Rationalization of parameters of hydraulic dampers of a carriage

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Abstract. The possibility of the car dynamic performance improvement is studied through structural changes (modernization) of the passenger bogie. Based on the calculations of the natural oscillations of the car, the critical values were determined and rational values of the damper resistance coefficient were chosen for vertical and lateral vibrations. It was concluded that with the forced spatial fluctuations of a carriages on KCCW-CRI bogies the required damping value of the vertical oscillations should be twice as large as for the lateral ones; important factors in damping of the car oscillations are the central suspension dampers. The rational angle of installation of the hydraulic damper in the central suspension and the use of separate action in the central suspension of hydraulic dampers are justified. The proposed version of the modernization of the bogie can improve the smoothness indicators of the car by 25-30%.

1. Introduction
Bogies are the main nodes of cars in traffic safety. Therefore work is continuously carried out for creating bogies, which are more reliable and convenient for maintenance and repair [1-11]. Passenger bogies use a two-stage spring suspension which allow increasing its static deflection and flexibility, which is very important to ensure the necessary speed and smoothness. Two steps of suspension allow various elements of spring suspension to be placed in a rational combination: springs, rubber springs, hydraulic and friction dampers.

Most of the cars in the fleet are equipped with KCCW-CRI bogie. Studies conducted by JSC Railwaycar-Building Works Tver, Tver Institute of Car Building, Joint Stock Company Railway Research Institute and others, showed that the KCCW-CRI bogie with its dynamic driving characteristics meets the traffic safety criteria in straight and curved sections of the track, it has an indicator of smoothness course at the level of "satisfactory" in the range of speeds of movement up to 160 km/h. Analysis of the experimental data showed that friction dampers in the axle suspension stage worsen the smoothness performance. In addition, hydraulic dampers of the central suspension do not provide optimal damping characteristics of the vertical and lateral vibrations and the angle of location in the bogie.

Therefore, as an option to upgrade the bogie, it was proposed to determine the rational resistance parameters of the hydraulic damper [12] in the central suspension and determine its installation angle in the bogie.

When determining the rational values of damper resistance coefficients separate damping of oscillations in the central suspension in the vertical and lateral directions was considered.

The tasks related to the specification of the mathematical description were solved:
- the angular displacements of the crowning suspension under the action of transverse and vertical forces and taking into account the friction in the hinges;
- dynamic deformations of elastic pads, located under the springs, which are the inertia-free elements but with elastic and damping properties;
- operation of a stopped hydro-damper damping which performs oscillations in two planes and having a pressure relief valve of fluid pressure;
- limitations of lateral movements of the bogie bar of the bogie with rubber shock absorbers located on the frame.

When making calculations related to the modernization of a KCCW-CRI bogie, the following structural elements should be taken into account in the mathematical model of the car:
- thrust cradle suspension committing pendulum oscillations under the action of lateral and vertical forces;
- hydraulic dampers, which are installed obliquely and have safety valves for fluid pressure;
- elastic pads located under the spring springs;
- limiters of transverse movements of the bolster relative to the bogie frame.

2. Results and Discussion
2.1. Determination of critical values of damper resistance coefficients for vertical and lateral vibrations

As it is known, the critical value of the damper resistance coefficient is its minimum value at which the process of its own damped oscillations becomes aperiodic (it decays exponentially). It is extremely important to know the critical value of the damper resistance coefficient for vertical and lateral vibrations, since it is recommended to take rational values [13] within 25% of the critical values.

The critical values of damper drag coefficients were determined on the basis of calculations of the natural vertical and lateral oscillations of the car [14-21].

Differential equations describing the oscillations of all bodies of the design scheme of the car can be more compactly represented in a vector-matrix form

\[
[M] \ddot{\mathbf{U}} + [B] \dot{\mathbf{U}} + [C] \mathbf{U} + \mathbf{F} + \mathbf{Q} = 0,
\]

where \( \mathbf{U} \) is the coordinate vector of the computational scheme, defined by the components of expression (1);

- \([M]\) is an inertial matrix;
- \([B]\) is a damping matrix;
- \([C]\) is a stiffness matrix;
- \(\mathbf{F}\) is a nonlinear vector defined by accepted physical and geometric hypotheses;
- \(\mathbf{Q}\) is a cargo vector determined by external perturbations.

