RIGA TECHNICAL UNIVERSITY

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INVESTIGATION OF WAGON DERAILMENT WHEN PASSING CURVILINEAR SPACE INTERVALS AND DETACHING AT GRAVITY HUMP YARDS

Summary of Doctoral Thesis

Riga 2010

RIGA TECHNICAL UNIVERSITY
Department of Transport and Machine Engineering
Institute of railway transport

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CONFIRMATION

It is hereby confirmed, that I completed the following thesis, presented in Riga Technical University for doctoral degree in engineering independently. The thesis is not presented in any other university for academic degree.

Pavel Gavrilov ……………… (autograph)

Date: 11th of June 2010

The thesis is written in Latvian, includes introduction, 4 chapters, conclusions, list of references, 7 annexes, 66 drawings, 5 tables, 121 pages in total. List of references consists of 139 items.

ANNOTATION

Thesis “Investigation of wagon derailment when passing curvilinear spaces and detaching at gravity hump yards” is completed by Pavel Gavrilov in order to receive doctoral degree in engineering. Academic Advisor, Dr.sc.ing, associated professor Diy Sergeyev.

The work contains a collection and analysis of freight wagon derailment statistics on Latvian Railway in the period of 1997 – 2009.

An experiment of defining loads, promoting derailment of freight wagons when passing curvilinear space intervals, was carried out.

Values of Coulomb friction parameters in supporting nodes were figured out experimentally, allowing the definition of bogie rotation antitorque moment towards wagon body for various type freight wagons, when passing curvilinear space intervals.

Cut dynamics in retarder position of gravity hump was investigated in order to find out the possible reasons of wagon derailment at gravity marshalling yards.

Recommended guidelines are offered to decrease bogie rotation antitorque moment towards wagon body and derailment at gravity marshalling yards.

Content

1. RELEVANCE OF WORK

Railway transport, a mean of transporting heaps of people and huge volumes of freights, including hazardous and highly dangerous goods, is classified as a branch of national economy with increased risk of emergency situations.

Emergency situations involving wagons with chemicals, oil products and poisons cause contamination of environment and require substantial expenses on rectification of ecologic consequences of troubles.

Long operational experience of railway worldwide shows, that in spite of efforts to provide safe transportation of freights, accidents such as wheel pair derailment happen regularly.

On Latvian railway, the percentage of derailment is as follows: 40% – derailment when detaching at gravity hump yards, 27% – derailment when passing curvilinear space intervals. Wagon derailments on railway yards when detaching wagons lead to delays in freight train forming and consignment, as well as cause expenses dealing with freight damage, railway and wagon repairs. Derailment on curvilinear space intervals also cause delays in train schedules. Derailments are sometimes related to environment contamination and expenses on rectification of ecologic consequences of troubles.

Financial losses from decline of amount of similar sort of failures are obvious. This is the reason why term “derailment” in the context of the discussed problem should be understood as “derailment of wagon wheel pair caused by wheel pair flange rolling over rail head”.

Investigation of wagon derailment when passing curvilinear spaces and detaching at gravity hump yards appears a relevant problem for Latvian railway.

2. WORK OBJECTIVE AND TASKS

Work objective. Investigation of wagon derailment case reduction when passing curvilinear space intervals and detaching at gravity hump yards, recommendations for cutting the likehood of such effect.

Tasks:

- ❑ collection and analysis of freight wagon derailment statistics on Latvian Railway in the period of the last ten years and systematization in order to figure out the extent of problem aggravation;
- ❑ investigation of bogie supporting node state of freight wagons sent for roundhouse servicing and overall repair;
- ❑ definition of parameters for friction between bolster and wagon frame in supporting nodes;
- ❑ definition of freight wagon bogie rotation antitorque moment, depending on conditions of wagon body supporting on bogie and the presence of lubricant;
- ❑ investigation of cut dynamics in retarder position of gravity;
- ❑ work out recommendations for:

  - reduction of bogie rotation antitorque moment towards wagon body, allowing derailment case reduction when passing curvilinear space intervals;
  - choosing regime of breaking when detaching cuts at gravity hump yards, in order to reduce derailment cases.

3. RESEARCH METHOD AND METHODICS

Research includes experimental method of freight wagon bogie rotation antitorque moment definition towards wagon body, depending on conditions of wagon body supporting on bogie and the presence of lubricant. Required information was received basing on multivariate numerical calculation of differential equations, describing wagon dynamics, using standard features of MATH CAD package. Depending on its mass centre position, on the total length of cut, i.e. on the whole delayed action in front and behind, on the place of breaking application on the cut.

4. SCIENTIFIC NOVELTY OF WORK AND MAIN RESULTS

1. An assembly is designed, providing an opportunity of modelling work environment of supporting nodes of freight wagon bolsters when passing curvilinear space intervals.
2. Experimentally proved friction parameters were defined in freight bogie bearing node, depending on conditions of wagon body supporting on bogie and the presence of lubricant.
3. Bogie rotation antitorque moments towards wagon bodies of various types were defined.
4. It is stated, that that the absence of lubricant in supporting nodes substantially (approximately 1,7...2,4 times) increases bogie rotation antitorque moment towards wagon body.
5. Experimental researches showed that the installation of expansion rollers on bearers substantially decreases bogie rotation antitorque moment towards wagon body by factor of 4...6.
6. Wagon oscillation mathematical model for examination of cut target breaking at detaching at gravity hump yards is obtained.
7. A condition to define a state of increased risk of cut derailment when detaching at gravity hump yards is conceived.

5. PRACTICAL VALUE OF WORK

Basing on completed research, recommended guidelines were worked out in order to decrease bogie rotation antitorque moment towards wagon body, eliminating the probability of freight wagon derailment when passing curvilinear space intervals. Also, guidance is given to choose cut braking mode when detaching at gravity hump yards in order to minimize probability of derailment, providing safe formation of car sets.

6. WORK APPROVAL

On the results of the work has been reported and are discussed:

Latvia:

1. RTU 47. starptautiskajā zinātniskajā konferencē, Rīga, RTU, 2006;

2. 5-ая международная конференция MET-2007**,** Металлы, сварка и порошковая металлургия", Юрмала, Латвия, 2007;
3. RTU 48. starptautiskajā zinātniskajā konference, Rīga; RTU, 2007;
4. RTU 49. starptautiskajā zinātniskajā konference, Rīga; RTU, 2008.

Abroad:

1. Конференция молодых ученых Литвы, Наука – будущее Литвы, Вильнюс, ВГТУ, 2007;
2. IV международная научно-практическая конференция, Белоруссия, г. Гомель (БелГУТ), 2007;
3. V Konferencija Naukowo Techiczna, Poland, Szczyrk, 2008;
4. V Международная научно-практическая конференция, TRANS-MECH-ART, МИИТ, Россия, Москва, 2008;
5. VIII Scientific Conference TLTS'08, Poland, Katowice-Cieszyn, 2008;
6. II международная научно-практическая конференция, «Проблемы и перспективы развития транспортных систем и строительного комплекса», Белоруссия, г. Гомель (БелГУТ), 2008;
7. 13TH International conference on computer systems aided science, industry and transport, konference **„**TRANSCOMP 2009**"**, Zakopane, Poland, 2009.

Publications

The main theses, conclusion and recommendations are reflected in the following scientific publications:

1. Гаврилов П. „Вкатывание гребня колеса колесной пары на головку наружной рельсовой нити", Конференция молодых ученых Литвы, Наука – будущее Литвы, Вильнюс, ВГТУ, 2007, Сборник статей конференции: 299 – 304 стр.;
2. Гаврилов П. „Восстановление подпятника тележки вагона наплавкой", 5-ая международная конференция MET-2007**,** Металлы, сварка и порошковая металлургия", Юрмала, 2007, Сборник статей конференции: 123 – 128 стр.;
3. Гаврилов П., Сергеев Д. „Анализ сходов вагонов на Латвийской железной дороге" IV международная научно-практической конференция. Белоруссия. г.Гомель (БелГУТ), 2007, Сборник статей конференции: 7 – 8 стр.;
4. D. Sergeyev, P. Gavrilov "Analysis of derailment of carriages on the Latvian railway", RTU zinātniskie raksti. 6. sēr., Mašīnzinātne un transports: Dzelzceļa transports. - 25. sēj. (2007), 164 – 170. p.;
5. D. Sergeyev, P. Gavrilov "Analysis of causes of wagons derailment", RTU zinātniskie raksti. 6. sēr., Mašīnzinātne un transports: Dzelzceļa transports. - 25. sēj. (2007), 171 – 176. p.;
6. Gavrilov P., Sergeyev D. „Derailments of wagons during sorting on marshalling humps at the hump yards", V Konferencija Naukowo Techiczna, Poland, Szczyrk, 2008. Reports of conference: 92 – 92. p.;
7. Гаврилов П., Сергеев Д. "О сходах вагонов при маневровой работе", TRANS-MECH-ART, МИИТ, Москва, 2008, Труды: 35 – 37 стр.;
8. П. Гаврилов, Д. Сергеев "Определение момента сопротивления повороту тележки грузового вагона в опорном узле", RTU zinātniskie raksti. 6. sēr., Mašīnzinātne un transports: Dzelzceļa transports. - 30. sēj. (2008). - 37 – 40. lpp.;
9. Д. Сергеев, А. Сергеев, П. Гаврилов "Колебания вагонов, тормозимых замедлителем, при роспуске отцепов с горки на сортировочной станции", RTU zinātniskie raksti. 6. sēr., Mašīnzinātne un transports: Dzelzceļa transports. - 30. sēj. (2008). - 41 – 48. lpp.;

10. Gavrilov P., Sergeyev D. „Resisting moment in the abutment to turning of freight car bogie", VIII Scientific Conferece TLTS'08, Poland, Katowice-Cieszyn, 2008. Reports of conference: 259 – 264. p.;
11. Гаврилов П., Сергеев Д. „Экспериментальное определение момента сопротивления вращению тележки грузового вагона в подпятниковом узле", II международная научно-практическая конференция, «Проблемы и перспективы развития транспортных систем и строительного комплекса», Белоруссия, 2008, Материалы докладов II-ой международной научно-технической конференции: 180 – 182 стр.;
12. D. Sergeyev, A. Sergeyev, P. Gavrilov „Determination of amplitudes at oscillation of carriages on brake position of moderator-coolant", 13TH International conference on computer systems aided science, industry and transport, konference **„**TRANSCOMP 2009**"**, Zakopane, Poland, 2009. Reports of conference: 226 – 231. p.

