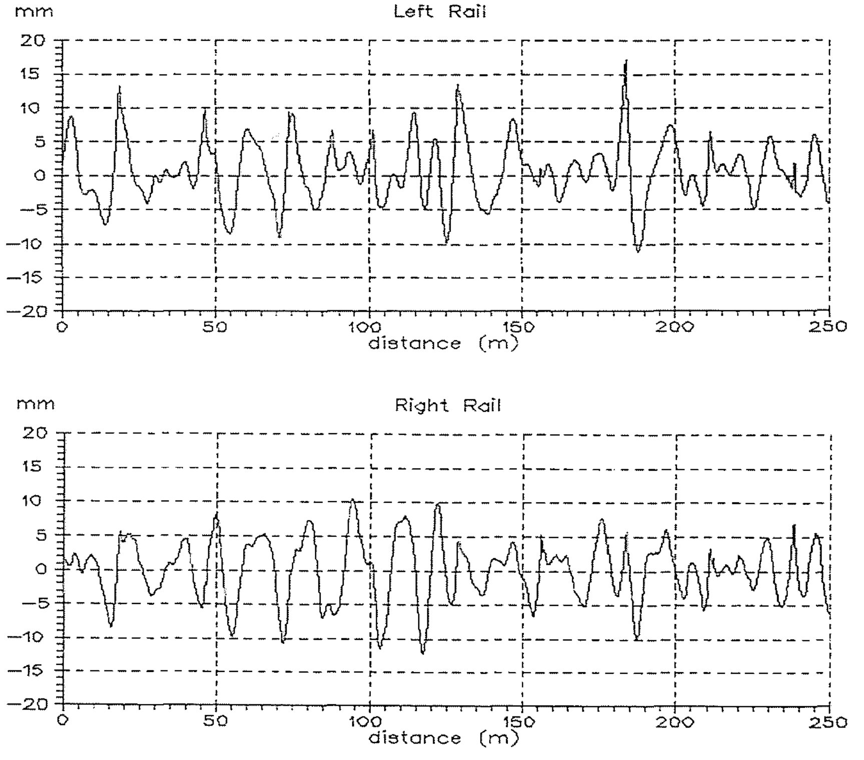

Fig. 2. Measured rail profiles

- after processing the data, recorded with a track-test car (an example of the vertical irregularities, obtained in such a way, is shown in *Fig. 2*);
- by inverse Fourier transform of the vertical profile, cross level, alignment and gage power spectral densities and subsequent change-over to the chosen type of irregularities (the authors use well-known expressions for the PSDs from [6]);
- by modelling the track with the periodically repeating joints;
- by modelling the isolated track geometry variations like bump, jog, plateau, etc.

5. Validation of the Model

The track irregularities shown in *Fig. 2* were recorded during the on-track tests simultaneously with some wagon's performance (vertical and lateral accelerations of the axle boxes and car body). The measurements were carried out with various speeds both on jointed and on continuously welded track. Thus, the authors got an opportunity to compare the results, obtained by computer simulation, with the corresponding experimental data. For the vertical dynamics performance good coincidence was received. As to horizontal accelerations of the car body, the situation proved to be not so optimistic. The amplitudes of the theoretical accelerations coincide closely with the test ones. As to power spectrum densities of both processes, the coincidence in the low frequencies interval (0.5–3 Hz) is good enough but for higher frequencies (3–10 Hz) the difference occurred. The authors prone to see the origins of this in rather rough modelling of a dry friction in the system.

6. Criterion of Safety and Estimation of the Dynamic Performance

The so-called *coefficient of stability of a wheelset during flanging* (for briefness we shall refer to it simply as to the 'coefficient of stability') is used as a criterion of safety on the railways of the former USSR according to [7]. This coefficient is determined by the following expression:

$$C_S = \frac{|P_1 C_1 - P_2 C_2 - C_3|}{|H_b|}, \tag{1}$$

where

C_s – coefficient of stability;
P_1 – left-axle-box-to-side-frame vertical force;
P_2 – right-axle-box-to-side-frame vertical force;
H_b – frame force (the sum of left and right axle-boxes-to-side-frames lateral forces).

The coefficients C_1, C_2, C_3 depend upon the mass and geometric characteristics of the vehicle [8]. The motion of the wheelset is assumed to be unstable when the coefficient of stability is less than 1.