In determining the critical value of the damper resistance coefficient for the body’s vertical oscillations, the general system of differential equations (1) was integrated with initial conditions corresponding to the initial deviation of the body from the position of static equilibrium in the absence of external disturbances. As a result of the series of calculations, it was found that for vertical body oscillations, the critical value of the drag coefficient of the damper is \( \beta_{z} = 20 \text{ t/m} \). Figure 1 shows the graphs of damped processes for three cases when \( \beta_{z} < \beta_{z, cr} \), \( \beta_{z} = \beta_{z, cr} \) and \( \beta_{z} > \beta_{z, cr} \).

![Figure 1. Vertical body travel (m).](image-url)
In the same way the critical value of the damper resistance coefficient for lateral oscillations was determined. When integrating the system of differential equations (1), the initial lateral deviation of the body from the equilibrium position was set and the value $\beta_y$ was selected by calculation so that the process of its own damped oscillations became aperiodic. Fragments of these calculations are shown in figure 2 graphs. As a result of the calculations performed, it was found that for lateral body vibrations, the critical value of the damper resistance coefficient is $\beta_{y,cr} = 10 \, t \cdot s/m$.

From the above calculations it follows that with the forced spatial oscillations of a carriage on KCCW-CRI-type bogies the necessary amount of vertical oscillation damping should be twice as large as for the lateral ones.

This circumstance must be taken into account when determining the rational angle of installation of the damper in the trolley for simultaneous damping of the vertical and lateral vibrations, as well as when choosing the resistance coefficients of the vertical and lateral dampers with separate damping of the oscillations.

2.2. Selection of damping drag coefficients with separate vibration damping in the central suspension and the absence of friction dampers in the axle box gear

It has been suggested that side-oscillation damping can be carried out by friction in the hinges of the cradle suspension without setting up an additional damper.

According to this variant, the series of calculations were carried out which showed that the absence of side oscillation damper in the central suspension leads to an unacceptable level of lateral and frame forces, as well as lateral accelerations on the frame and body.

The friction in the hinges of the cradle suspension is insufficient for damping lateral vibrations. In order to establish the necessary amount of damping of lateral oscillations calculations were carried out at $\beta_z = 5 \, t \cdot s/m$ and varying the value of $\beta_y$.

From the results of calculations it follows that such indicators as lateral forces in the central suspension, frame forces, lateral accelerations of the body and lateral accelerations of the bogie frame have satisfactory values for $\beta_y$ not less than 2.5 $t \cdot s/m$. In other words when extinguishing separately the damping resistances of the dampers in the vertical and lateral directions should be within 0.25 of their critical values. Moreover, the overestimation of the resistance coefficient of the vertical absorber over 0.25 $\beta_{z,cr}$ does not lead to a noticeable change in the dynamic performance of the car in a vertical plane. At the same time, an increase in the drag coefficient of a horizontal absorber above 0.25 $\beta_{y,cr}$ significantly impairs the side dynamics: lateral and frame forces and lateral accelerations.

Along with this, a series of calculations was carried out to determine the significance of the central and axle suspension dampers for the ride quality of the car. It turned out that the decisive factor in damping of the car oscillations are the central suspension dampers. When they are cut off, axle suspension dampers cannot provide a safe course even when placing additional hydraulic dampers in the axle box.

With the right choice of damper parameters in the central suspension, the placement of dampers in the axle box does not lead to a significant improvement in the dynamic performance of the car.
2.3. Determination of rational installation angles of hydraulic dampers for joint damping of vertical and lateral vibrations

Since preliminary calculations of the dynamic indicators of the car on bogies with separate damping of oscillations showed that the rational values of damping resistance coefficients in the vertical and lateral directions are different, it is very important to choose the angle of the damper in the bogie in which the inclined damper damps oscillations in two planes. In this regard, in the next series of calculations, the dynamic indicators of the car were analyzed at damper installation angles of 45°, 60°, 75° to the horizontal plane and various coefficients of fluid resistance. The maximum values were analyzed as dynamic indicators of the car:

- total vertical forces in the central suspension acting on the bolster;
- vertical forces acting on the spring set of the axle box springs;
- lateral forces acting on the spring sets of the central suspension;
- lateral forces acting between the wheel and the rail;
- frame forces acting on wheelsets;
- vertical acceleration of the body and frames of carriages;
- lateral accelerations of the body and frames of carriages.

The graphs of these dynamic indicators at the angle of installation of absorbers 45° are shown in figures 3 and 4.

![Figure 3](image1)

**Figure 3.** Vertical reaction of the central suspension of the bogie (a) and vertical reaction of the spring set of the axle box suspension (b)

1 – $\beta_h = 5.0$; 2 – $\beta_h = 6.0$; 3 – $\beta_h = 7.0$.