7. STRUCTURE OF WORK

The work consists of introduction, 4 chapters, conclusions, list of references and annexes.

Introduction includes thesis relevance argumentation; objective, scientific novelty and practical value of work are conceived.

7.1. Status of the question. Problem investigation

In the first chapter, the statistics and analysis of freight wagon derailment on Latvian railway is presented for the period of 1997-2009 (Fig. 1.1).

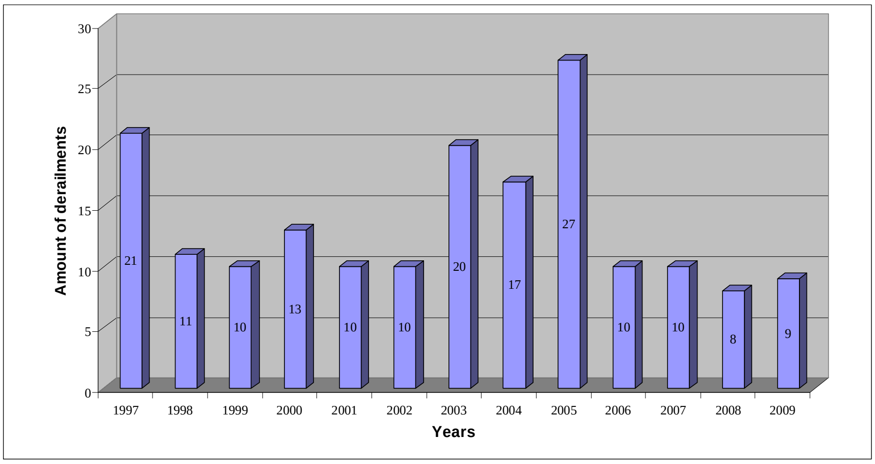


Fig. 1.1. Statistics of total amount of freight wagon derailment on Latvian railway from 1997 to 2009

Amount of loaded and empty wagon derailment from 1997 to 2009 is shown in Fig. 1.2. Statistics shows the prevalence of empty wagon derailment. Collected data gives opportunity to see that there are cases of derailment of wagons with high-positioned centres of gravity. Gravity centre of tank-wagon is situated 2,820 m above rail head, so for open wagons this value reaches 1,742 m, for covered wagons – 2,344 m, for ballast cars – 2,326 m. The main part of derailments is referred to open wagons and tank-wagons.

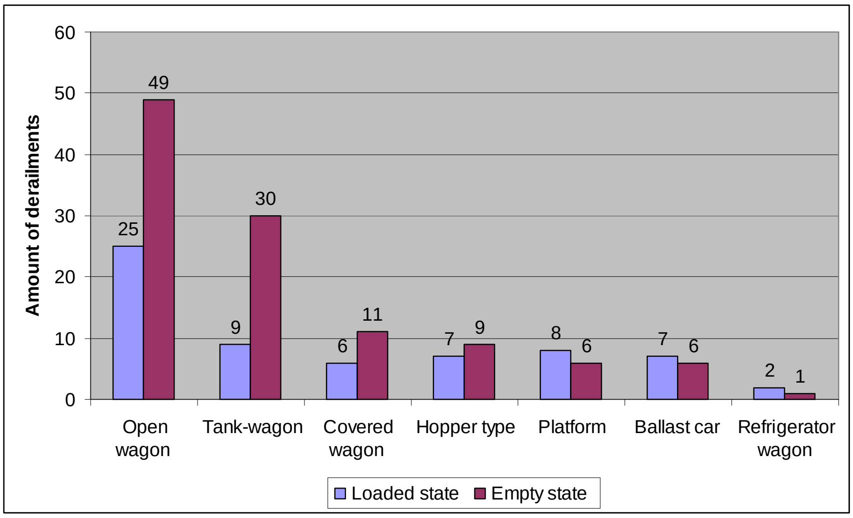


Fig. 1.2. Wagon derailment ratio depending on weight

Statistical data shows that the main part of derailments (41%) happened in period from June to August, while dry weather. Dry weather wagon derailments are caused by a higher friction coefficient between rail head and wheel pair wheel. Dry weather promotes increase of friction coefficient both on side surface of the rail and on thread surface. Due to friction increase in contact area of system "wheel-rail", conditions appear for rolling of wheel flange on rail head. The least number of derailments happened when damp weather – 14% of total wagon derailments (Fig. 1.3).

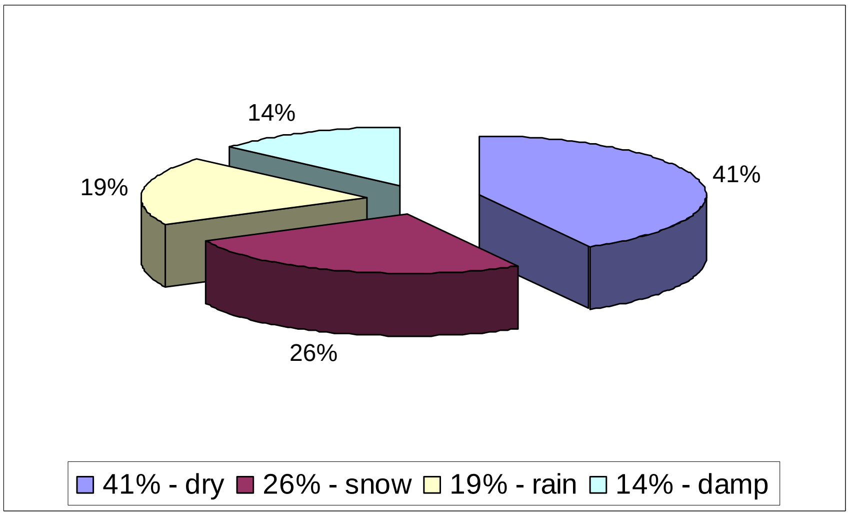


Fig. 1.3. Connection between wagon derailment and weather conditions

According to statistics shown in Fig. 1.4, the biggest amount of derailments happened when detaching cuts at gravity humps and when passing curvilinear space intervals.

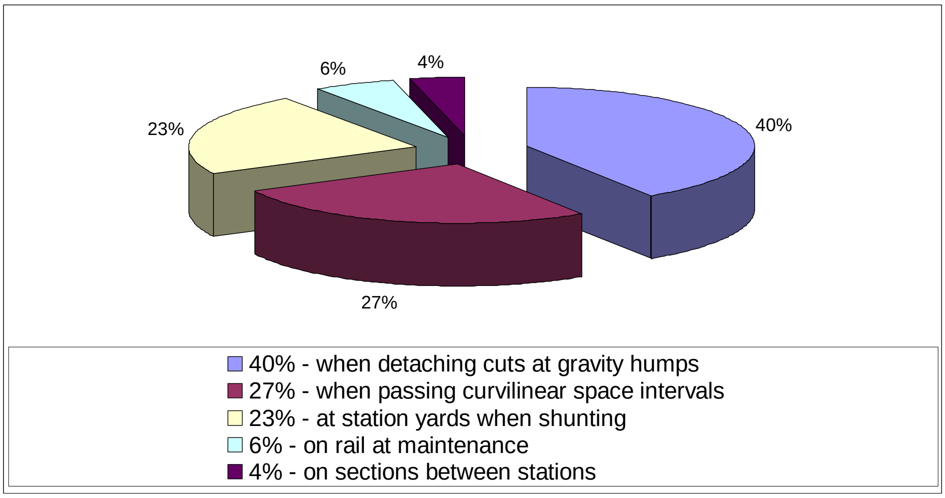


Fig. 1.4. Allotment of freight wagon derailment from 1997 to 2009

Term “wagon derailment” is used quite informally and, due to this, can be understood too generally. V. Lisyuk in [41] distinguishes wheel pair derailments due to rolling of flange on rail head, due to temperature overshoot of rail, due to spreading of rail, due to car lift and finally due to rail breakage. When discussing problem of derailments it is advised to use a more precise term – “derailment of wagon wheel pair”. In given work derailment due to rolling of wheel flange over rail head is of particular interest.

An assumption was set forward that one of the probable reasons for wheel pair derailment when passing curvilinear space intervals could be improper condition of wagon body and bogie supporting nodes (Fig. 1.5). Examination of bogie bearing nodes of 17 tank-wagons, 16 open wagons, 18 covered wagons, 14 platforms and 14 grain carrier wagons sent for roundhouse servicing and overall repair in railroad car shed of Daugavpils city showed that 80…90% had no lubricant in supporting nodes, had score marks and irregular wear of friction surfaces “centre plate – bearing”, regardless of wagon type. This was discovered upon condition of wagon maintenance regulations supposing presence of lubricant in supporting nodes.


Fig. 1.5. Wagon derailment due to bogie jamming

7.2. Freight wagon bogie rotation resistance of the impact relative to its body

In second chapter experimental researches were conducted in order to figure out the reasonableness of improper condition of supporting nodes – model 18-100 freight wagon bogie rotation antitorque moment in bearing node was defined.

Wagon passage of curvilinear space intervals is followed by turn of centre plate bogie towards bearing. The bearing is under vertical load of 15 – 47 tf. Dry friction between centre bowl and bearing creates moment of friction forces that prevents bogie from turning. Thereat, resisting forces in bearing prevent wagon bogie from turning when moving from curvilinear space interval to straight one.

Information required to define bogie rotation antitorque moment towards wagon body was obtained using specially created laboratory assembly, shown in Fig. 2.1. The following laboratory assembly allowed modelling freight wagon bolster supporting node operating conditions when passing curvilinear space intervals.

In the conducted experiment, load on bearing was transmitted by centre bowl that was attached to wagon frame model. Wagon frame model as welded from two channel beams of mass $m_0 = 200kg$. Symmetric freights of mass m_1 (Fig. 2.1) were placed on frame model. Two cases of centre bowl node loading were considered. In first case, the freight was represented by wheel pair of passenger wagon weighting 1400 *kg*. In the second case, freight of 450 *kg* was used.