The program determines the minimum values of the coefficient of stability for all four wheelsets together with distances where those values were registered. The total length (in meters and seconds) of the longest continuous series for $C_s < 1$ also is printed for each wheelset.

Besides the coefficients of stability the program analyses more than a hundred dynamic characteristics of the wagon. For each of them its minimum and maximum values, the distances, where these values were registered, arithmetic mean, dispersion, mean square and confidence limit are determined. The *Fig. 3* illustrates the above-mentioned statistics for some dynamic performance (frame forces (Hb) and coefficients of stability (CS) values for all four wheelsets as well as car body roll angle (THETA)).

Wagon: empty
Speed: 75.0 km/h
Track: Straight

	F_{min}	X_{min}	F_{max}	X_{max}	M	D	S	$M+2.5S$
Hb(1,1)	-17.06	80.29	13.22	86.71	-.3517E-01	10.22	3.196	8.026
Hb(1,2)	-14.93	80.21	10.74	86.61	-.4579E-01	11.90	3.450	8.671
Hb(2,1)	-7.963	68.45	7.769	62.76	.4284E-01	2.110	1.453	3.674
Hb(2,2)	-8.088	69.10	8.232	63.02	.4936E-01	3.044	1.745	4.411
				...				
THETA	-8.191	89.99	7.060	84.80	-.4488E-01	19.26	4.388	11.02
				...				

Coefficients of stability

	CS_{min}	X_{min}	Longest continuous series < 1
1,1	3.086	73.56	from .000 m to .000 m (.00000 sec)
1,2	3.254	71.35	from .000 m to .000 m (.00000 sec)
2,1	3.720	70.30	from .000 m to .000 m (.00000 sec)
2,2	4.770	71.41	from .000 m to .000 m (.00000 sec)

Fig. 3. Fragment of the program's output

7. Anomalies Modelling

The vast number of the model input parameters makes it possible to simulate various anomalies in the track and in the vehicle. Typical anomalies in the track could be modelled with the help of the program, described in section 4. Among the most important anomalies of the vehicle which are possible to imitate one should mention the following ones:

- nonstandard wheel–rail profiles, appeared because of wear or improper machining;
- wear of the different vehicle suspension elements;
- displacement of the car body centre of gravity both in longitudinal and in lateral directions due to inaccurate loading of the vehicle.

8. Some Results

The material of current section has been obtained as a result of analysis of the derailment accident which occurred in reality. The empty tank wagon derailed at the speed of 75 km/h. Careful analysis has revealed some severe anomalies both in the rolling stock (mainly in the wheel profiles that is shown in *Fig. 4* in comparison with a standard profile) and in the track (deflections in alignment and cross-level).

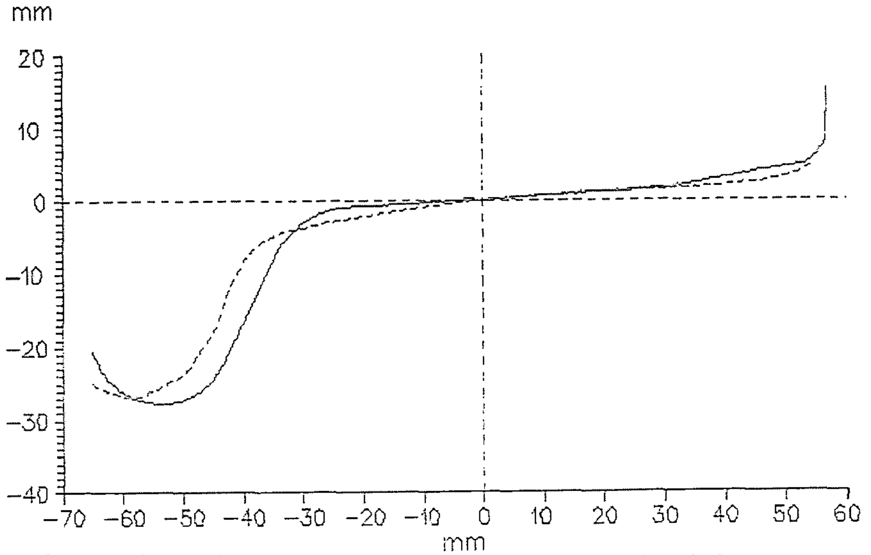

Fig. 4. Standard wheel profile (solid line) and the example of the worn profile (dashed line)

Then numerous calculations were carried out in order to clarify the circumstances of the accident. They were performed for four following combinations of anomalies:

- 'nominal case', i.e. the standard wheel profile and the 'good' track (the real track with average deflections was chosen as a 'good' one);
- the wheel profile from the derailed tank wagon and the 'good' track;
- the standard wheel profile and the track from the accident;
- combination of anomalies in the wheel profile and the track.