![Figure 4](image2)

**Figure 4.** Side reaction in the center suspension of the trolley (a) and the frame force on the wheel pair (t):

1 – $\beta_h = 5.0$; 2 – $\beta_h = 6.0$; 3 – $\beta_h = 7.0$.

Similar dependences were obtained at installation angles of 60° and 75°. Comparison of the calculation results showed that the 45° installation angle of the damper is the most unfavorable, since at the same time, approximately the same damping forces are received both in the vertical and in the lateral directions. Preliminary calculations of the values of critical damping with vertical and lateral oscillations showed that there are two times different and therefore there is not enough damping force for vertical oscillations at a 45° angle, and for lateral oscillations, it turns out to be excessive.

Analysis of the calculated data showed that the most rational is the angle of installation of the damper 60-75°.
An increase in the drag coefficient of absorbers above 0.25 $\beta_{cr}$ does not significantly affect the vertical forces however it noticeably worsens the forces and accelerations in the lateral direction.

Therefore, with an increase in the damping resistance coefficient, it is necessary to increase the angle of the damper installation to the horizontal plane to about 60° - 75°.

The calculations have also shown that the best dynamic indicators occur when the damping resistance coefficient is 6 - 8 t·s/m and the installation angle is 60° - 75°. So accelerations on the body are 0.06 - 0.08g, and on the frame do not exceed 0.5 - 0.6g in the range of speeds of movement from 15 to 50 m/s (54 - 180 km/h). With such damper resistance coefficients and its installation angles the required ratio of damping forces in the vertical and transverse directions is ensured, and the values of the drag coefficients are close to 0.25 of $\beta_{cr}$.

The standard damper of the KCCW-CRI bogie has a drag coefficient determined from a bench chart of the order of $\beta = 12$ t·s/m. The cut-off diagram of the safety valve is about 1.5 tons. For the indicated damper parameters calculations were made for three installation angles of 60°, 75° and 80°. From the results of the calculations, it follows that for a standard hydraulic damper the rational value of the installation is 65° - 75° since at the same time all the considered dynamic indicators of the car take on minimal values.

2.4. Indicators of the smoothness of the car for typical and upgraded versions of trucks

Ultimately, the effectiveness of measures to modernize the central stage of spring suspension is determined by the indicators of smoothness in the vertical and lateral directions.

In this paper the indicators of smoothness were determined by calculation when integrating the system which describes the dynamic state of the car. The indicator of smoothness of motion according to [21] was determined by the formula

$$ W = 2.7 \cdot K \cdot \sqrt[3]{z_0^2 \cdot \omega^5} \quad (2) $$

where $z_0$ is the amplitude of oscillation;
$\omega$ is the oscillation frequency;
$K$ is an empirical coefficient depending on the frequency and the plane in which oscillations occur.

In the calculations the values of the coefficient $K$ were determined according to the schedule [21] showing the change in its values from the frequency and direction of oscillations.

The results of calculations of smoothness in the vertical $W_z$ and horizontal $W_y$ directions are presented in table 1 and figure 5.

| $V$, м/c | $\omega$, с⁻¹ | $z_0$, М | $y_0$, М | $W_z$ | $W_y$ | $k_z$ | $k_y$ |
|----------|----------------|----------|----------|-------|-------|-------|-------|
| 10       | 2.512          | 0.0117   | 0.0161   | 1.106 | 1.328 | 1.10  | 1.20  |
| 15       | 3.678          | 0.0136   | 0.0142   | 1.550 | 1.830 | 1.20  | 1.40  |
| 20       | 5.024          | 0.0171   | 0.0095   | 2.155 | 2.090 | 1.21  | 1.40  |
| 25       | 6.280          | 0.0227   | 0.0081   | 2.590 | 2.260 | 1.22  | 1.45  |
| 30       | 7.536          | 0.0590   | 0.0137   | 3.698 | 2.662 | 1.21  | 1.35  |
| 35       | 8.792          | 0.0257   | 0.0102   | 3.055 | 2.500 | 1.20  | 1.30  |
| 40       | 10.048         | 0.0175   | 0.0099   | 3.020 | 2.560 | 1.19  | 1.20  |
| 45       | 11.300         | 0.0081   | 0.0069   | 2.430 | 2.337 | 1.15  | 1.18  |
| 50       | 12.560         | 0.0062   | 0.0071   | 2.340 | 2.500 | 1.10  | 1.15  |

Variants were calculated for the following path condition: the lengths of the vertical and horizontal irregularities were 25 m; amplitudes of vertical and horizontal irregularities of 0.02 m; track width 1520 mm.

A typical bogie variant (Table 1) had a hydraulic damper angle of 45° and friction dampers with a friction force of 0.07 tons. A modernized version of the trolley with separate damping of vertical and lateral oscillations in the central suspension (Tables 2, 3) was investigated.