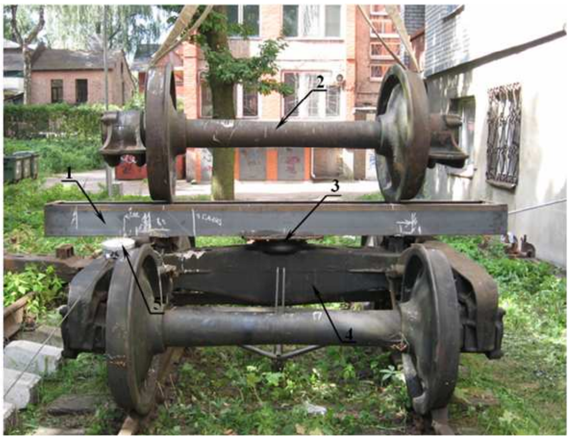


Fig. 2.1. Experimental assembly for defining moment by bogie rotation resisting force in supporting node:
1 – frame, 2 – freight, 3 – central supporting node, 4 – model 18-100 bogie,
5 – dynamometer and cable connected to jack

By force *P*, applied to frame model at the distance of $d = 0{,}75m$ from the central supporting node, rotation antitorque moment was defined, caused by friction in supporting node. Value of force *P*, providing a visually recorded displacement in moment of wagon body frame model displacement with freight towards bolster. The registered displacement was frame model rotation around the centre of supporting node.

Scheme of experimental measurement assembly is shown in Fig. 2.2.

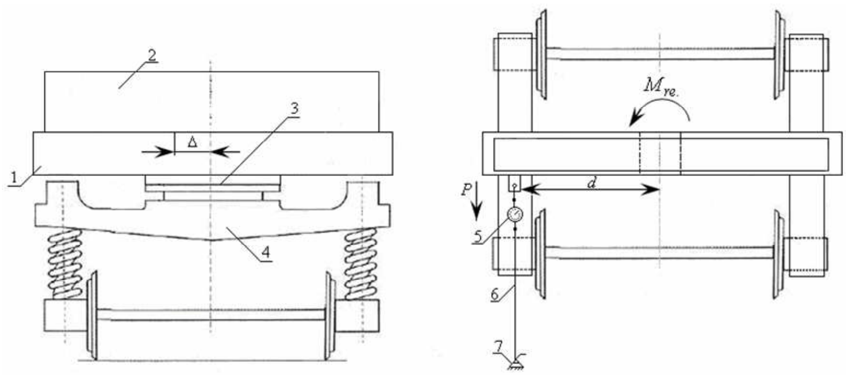
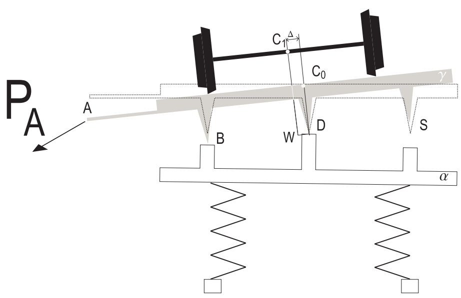
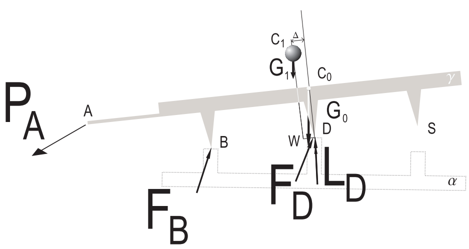


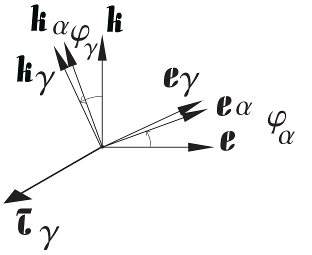
Fig. 2.2. Scheme of experimental assembly:
1 – body frame model; 2 – freight; 3 – bogie supporting node; 4 – model 18-100 freight bogie; 5 – dynamometer; 6 – cable; 7 – jack

Variation of parameter Δ referred to distance change between freight mass centre and track centreline. This was a way of modelling the supposed displacement of freight in wagon when passing curvilinear space intervals. Analytical model of laboratory assembly (Fig. 2.1) is shown in Fig. 2.3. When identifying loading on scheme Fig. 2.3, $P = P_A$ is used. Lower index indicates application point. In particular, probing force P, applied in point A, is noted as P_A.



a)





b) *c*)

Fig. 2.3. Analytical model for defining balance conditions of laboratory assembly, loaded with probing effect of force $\mathbf{P}_A$;
α – bolster, γ – wagon body frame model
$\boldsymbol{\tau}_\gamma, \mathbf{e}_\gamma, \mathbf{k}_\gamma$ – basis rigidly bound with wagon frame model,
$\boldsymbol{\tau}_\alpha, \mathbf{e}_\alpha, \mathbf{k}_\alpha$ – basis rigidly bound with bolster, φ_α – angle of bolster rotation around crossheading $\boldsymbol{\tau}$, φ_γ – angle of frame rotation around crossheading $\boldsymbol{\tau}$

A system of five equations of laboratory assembly balance is the following:

$$
\begin{aligned}
&0 = P_A - F_B^{fr} + N_D^{\tau}\\
&0 = -G_0 \sin\varphi_\alpha - G_1 \sin\varphi_\alpha + N_B^e + N_D^e\\
&0 = -G_0 \cos\varphi_\alpha - G_1 \cos\varphi_\alpha + N_B^k + N_D^k\\
&0 = l_{DC_0} G_0 \sin\varphi_\gamma + l_{DW} G_1 \cos\varphi_\gamma + l_{WC_1} G_1 \sin\varphi_\gamma - l_{DB} N_B^k\\
&0 = l_{DA} P_A - l_{DB} F_B^{fr} - L_D^{fr}
\end{aligned}
\qquad (1)
$$

In (1) P_A – the value of probing force applied to body frame model in point A; F_B^{fr} – the value of friction load appearing in point B, N_B^i, N_D^i – normal reactions in bearer B and centre bowl D; $G_0 = m_0 g$, $G_1 = m_1 g$, where g – gravitation acceleration; φ_α – bolster deflection angle from horizontal; l_{DC_0} = 0,1 m – distance from point D till mass centre of wagon frame; l_{DW} = 0,23 m – distance Δ, at which the freight is moved in the direction of bearer towards the centre; l_{WC_1} = 0,475 м – distance from point W till mass centre of load placed on wagon frame; l_{DB} – distance from point D till bogie bearer, l_{DB} = 0,75 m; l_{DA} – distance from point D till probing force application point, l_{DA} = 0,75 m; φ_γ – frame model angle of deflection from horizontal; L_D^{fr} – point resistive torque appearing in bearing D (Fig. 2.3).

Suggestions about Coulomb type friction offer the following correlations for L_D^{fr} and F_B^{fr}

$$
L_D^{fr} = \theta N_D^k,\ F_B^{fr} = f^{fr} N_B^k \qquad (2)
$$

In (2) θ and f^{fr} – parameters of Coulomb friction.

For supporting reactions N_B^k and N_D^k from (1) we get

$$
\begin{aligned}
N_B^k &= \left[\frac{l_{DC_0}}{l_{DB}} G_0 \sin\varphi_\gamma + \left(\frac{l_{DW}}{l_{DB}}\cos\varphi_\gamma + \frac{l_{WC_1}}{l_{DB}}\sin\varphi_\gamma\right) G_1\right],\\
N_D^k &= \left(\cos\varphi_\alpha - \frac{l_{DC_0}}{l_{DB}}\sin\varphi_\gamma\right) G_0 + \left[\left(\cos\varphi_\alpha - \frac{l_{DW}}{l_{DB}}\cos\varphi_\gamma - \frac{l_{WC_1}}{l_{DB}}\sin\varphi_\gamma\right) G_1\right].
\end{aligned}
\qquad (3)
$$

The fifth equation of system (1) shows that the state of limit equilibrium is disrupted, when the following inequality is satisfied

$$
l_{DA} P_A > L_D^{fr} + l_{DB} F_B^{fr} \qquad (4)
$$

Inequality (4) is a condition of laboratory assembly equilibrium. In the experiment, the satisfaction of such condition is reached through controlled increase of P_A value.

On the left in (4) is the experimentally measured value, on the right – a combination of friction-generated stresses, exerting form wagon body model on bolster. The given correlation allows making conclusions about limit values of support reaction combinations preventing the rotation of bolster (as well as bogie) towards wagon body.

The aim of measurements done using laboratory assembly is the definition of friction parameter values – θ and f^{fr}, depending on supporting node condition.

When wagon frame model is supported by bolster bearer only, the contact between frame and bearer is absent, $F_B^{fr} = 0$. In this case,

$$N_D = G_0 + G_1 \tag{5}$$

That is why correlation (4), taking into account (2), allows relating experimentally measured force P_A through friction parameter θ to weight of load supported by bolster

$$l_{DA} P_A = \theta N_D \tag{6}$$

Program of experiment consisted of the following tests:

– with lubricant in supporting node and on bearers;

– without lubricant in supporting node and on bearers;

– experiments were conducted in case of putting carrying rollers on bearers. Rollers were used in a quality of expansion rollers (Fig. 2.4).

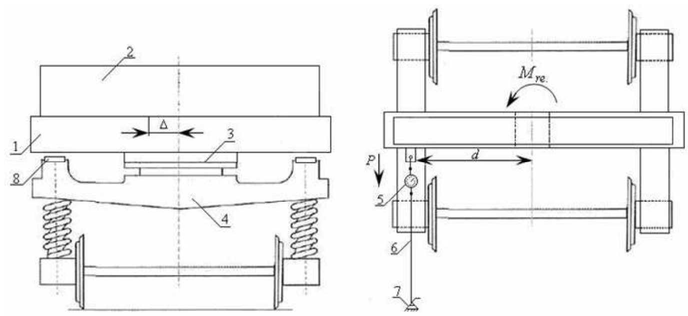


Fig. 2.4. Experimental assembly scheme:
1 – body frame model; 2 – load; 3 – bogie supporting node; 4 – type 18-100 freight bogie; 5 – dynamometer; 6 – cable; 7 – jack; 8 – expansion rollers

Obtained values of parameter θ at symmetric configuration of wagon frame with loads are shown in Table 2.1.

Table 2.1.

Values of friction parameter θ, obtained in various experimental conditions for loads of 1600 kg and 650 kg

Experiment No.	Load weight 15680 N		Load weight 6370 N	
	Coefficient θ with lubricant in supporting node			
	$l_{DA}P_A$, Nm	θ	$l_{DA}P_A$, Nm	θ
1	297.92	0.019	114.66	0.018
2	329.28	0.021	127.40	0.020
3	313.60	0.020	108.29	0.017

Experiment No.	Coefficient θ at dry friction in supporting node			
	$l_{DA}P_A$, Nm	θ	$l_{DA}P_A$, Nm	θ
1	548.80	0.035	210.21	0.033
2	439.04	0.028	229.32	0.036
3	470.40	0.030	222.95	0.035
Experiment No.	Coefficient θ with sand dust in supporting node			
	$l_{DA}P_A$, Nm	θ	$l_{DA}P_A$, Nm	θ
1	721.28	0.046	299.39	0.047
2	658.56	0.042	318.50	0.050
3	705.60	0.045	286.65	0.045

Mean values of parameter θ_{med} for every state of supporting nodes are as follows: with lubricant in supporting node, $\theta_{med} = 0{,}019$; at dry friction in supporting node, $\theta_{med} = 0{,}033$; with sand dust in supporting node, $\theta_{med} = 0{,}046$.