The influence of speed on the coefficient of stability for the first wheelset is shown in *Fig. 5.* The dependence of the maximum car body roll angle on the vehicle speed also was studied. The correspondent plots are shown in *Fig. 6.*

While analysing these plots one can draw a conclusion that the non-standard profile has decreased the dynamic performance of the wagon in comparison with the 'nominal case', and the qualitative behaviour of the 'parameter value vs speed' line has remained almost the same. The particular track geometry has caused the sufficient changes in this behaviour: the speed interval of 65 to 80 km/h has become a critical one, the most dangerous interval is 70 to 75 km/h. The combination of the anomalies has resulted in decrease of the stability coefficient much below the safe level.

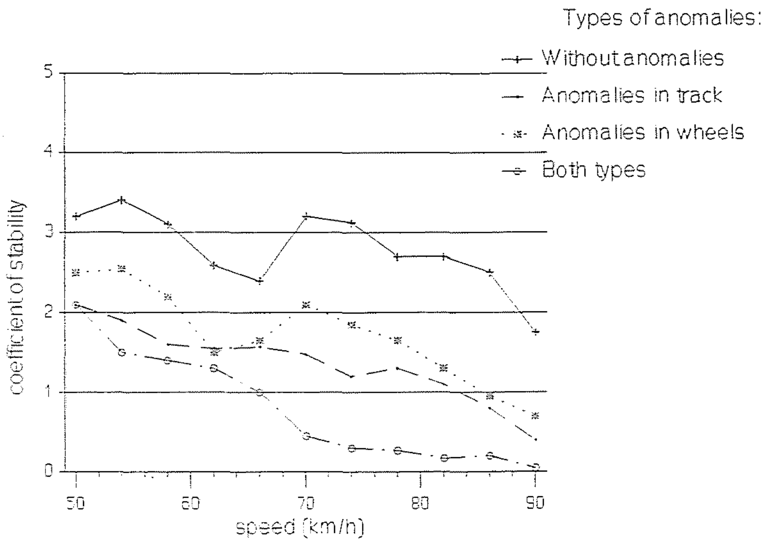

Fig. 5. Effect of speed on the coefficient of stability

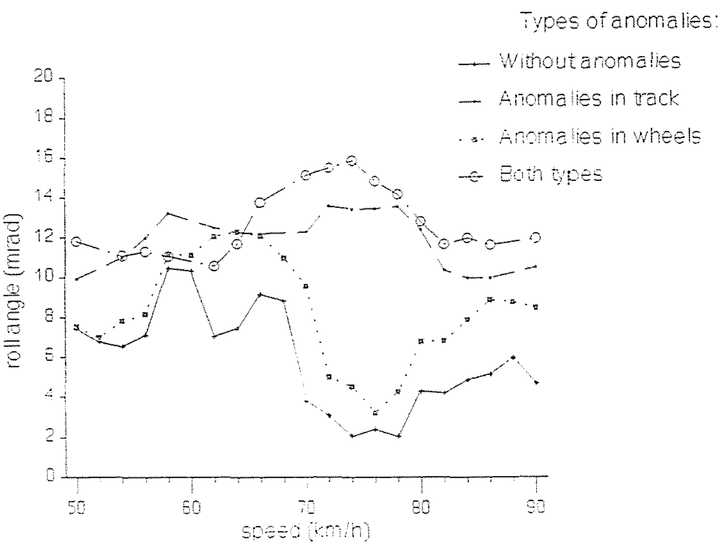

Fig. 6. Effect of speed on car body maximum roll angle

9. Conclusions

The created software package allows to simulate the behaviour of the four-axle freight wagon while travelling over the straight or curved track and to analyse therefore its dynamic performance. Existing of deflections from a nominal value in the parameters of the vehicle or track causes the deterioration of the performance as a rule. Thus, it becomes possible to determine the anomalies limits exceeding of which may cause a railway accident.

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