The data of the Table 1 shows that the maximum values of the smoothness indicators in the vertical direction are 3.698 and in the horizontal 2.662 at a speed of 30 m/s (108 km/h) which does not fit the norms of satisfactory performance equal to 3.25.

The upgraded version (Tab. 2, 3) with separate damping of oscillations in the central stage has the best smoothness. This option has indicators of smoothness in the vertical and lateral directions respectively
2.847 and 2.088 at speeds of 30 and 45 m/s. In other words when using the upgraded version of the bogie, it is possible to reduce the maximum values of the smoothness indicators by 25-30% compared to the standard version.

![Figure 5. Vertical ride index:
1 - standard option; 2 - separate cancellation.](image)

### Table 2. Separate cancellation (I option, $\beta_z = 5 \cdot t \cdot s/m$ and $\beta_y = 2.5 \cdot t \cdot s/m$).

| $V$, m/c | $\omega$, c$^{-1}$ | $z_0$, M | $y_0$, M | $W_z$ | $W_y$ | $k_z$ | $k_y$ |
|----------|-------------------|-----------|-----------|-------|-------|-------|-------|
| 10       | 2.512             | 0.0117    | 0.0333    | 1.105 | 1.651 | 1.10  | 1.20  |
| 15       | 3.678             | 0.0139    | 0.0111    | 1.556 | 1.696 | 1.20  | 1.40  |
| 20       | 5.024             | 0.0179    | 0.0055    | 2.185 | 1.774 | 1.21  | 1.40  |
| 25       | 6.280             | 0.0298    | 0.0057    | 2.812 | 2.035 | 1.22  | 1.45  |
| 30       | 7.536             | 0.0318    | 0.0061    | 3.072 | 2.088 | 1.21  | 1.35  |
| 35       | 8.792             | 0.0219    | 0.0055    | 2.912 | 2.084 | 1.20  | 1.30  |
| 40       | 10.048            | 0.0139    | 0.0051    | 2.817 | 2.102 | 1.19  | 1.20  |
| 45       | 11.300            | 0.0094    | 0.0049    | 2.539 | 2.142 | 1.15  | 1.18  |
| 50       | 12.560            | 0.0063    | 0.0053    | 2.250 | 2.230 | 1.10  | 1.15  |

### Table 3. Separate blanking (II variant, $\beta_z = 7 \cdot t \cdot s/m$ and $\beta_y = 2.5 \cdot t \cdot s/m$).

| $V$, m/c | $\omega$, c$^{-1}$ | $z_0$, M | $y_0$, M | $W_z$ | $W_y$ | $k_z$ | $k_y$ |
|----------|-------------------|-----------|-----------|-------|-------|-------|-------|
| 10       | 2.512             | 0.0116    | 0.0302    | 1.103 | 1.603 | 1.10  | 1.20  |
| 15       | 3.678             | 0.0136    | 0.0118    | 1.545 | 1.728 | 1.20  | 1.40  |
| 20       | 5.024             | 0.0149    | 0.0062    | 2.247 | 1.839 | 1.21  | 1.40  |
| 25       | 6.280             | 0.0220    | 0.0054    | 2.567 | 2.000 | 1.22  | 1.45  |
| 30       | 7.536             | 0.0247    | 0.0055    | 2.847 | 2.024 | 1.21  | 1.35  |
| 35       | 8.792             | 0.0201    | 0.0049    | 2.838 | 2.013 | 1.20  | 1.30  |
| 40       | 10.048            | 0.0146    | 0.0049    | 2.859 | 2.078 | 1.19  | 1.20  |
| 45       | 11.300            | 0.0095    | 0.0045    | 2.547 | 2.088 | 1.15  | 1.18  |
| 50       | 12.560            | 0.0054    | 0.0049    | 2.148 | 2.181 | 1.10  | 1.15  |

### 3. Conclusions

1. The calculations have established that the critical value of the drag coefficient of the hydraulic damper in the central suspension of the trolley is $20 \cdot t \cdot s/m$ with vertical oscillations and $10 \cdot t \cdot s/m$ with lateral ones.
2. The rational resistance coefficients for separate oscillations are 25% of their critical values.
3. When inclined installation of hydraulic dampers in the bogie, the angle of inclination of the damper to the horizontal plane should be 60-70 degrees for joint damping of vertical and lateral oscillations.
4. In case of separate damping of vertical and lateral oscillations in the central stage rational values of the resistance coefficients of hydraulic dampers should be in the vertical direction 5-7 $\cdot t \cdot s/m$, in the lateral direction 2.5 $\cdot t \cdot s/m$. 