Knowing mean value of parameter θ_{med} and having measured value of P_A, friction parameter f^{fr} is defined for correlation

$$l_{DA}P_A = \theta_{med} N_D^K + f^{fr} l_{DB} N_B^K \tag{7}$$

in which values of normal reactions N_D^K and N_B^K are calculated according to (3), pursuant to Δ and angle $\varphi_\gamma = 0{,}01$.

Obtained values of friction parameter f^{fr}, defined from correlation (7) are presented in Table 2.2.

Table 2.2.

Values of friction parameter f^{fr}, obtained in various experimental conditions for loads of 1600 kg and 650 kg

Load weight 15680 N			Load weight 6370 N		
With lubricant					
$l_{DA}P_A$, Nm	θ_{med}	f^{fr}	$l_{DA}P_A$, Nm	θ_{med}	f^{fr}
467.95	0.019	0.181	125.03	0.019	0,147
At dry friction					
$l_{DA}P_A$, Nm	θ_{med}	f^{fr}	$l_{DA}P_A$, Nm	θ_{med}	f^{fr}
565.95	0.033	0.210	229.50	0.033	0.255
With sand dust					
$l_{DA}P_A$, Nm	θ_{med}	f^{fr}	$l_{DA}P_A$, Nm	θ_{med}	f^{fr}
845.25	0.046	0.301	311.25	0.046	0.328

According to the results of experimental measurements, presented in Table 2.1 and Table 2.2, mean values of friction parameter θ_{med} and parameter f_{med}^{fr} are defined. Mean values θ_{med} and f_{med}^{fr} are presented in Table 2.3.

Table 2.3.

Mean values of friction parameter θ_{med} and f_{med}^{fr}

With lubricant		Dry friction		With sand dust	
θ_{med}	f_{med}^{fr}	θ_{med}	f_{med}^{fr}	θ_{med}	f_{med}^{fr}
0.019	0.164	0.033	0.232	0.046	0.314

Values of friction parameters, presented in Table 2.3, can be used when estimating certain wagon bogie antitorque moment value on curvilinear space intervals. They allow defining antitorque moment M_{re} for arbitrarily loaded freight wagons according to (7).

Results of M_{re} calculation for empty freight wagons are shown in Table 2.4, for loaded wagons – in Table 2.5 accordingly. The calculations were conducted for one bogie, supposed that the load is distributed symmetrically on two wagon bogies for eight-wheel wagons.

Table 2.4.

Antitorque moment values M_{re} for empty wagons

	Antitorque moment M_{re} with lubricant		
	when loaded frame basing on bolster centre bowl only, *Nm*	when loaded frame basing on centre bowl and one of bearers, *Nm*	and with loaded frame basing on centre bowl and rollers, installed on bearer, *Nm*
Tank-wagon	1298.11	1481.24	1264.92
Open wagon	1186.88	4462.14	593.17
Hopper wagon	1186.88	6019.38	310.88
	Antitorque moment M_{re} without lubricant		
	when loaded frame basing on bolster centre bowl only, *Nm*	when loaded frame basing on centre bowl and one of bearers, *Nm*	and with loaded frame basing on centre bowl and rollers, installed on bearer, *Nm*
Tank-wagon	2257.84	2505.73	2200.10
Open wagon	2064.37	6498.00	1031.71
Hopper wagon	2064.37	8606.01	540.72
	Antitorque moment M_{re} with sand dust present		
	when loaded frame basing on bolster centre bowl only, *Nm*	when loaded frame basing on centre bowl and one of bearers, *Nm*	when loaded frame basing on centre bowl and rollers, installed on bearer, *Nm*
Tank-wagon	3143.52	3476.92	3063.13
Open wagon	2874.17	8837.10	1436.42
Hopper wagon	2874.17	11672.23	752.82

Table 2.5.

Antitorque moment values M_{re} for loaded wagons

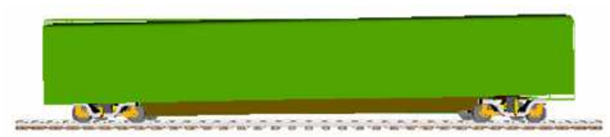
	Antitorque moment M_{re} with lubricant		
	when loaded frame basing on bolster centre bowl only, *Nm*	when loaded frame basing on centre bowl and one of bearers, *Nm Нм*	when loaded frame basing on centre bowl and rollers, installed on bearer, *Nm*
Tank-wagon	6868.46	27576.71	4174.26
Open wagon	7592.43	29091.99	3695.11
Hopper wagon	7512.93	37394.58	1751.50
	Antitorque moment M_{re} without lubricant		
	when loaded frame basing on bolster centre bowl only, *Nm*	when loaded frame basing on centre bowl and one of bearers, *Nm*	when loaded frame basing on centre bowl and rollers, installed on bearer, *Nm*
Tank-wagon	11946.46	40324.65	7260.38
Open wagon	13205.68	42309.11	6426.99
Hopper wagon	11924.67	53404.94	3046.42
	Antitorque moment M_{re} with sand dust present		
	when loaded frame basing on bolster centre bowl only, *Nm*	when loaded frame basing on centre bowl and one of bearers, *Nm*	when loaded frame basing on centre bowl and rollers, installed on bearer, *Nm*
Tank-wagon	16632.71	54873.34	10108.42
Open wagon	18385.89	57528.05	8948.121
Hopper wagon	16933.29	72420.64	4241.45

The results of experiments showed that the state of bolster supporting nodes affects substantively the value of moment influence that prevents bogie rotation around centre pivot. In comparison with friction parameter value θ, referred to properly lubricated supporting nodes, the values of the same parameter appear 1,7…2,4 times bigger when not lubricated. Approximately analogue correlation is obtained for parameter f^{fr}. The presence of sand dust aggravates the case.

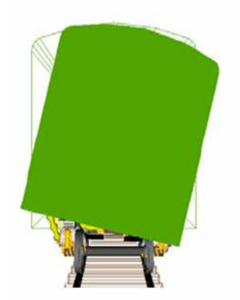
It is recommended to prefer new age bogie wagons having elastic roller bearings on bolsters, because the supporting of wagon body on expansion rollers, when passing curvilinear space intervals, provides 4…6 tuple decrease of bogie antitorque moment towards wagon body by the side of traditional construction. Given provisions will promote a substantial reduction of supporting node, wheel pair flange and rail head side flange wear, as well as will decrease wagon derailment probability when passing curvilinear space intervals.

7.3. Cut braking modelling for sorting hill

According to the statistics, 40% of derailments on Latvian railway happen when detaching wagons at gravity humps. Connection between wagons, the design of spring suspension and other objective circumstances precondition the probability of wagon cut disturbing moments when moving on rail. An inescapable result of cut breaking operation is the appearance of longitudinal dynamics in cut that causes rolling motion and cut car body pitching due to design features (Fig. 3.1).



a) *b*)



c) *d*)

Fig. 3.1. Typical movements of rail carriages:
a – wagon pitching; *b* – bouncing; *c* – lower rolling motion;
d – upper rolling motion

The mentioned wagon body movements are related to wagon spring set deformations (Fig. 3.1). Such deformations are the reason for several spring sets to pass into extended state after being pressed under the dead weight of wagon, and so-called unloading appears. The extent of spring set unloading is one of factors that define the stability of "wheel – rail" contact. The extended state of buffer springs weakens this contact. The passage of buffer springs into extended state can be interpreted as unloading of wheel and rail thread surface. Service experience affords ground for an assumption that unloading eases the rolling of wheel flange over rail head. According to the first and second chapters, the given circumstance is naturally referred to the appearance of favourable conditions for the loss of contact between the wheel and the rail, i.e. with the increase of wheel pair derailment possibility.

In the third chapter differential equations of retarding mechanism-provoked wagon body movements are given, describing the situation when detaching cuts at gravity hump yards.

The movement of chain consisting of four spring borne elements in unit vector plane $\boldsymbol{\tau}$ and $\mathbf{k}$ (Fig. 3.2) is considered as a model. The first and the last bodies of chain modulate the features of cut head and tail. Between them a so-called emergency cut member is situated, consisting of two spring borne elements.

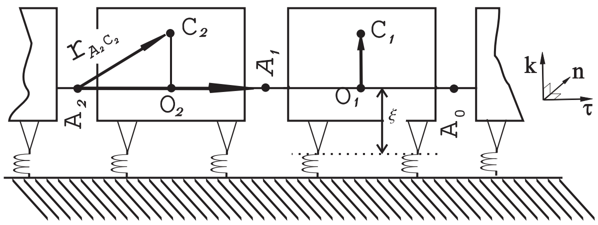


Fig. 3.2. Cut reference configuration before breaking action

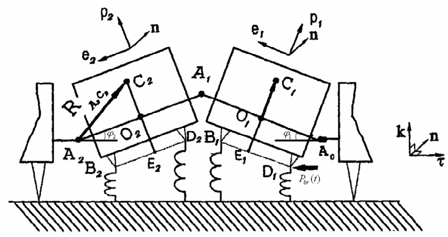


Fig. 3.3. Configuration of breaking cut

The first and the last inertial element of the chain have no opportunity of vertical movements, so the theoretical system has two degrees of freedom: movement of the whole chain along guide rail u_0 and conformal rotation of emergency member elements around axis in parallel with crossheading $\mathbf{n}$ (Fig. 3.3), assumed by $\varphi_1 = \varphi_2 = \varphi$ applied to the system. Elastic forces appearing in spring sets are described as follows:

$$
\begin{aligned}
F_{B_1}^{el} &= F_{B_1}^{el.,st.initial} - c_v (l_{A_0O_1} + l_{E_1B_1}) \sin\varphi \\
F_{D_1}^{el} &= F_{D_1}^{el.,st.initial} - c_v (l_{A_0O_1} - l_{E_1B_1}) \sin\varphi \\
F_{B_2}^{el} &= F_{B_2}^{el.,st.initial} - c_v (l_{A_2O_2} - l_{E_2B_2}) \sin\varphi \\
F_{D_2}^{el} &= F_{D_2}^{el.,st.initial} - c_v (l_{A_2O_2} + l_{E_2B_2}) \sin\varphi
\end{aligned}
\qquad (8)
$$

In (8) $l_{A_0O_1}$ – distance between automatic coupling till the middle of the first wagon of emergency member; $l_{E_1B_1}$ – second part of the first emergency member wagon basis; $l_{A_2O_2}$ – distance between automatic coupling till the middle of the second wagon in emergency element; $F_{D_1}^{el.st.init.}$, $F_{B_1}^{el.st.init.}$ – elastic power of static compression of the first and the second bogie of the first wagon in emergency element; $F_{D_2}^{el.st.init.}$, $F_{B_2}^{el.st.init.}$ – elastic power of static compression of the first and the second bogie of the second wagon in emergency element, φ – turning angle of every wagon in emergency member; c_v – vertical spring stiffness (Fig. 3.3).

Lagrange's equations of the second kind used to conclude equations of movements in spring borne chain with two degrees of freedom led to a system of the following system of differential equation system:

$$
\begin{aligned}
&\dot{u}_0 = u_1, \quad \dot{\varphi} = \psi_1, \\
&\dot{\psi}_1 = -(a_{11}a_{22} - a_{12}a_{21})^{-1}(-a_{21}P_{\Sigma}^{br} - a_{11}g\varphi) \\
&\dot{u}_1 = (a_{11}a_{22} - a_{12}a_{21})^{-1}(-a_{22}P_{\Sigma}^{br} - a_{12}g\varphi)
\end{aligned}
\qquad (9)
$$

Here the following notations are used:

$$
\begin{aligned}
&a_{11} = (m_f + m_1 + m_2 + m_r) \\
&a_{12} = [m_1 z_{O_1C_1} - m_2 z_{O_2C_2}] \\
&a_{21} = m_1(l_{A_0O_1}\varphi + z_{O_1C_1}) + m_2((l_{A_0A_1} + l_{A_1O_2})\varphi - z_{O_2C_2}) + m_r(l_{A_0A_1} + l_{A_1A_2})\varphi \\
&a_{22} = m_1(l^2_{A_0O_1} + z^2_{O_1C_1}) + J^{C_1}_{1n} + J^{C_2}_{2n} + m_2 l^2_{A_0A_1} - 2m_2 l_{A_0A_1} l_{A_1O_2} + m_r(l^2_{A_0A_1} + l^2_{A_1A_2}) \\
&P^{br}_{\Sigma} = -P^{br}_{D_1} - P^{br}_{B_1} - P^{br}_{D_2} - P^{br}_{B_2}
\end{aligned} \tag{10}
$$

$$
\begin{aligned}
g = {} & G_1 z_{O_1C_1} + G_2 z_{O_2C_2} + G_1 l_{O_1E_1} - G_2 l_{O_2E_2} - \\
& - c_v (l_{A_0O_1} - l_{E_1B_1})^2 - c_v (l_{A_0O_1} + l_{E_1B_1})^2 - \\
& - c_v (l_{A_2O_2} + l_{E_2B_2})[l_{A_0A_1} - l_{A_1O_2} + l_{E_2D_2}] - \\
& - c_v (l_{A_2O_2} - l_{E_2B_2})[l_{A_0A_1} - l_{A_1O_2} - l_{E_2B_2}].
\end{aligned} \tag{11}
$$

Initial conditions corresponding to the breakage of undeformed cut that reaches retarder position at a speed of v_0, are

$$
u_0(0) = 0, \varphi(0) = 0, \psi_1(0) = 0, u_1(0) = v_0. \tag{12}
$$

The abovementioned system of two differential (9) equations is easily calculated using the features of MathCad program. The obtained equations fully describe the movement of cut along the way on an interval after retarder position. Thereat, in process of calculation not only the information about amplitude of transversal car cut oscillations is obtained, which allows estimating regime of breaking in view of providing a wheel pair derailment safety, but defines the movement of the cut along the guide as a whole. So, equation data can be used to solve tasks of target breaking. They allow predicting longitudinal speed of the cut as a result of breaking intensity on space interval after position of brake application.

As the contact between the wheel and the rail in the considered system shown in Fig. 3.2 and Fig. 3.3 is irretentive, operable buffer springs of any emergency member wagon must remain abutted. Static deformation of spring suspension Δ_{st}, made by wagon weight, is designed basing on conditions providing required ride performance of wagon. Herewith, oscillation frequency of wagon body should be reduced. The least is reached through increase of spring suspension compliance. Unloading requires dynamic deformation of buffer springs while oscillating on the stage of extension to exceed static compression of buffer springs by its own weight, i.e. cumulative value in brackets of elastic power formulas (8) must turn negative from positive. It is obvious that when checking this condition through equations (9), monitoring of $F^{el}_{D_2}$ in formulas (8) is absolutely enough. Regulations offer the following values of unloading levels for freight wagon bogies in loaded state $\Delta_{\text{st.gr.}}$ = 0.045...0.065 m, in empty state $\Delta_{\text{st.t.}}$ = 0.006...0.009 m.

Spring kit relief angle provides the following expression:

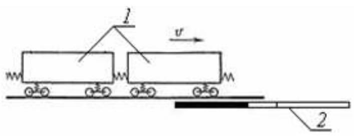
$$
\psi_{st} = \frac{\Delta_{st}}{(l_{A_1O_0} + l_{E_1B_1})} \tag{13}
$$

Where Δ_{st} – bogie spring kit statistical curve; $l_{A_0O_1}$ – distance between automatic coupling till the middle of the first wagon of emergency member; $l_{E_1B_1}$ – second part of the first emergency member wagon basis.

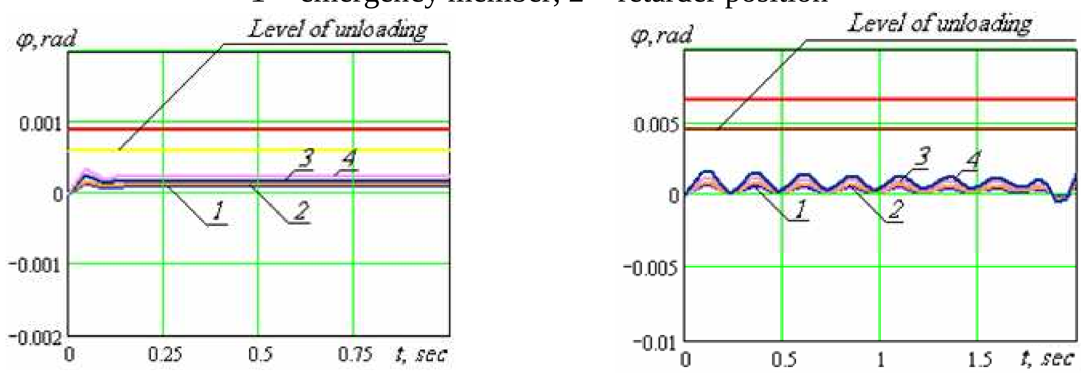
7.4. Volatility study brake line

The fourth chapter offers solutions of differential equations using MATH CAD features. Cut length, types and models of freight wagons and their loading level varied when solving differential equations. Braking force value variations due to friction coefficient change, depending on weather and car set were also considered. Friction coefficient decreases substantially in winter, when damp weather and because of transported items (salt, sugar, potash fertilizers, etc.), which appear on wheel pair and decrease friction coefficient. Cuts of two, three, four, six and ten wagons were explored to define oscillation amplitudes. Both loaded and empty cuts were considered. Situations when friction coefficients between the wheel and brake bar is 0,08; 0,1; 0,15 and 0,2.

Fig. 4.1. shows the location of two wagon cut on retarder position. Below are the amplitude variation of the angle φ, arising from the occurrence of empty and laden unhooked from the tanks into the braking position of the moderator, when the coefficient of friction between the pair of wheels and brake system bus is equal to 0,08; 0,1; 0,15; 0,2.



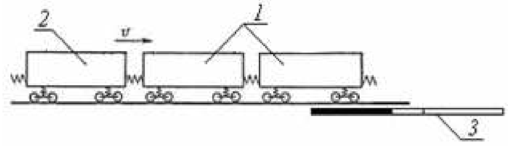
1 – emergency member; *2* – retarder position



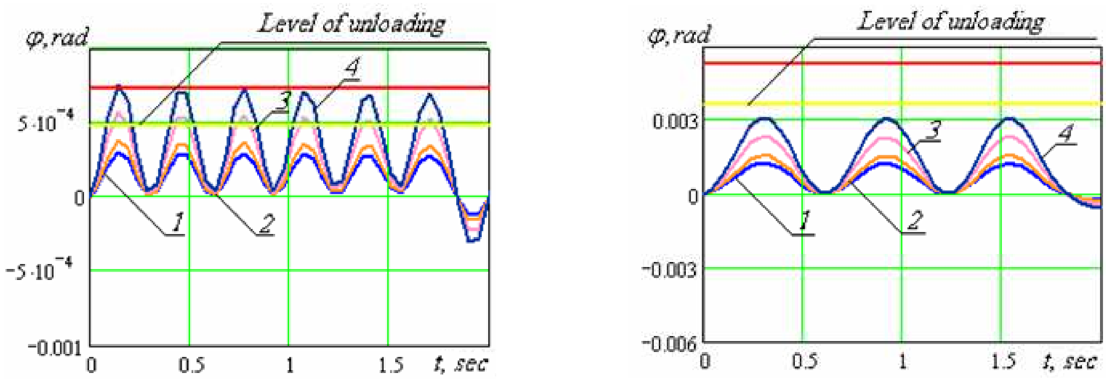
I) *II*)

Fig. 4.1. Amplitudes of angle φ changes for the first wagon of cut, consisting of 2 *tank-wagons,* approaching retarding interval at the speed of v_0, appearing as a result of braking:
1 line, coefficient of friction between the pair of wheels and brake system bus – 0,08;
2 line, coefficient of friction between the pair of wheels and brake system bus – 0,1;
3 line, coefficient of friction between the pair of wheels and brake system bus – 0,15;
4 line, coefficient of friction between the pair of wheels and brake system bus – 0,2;
I – empty cut; *II* – loaded cut

Fig. 4.2 and 4.3 show two locations of retarder positions of cut emergency member, consisting of three wagons. Oscillation amplitudes of angle φ change appearing when empty and loaded cut of covered wagon are entering retarding interval, when friction coefficient between wheel pair and brake bar is 0,08; 0,1; 0,15; 0,2, are presented below.



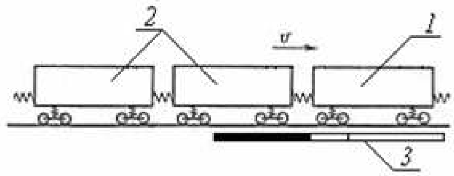
1 – emergency member; *2* – tail part; *3* – retarder position



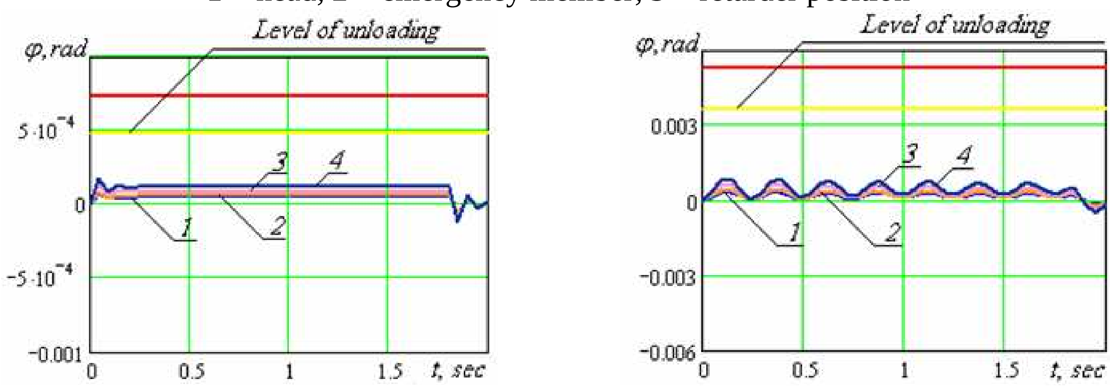
I) *II*)

Fig. 4.2. Amplitudes of angle φ changes for the first wagon of cut, consisting of 3 *covered wagons*, approaching retarding interval at the speed of v_0, appearing as a result of braking:

1 line, coefficient of friction between the pair of wheels and brake system bus – 0,08;
2 line, coefficient of friction between the pair of wheels and brake system bus – 0,1;
3 line, coefficient of friction between the pair of wheels and brake system bus – 0,15;
4 line, coefficient of friction between the pair of wheels and brake system bus – 0,2;
I – empty cut; *II* – loaded cut



1 – head; *2* – emergency member; *3* – retarder position

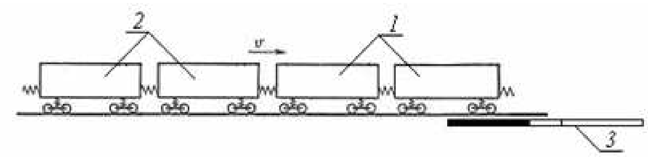


I) *II*)

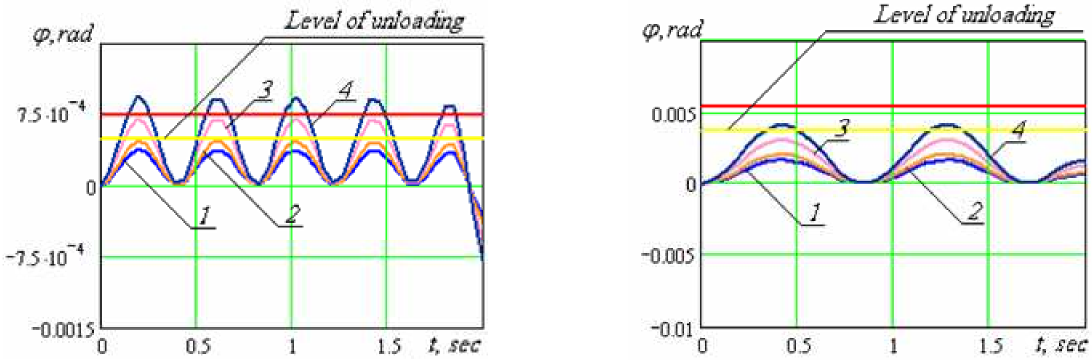
Fig. 4.3. Amplitudes of angle φ changes for the first wagon of cut, consisting of 3 *covered wagons*, approaching retarding interval at the speed of v_0, appearing as a result of braking:

1 line, coefficient of friction between the pair of wheels and brake system bus – 0,08;
2 line, coefficient of friction between the pair of wheels and brake system bus – 0,1;
3 line, coefficient of friction between the pair of wheels and brake system bus – 0,15;
4 line, coefficient of friction between the pair of wheels and brake system bus – 0,2;
I – empty cut; *II* – loaded cut

Fig. 4.4, 4.5, 4.6 show the location of emergency member cut of four wagons on retarder position. Oscillation amplitudes of angle φ changes appearing when empty and loaded cuts of four platform are entering retarding interval, when friction coefficient between wheel pair and brake bar is 0,08; 0,1; 0,15; 0,2.



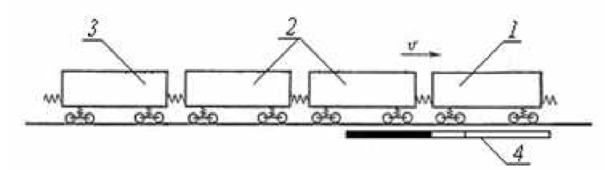
1 – emergency member; *2* – tail; *3* – retarder position



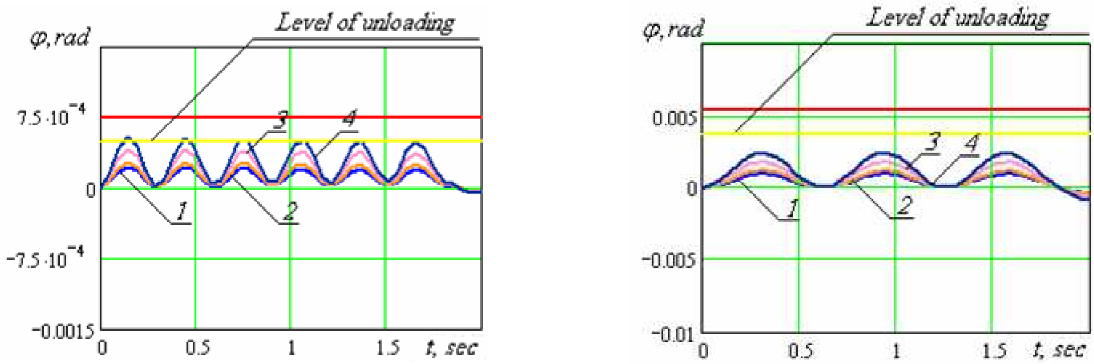
I) *II*)

Fig. 4.4. Amplitudes of angle φ changes for the first wagon of cut, consisting of 4 *platform*, approaching retarding interval at the speed of v_0, appearing as a result of braking:

1 line, coefficient of friction between the pair of wheels and brake system bus – 0,08;
2 line, coefficient of friction between the pair of wheels and brake system bus – 0,1;
3 line, coefficient of friction between the pair of wheels and brake system bus – 0,15;
4 line, coefficient of friction between the pair of wheels and brake system bus – 0,2;
I – empty cut; *II* – loaded cut



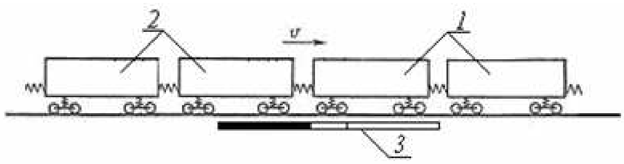
1 – head; *2* – emergency member; *3* – tail; *4* – retarder position



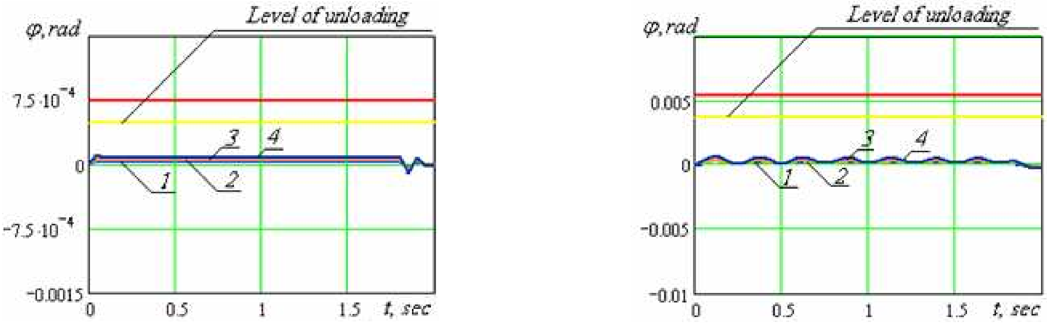
I) *II*)

Fig. 4.5. Amplitudes of angle φ changes for the second wagon of cut, consisting of 4 *platform*, approaching retarding interval at the speed of v_0, appearing as a result of braking:

1 line, coefficient of friction between the pair of wheels and brake system bus – 0,08;
2 line, coefficient of friction between the pair of wheels and brake system bus – 0,1;
3 line, coefficient of friction between the pair of wheels and brake system bus – 0,15;
4 line, coefficient of friction between the pair of wheels and brake system bus – 0,2;
I – empty cut; *II* – loaded cut



1 – head; *2* – member; *3* – retarder position

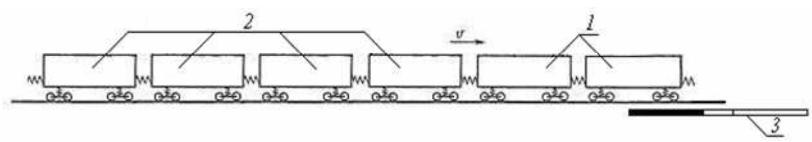


I) *II*)

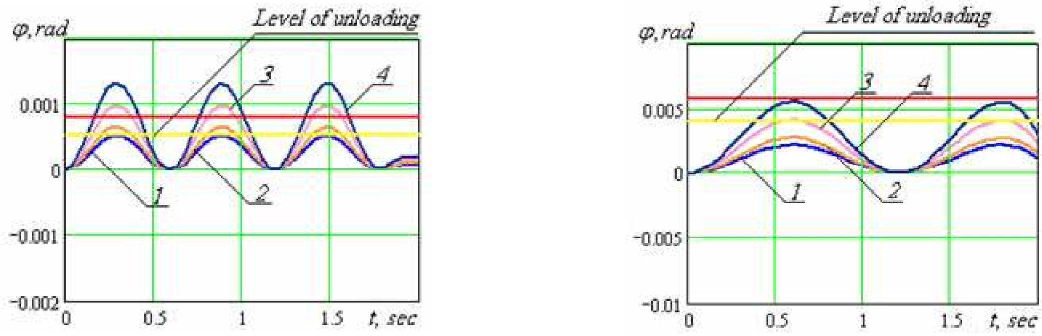
Fig. 4.6. Amplitudes of angle φ changes for the third wagon of cut, consisting of 4 *platform*, approaching retarding interval at the speed of v_0, appearing as a result of braking:

1 line, coefficient of friction between the pair of wheels and brake system bus – 0,08;
2 line, coefficient of friction between the pair of wheels and brake system bus – 0,1;
3 line, coefficient of friction between the pair of wheels and brake system bus – 0,15;
4 line, coefficient of friction between the pair of wheels and brake system bus – 0,2;
I – empty cut; *II* – loaded cut

Fig. 4.7, 4.8, 4.9 show the location of emergency member cut of six gondola on retarder position. Cases of braking were considered for the first, the third and the fifth cut wagons. Oscillation amplitudes of angle φ changes appearing when cut is entering retarding interval, when friction coefficient between wheel pair and brake bar is 0,08; 0,1; 0,15; 0,2.



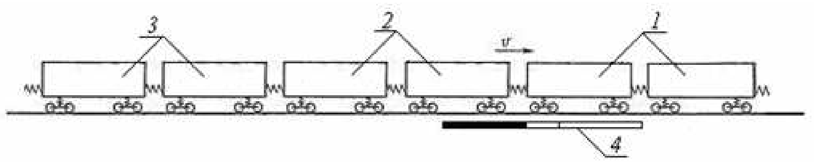
1 – emergency member; *2* – tail; *3* – retarder position



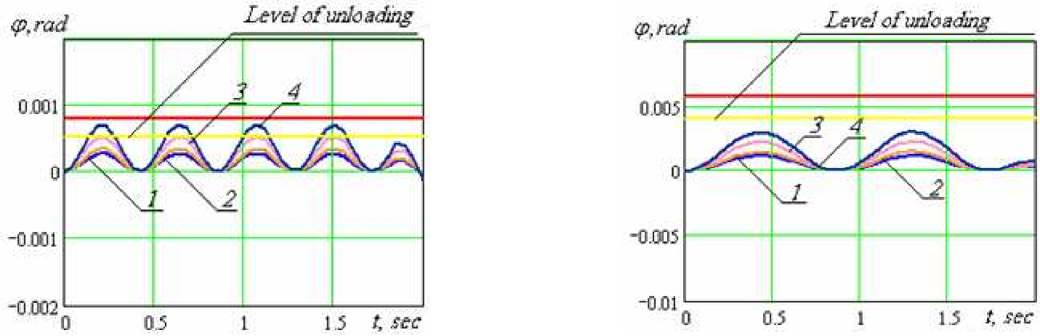
I) *II*)

Fig. 4.7. Amplitudes of angle φ changes for the third wagon of cut, consisting of 6 *open wagon*, approaching retarding interval at the speed of v_0, appearing as a result of braking:

1 line, coefficient of friction between the pair of wheels and brake system bus – 0,08;
2 line, coefficient of friction between the pair of wheels and brake system bus – 0,1;
3 line, coefficient of friction between the pair of wheels and brake system bus – 0,15;
4 line, coefficient of friction between the pair of wheels and brake system bus – 0,2;
I – empty cut; *II* – loaded cut

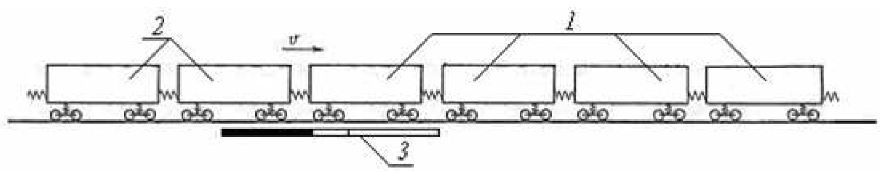


1 – head; *2* – emergency member; *3* – tail; *4* – retarder position

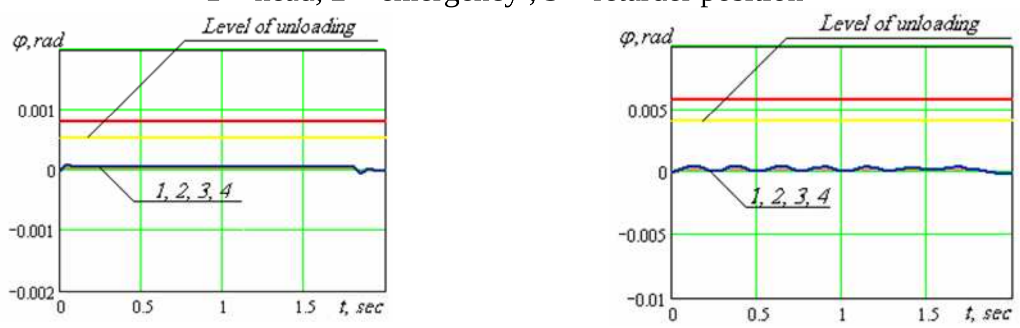


I) *II*)

Fig. 4.8. Amplitudes of angle φ changes for the third wagon of cut, consisting of 6 *open wagon*, approaching retarding interval at the speed of v_0, appearing as a result of braking:
1 line, coefficient of friction between the pair of wheels and brake system bus – 0,08;
2 line, coefficient of friction between the pair of wheels and brake system bus – 0,1;
3 line, coefficient of friction between the pair of wheels and brake system bus – 0,15;
4 line, coefficient of friction between the pair of wheels and brake system bus – 0,2;
I – empty cut; *II* – loaded cut



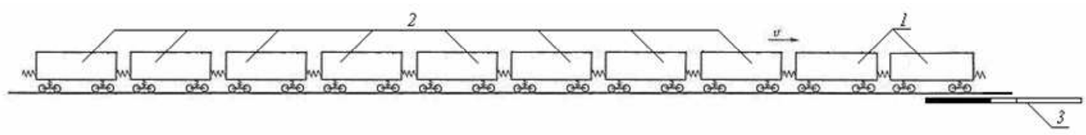
1 – head; *2* – emergency ; *3* – retarder position



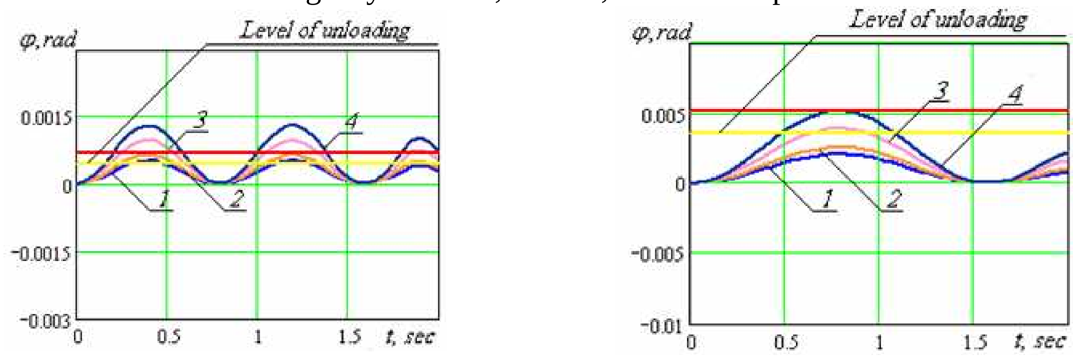
I) *II*)

Fig. 4.9. Amplitudes of angle φ changes for the third wagon of cut, consisting of 6 *open wagon*, approaching retarding interval at the speed of v_0, appearing as a result of braking:
1 line, coefficient of friction between the pair of wheels and brake system bus – 0,08;
2 line, coefficient of friction between the pair of wheels and brake system bus – 0,1;
3 line, coefficient of friction between the pair of wheels and brake system bus – 0,15;
4 line, coefficient of friction between the pair of wheels and brake system bus – 0,2;
I – empty cut; *II* – loaded cut

Fig. 4.10, 4.11, 4.12 show the location of emergency member cut of ten hoper cut on retarder position. Cases of braking were considered for the first, the fifth and the ninth cut wagons. Oscillation amplitudes of angle φ changes appearing when cut is entering retarding interval, when friction coefficient between wheel pair and brake bar is 0,08; 0,1; 0,15; 0,2.

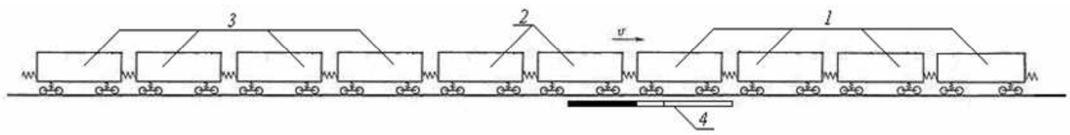


1 – emergency member; *2* – tail; *3* – retarder position

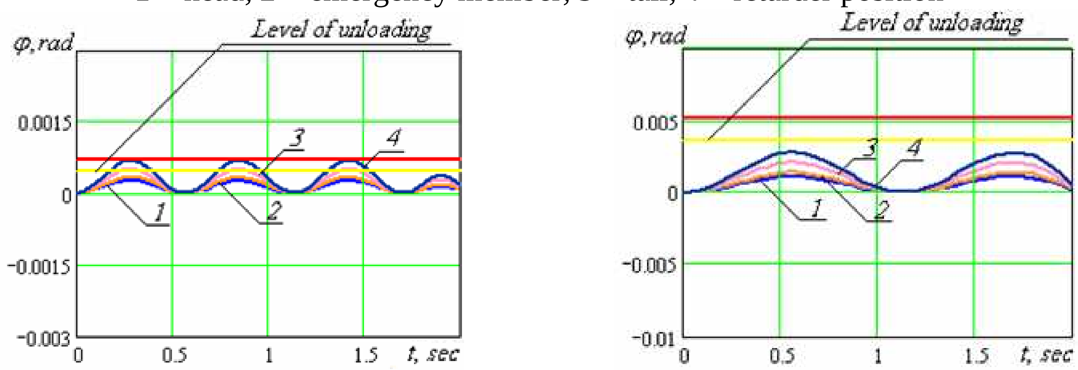


I) *II*)

Fig. 4.10. Amplitudes of angle φ changes for the first wagon of cut, consisting of 10 *hoper cut*, approaching retarding interval at the speed of v_0, appearing as a result of braking:
1 line, coefficient of friction between the pair of wheels and brake system bus – 0,08;
2 line, coefficient of friction between the pair of wheels and brake system bus – 0,1;
3 line, coefficient of friction between the pair of wheels and brake system bus – 0,15;
4 line, coefficient of friction between the pair of wheels and brake system bus – 0,2;
I – empty cut; *II* – loaded cut

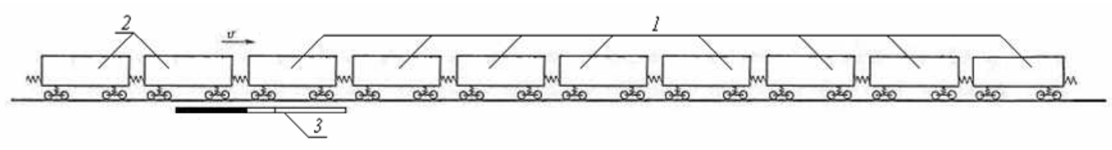


1 – head; *2* – emergency member; *3* – tail; *4* – retarder position

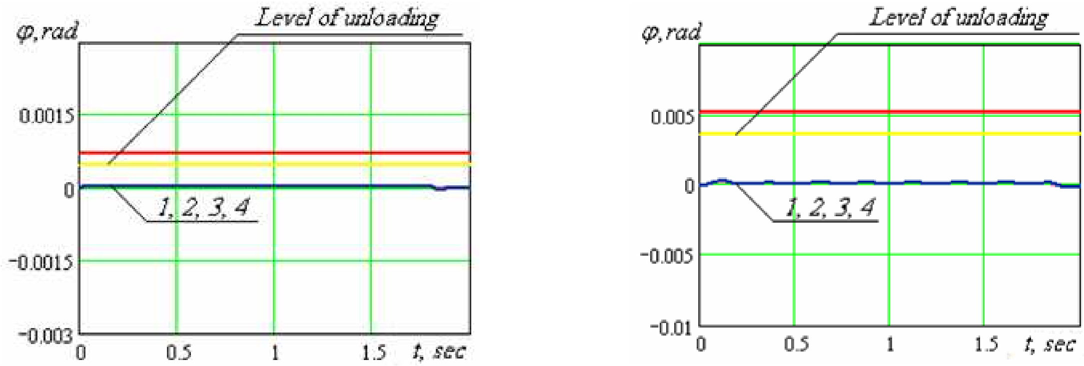


I) *II*)

Fig. 4.11. Amplitudes of angle φ changes for the fifth wagon of cut, consisting of 10 *hoper cut*, approaching retarding interval at the speed of v_0, appearing as a result of braking:
1 line, coefficient of friction between the pair of wheels and brake system bus – 0,08;
2 line, coefficient of friction between the pair of wheels and brake system bus – 0,1;
3 line, coefficient of friction between the pair of wheels and brake system bus – 0,15;
4 line, coefficient of friction between the pair of wheels and brake system bus – 0,2;
I – empty cut; *II* – loaded cut



1 – head; *2* – emergency member; *3* – retarder position



I) *II*)

Fig. 4.12. Amplitudes of angle φ changes for the ninth wagon of cut, consisting of 10 *hoper cut,* approaching retarding interval at the speed of v_0, appearing as a result of braking:
1 line, coefficient of friction between the pair of wheels and brake system bus – 0,08;
2 line, coefficient of friction between the pair of wheels and brake system bus – 0,1;
3 line, coefficient of friction between the pair of wheels and brake system bus – 0,15;
4 line, coefficient of friction between the pair of wheels and brake system bus – 0,2;
I – empty cut; *II* – loaded cut

Obtained results give ground for an assumption that the most derailment-safe braking is to be applied in the middle of the cut. Basing on calculations, the following recommended guidelines are worked out to decrease the number of derailments when detaching cuts at gravity humps:

1. When detaching cuts in dry weather and when friction coefficient between wheel pair and brake bar is 0,2, the brakes must not be applied to the first bogie of the wagon entering lifter retarder. The following is related to cuts consisting of 3…6 empty wagons and cuts of 5…8 loaded wagons.
2. Braking in the middle of the cut is the most derailment-safe and effective solution.
3. When detaching cuts in snowy and rainy weather, as well as in case of detaching wagons with rusty, freshly painted wheel rims and wagons carrying molasses, friction coefficient is reduced to 0,08...0,15 when braking with retarder. This is the reason why unloading of spring suspension does not happen when applying brakes on the first bogie of the first wagon, because the applied impulse of braking is reduced substantially. The main task here is the reduction of speed to the predetermined value.

8. CONCLUSION

The work offers solutions for the following assigned tasks:

1. The statistics of freight wagon derailment on Latvian railway in period 1997-2009 was collected, analyzed and systematized in order to estimate the extent of problem aggravation. 40% of derailments referred to cut detachment at gravity yards, 27% of derailments happen when passing curvilinear space intervals.

2. The state of freight wagon bogie supporting nodes was examined of wagons sent for roundhouse servicing and overall repair in Daugavpils city engine house. Research showed that freight wagon bogie supporting nodes work in circumstances of dry friction at high contact pressures between repairs. Investigation of roller bearing nodes showed that horizontal and vertical work surfaces of centre pivot and roller bearing wear unequally; there are score marks where sand dust collects. There are also marks of dry friction, unequal wear of vertical surfaces "centre pivot-bearing". As a result of investigations, it was stated that the most intensive wear of pivot supporting surface is situated on internal side of bolster (pivot surface inward wagon).
3. An experimental assembly is designed, allowing an opportunity of defining parameters that describe friction between bolster and wagon frame in supporting nodes. The following mean values were received for parameter θ_{med}, depending on supporting node condition:

   - with lubricant in supporting node, $\theta_{med} = 0{,}019$;
   - at dry friction in supporting node, $\theta_{med} = 0{,}033$;
   - with sand dust in supporting node, $\theta_{med} = 0{,}046$.

Mean values for parameter f_{med}^{fr} when frame supported by bolster supporting node and one of bearers are as follows:

- with lubricant in supporting node, $f_{med}^{fr} = 0{,}164$;
- at dry friction in supporting node, $f_{med}^{fr} = 0{,}232$;
- with sand dust in supporting node, $f_{med}^{fr} = 0{,}314$.

4. Experimentally defined Coulomb friction parameters in supporting nodes allowed defining bogie rotation antitorque moment towards wagon body of various types. Antitorque rotation moments for empty tank are as follows:
   - *with lubricant* and when loaded frame supported only by bolster bearing –1298,11*Nm*, when loaded frame supported by bolster bearing and one of bearers – 1481,24 *Nm*, when loaded frame supported by bolster and carrying rollers – 1264,92 *Nm*;
   - *without lubricant* when loaded frame supported only by bolster bearing –2257,84 *Nm*, when loaded frame supported by bolster bearing and one of bearers – 2505,73 *Nm*, when loaded frame supported by bolster and carrying rollers – 2200,10 *Nm*;
   - *with fine sand dust* and when loaded frame supported only by bolster bearing – 3143,52*Nm*, when loaded frame supported by bolster bearing and one of bearers – 3476,92 *Nm*, when loaded frame supported by bolster and carrying rollers – 3063,13 *Nm*;

   Antitorque rotation moments for loaded tank are as follows:
   - *with lubricant* and when loaded frame supported only by bolster bearing – 6868,46 *Nm*, when loaded frame supported by bolster bearing and one of bearers – 27576,71 *Nm*, when loaded frame supported by bolster and carrying rollers – 4174,26 *Nm*;
   - *without lubricant* when loaded frame supported only by bolster bearing –11946,46 *Nm*, when loaded frame supported by bolster bearing and one of bearers – 40324,65 *Nm*, when loaded frame supported by bolster and carrying rollers – 7260,38 *Nm*;
   - *with fine sand dust* and when loaded frame supported only by bolster bearing – 16632,71 *Nm*, when loaded frame supported by bolster

bearing and one of bearers – 54873,34 *Nm*, when loaded frame supported by bolster and carrying rollers – 10108,42 *Nm*;

5. Experimental researches showed that the installation of expansion rollers on bearers substantially decreases bogie rotation antitorque moment towards wagon body by factor of 4...6. It is recommended to prefer new age bogie wagons having elastic roller bearings on bolsters. Given provisions will promote a substantial reduction of supporting node, wheel pair flange and rail head side flange wear, as well as will decrease wagon derailment probability when passing curvilinear space intervals.
6. It is experimentally proved that the absence of lubricant in bogie supporting nodes considerably increases bogie rotation antitorque moment towards wagon body. Lubrication of freight wagon bogie supporting node must be done once in 1,5 years, marking lubrication date on wagon body or frame. There is currently an opportunity of freight wagon supporting node lubrication when sent to current uncoupling repair. In case of lubricant absence in supporting node bogie rotation antitorque moment towards wagon body increases 1,7…2,4 times, conducing wear of supporting nodes, wheel pair flanges, side edges of rail heads when passing curvilinear space intervals and promoting the likehood of derailment.
7. An analytic model is offered, presenting a spring chain of absolutely rigid bodies, giving an opportunity to study the oscillations of cut wagons provoked by application of brakes on gravity hump. The obtained equations of system with two degrees of freedom describe cut movement along the rail on an interval after retarder position. The designed mathematical model can be used to solve tasks of cut target braking at gravity humps. The results of theoretical analysis of cut wagon dynamics when braking at gravity humps confirm the possibility of short-term unloading of spring sets of separate wagons.
8. When detaching wagons in dry weather, at friction coefficient between retarder brake bar and the wheel equal to 0,2, the brakes must not be applied on the first bogie of wagon reaching lifted cut retarder for: cuts of 3 to 6 empty wagons and cuts of 5 to 8 loaded wagons. The optimum regime of cut braking is the application of braking on wagons in the middle of the cut.
9. Wagons with friction coefficient between brake bar and wheel rim is 0,08 to 0,15 (wagons with rusty, freshly painted wheel rims and wagons carrying molasses, when detaching in winter and rainy weather) can be retarded by applying brakes on the first wagon bogie reaching lifted retarder. In this case unloading of spring sets does not appear.