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5. The proposed version of the modernization of the bogie made it possible to obtain smoothness indicators by 25-30% less than the allowable equal to 3.25 and determining the level of «satisfactory» progress.

References
[1] Pradhan S, Menezes A, Samantaray A K and Bhattacharyya R 2014 Performance evaluation of steering bogies on various tracks Proc. Int. Conf. on Theoretical, Applied, Computational and Experimental Mechanics (IIT Kharagpur, India) ICTACEM-2014/221
[2] Voron O A, Bulavin Ju P and Volkov I V 2018 K voprosu vybora hodovyh chastej dlja perspektivnogo izotermicheskogo podvzhnogo sostava [On the choice of running gears for prospective isothermal rolling stock] Vestnik RGUPS iss 4 pp 63–70
[3] Zhang T, True H and Dai H 2018 The influence of the perturbation of the wheel rotation speed on the stability of a railway bogie on steady curve sections of a track Vehicle Syst Dyn doi: 10.1080/00423114.2018.1469778
[4] Levin A V, Pavljukov A Je and Smol'janinov A V 2014 Ocena dempfilrujushhih svojstv buksovogo podveshivaniya mnogoosnych telezhek gruzovyh vagonov [Evaluation of the damping properties of the axle suspension of multi-axle freight car bogies] Transport Urala [Transport of the Urals] iss 2(41) pp 27-32
[5] Ushkalov V F, Mokrij T F and Mashhenko I A 1996 Vlijanie parametrov hodovyh chastej passazhirskih vagonov na processy vzaimodejstvija i iznos koles i rel'sov [Influence of parameters of chassis parts of carriages on interaction processes and wear of wheels and rails] Theses of the reports of the IX Int. Conf. “Problems of railway mechanics” (Dnepropetrovsk) p 134.
[6] Hohlov A A 1981 Parametry perspektivnych dvuhosnych telezhek vagonov [Parameters of promising two-axle wagon bogies] Trudy VNIIZhT [Proceedings of the Railway Research Institute] iss 4 pp 51-60
[7] Romen Ju S 2015 Faktory, obuslovilivajushhie processy vzai-modejstvija v sisteme koleso-rel's pri dvizhenii poezda v krivyh [Factors contributing to the interaction processes in the wheel-rail system when the train moves in curves] Vestnik VNIIZhT [Vestnik of the Railway Research Institute] no 1 pp 17-26
[8] Pevzner V O, Romen Ju S, Orlova A M and Zavertaljuk A V 2010 Jeksperimental'nye issledovanija po ocenke vlijania shiriny kolei i sostojanija hodovyh chastej na uroven' bokovyh sil [Experimental studies to assess the influence of gauge width and condition of the running gears on the side curveslevel] Vestnik VNIIZhT [Vestnik of the Railway Research Institute] no 2 pp 39-41
[9] Vuong T T, Ermoshenko Yu V 2018 Opredelenie dinamicheskih reakcij v sobodeniyah elementov podvesok: novye podhody [Determination of dynamic responses in connections of suspension elements: new approaches] Sovremennye tehnologii. Sistemnyj analiz. Modelirovanie [Systems. Methods. Technologies] no 2(58) pp 118-125 doi: 10.26731/1813-9108.2018.2(58).118-125
[10] Kargapolcev S K, Kuptsov Y A, Novoselstev P V, Gozbenko V E 2018 Sposob opredeleniya koeficienta scepeleniya kolesnoj pary s rel'sami pri tormozhenii [A method for determining the coefficient of wheel pair adhesion with rails at braking] Sovremennye tehnologii. Sistemnyj analiz. Modelirovanie [Systems. Methods. Technologies] no 2(58) pp 112-117 doi: 10.26731/1813-9108.2018.2(58).112-117
[11] Akhmadeeva A A, Gozbenko V E and Kargapolcev S K 2014 Vertikal'naja dinamika vagona s uchetom nerovnostej kolei [Vertical dynamics of the carriage], Sovremennye tehnologii. Sistemnyj analiz. Modelirovanie [Systems. Methods. Technologies] no 3(23) pp 57-62.
[12] Burchak G P., Savos'kin A N., Fradkin G N and Kossov V S 1997 Modelirovanie vozmuşheniya v vide gorizontal'noj nerovnosti osi puti dlja issledovaniya izvistogo dvizhenia rel'sovogo jekipazha [Simulation of disturbances in the form of horizontal irregularities of the axis of the path for the study of the winding movement of the rail carriage] (Moscow: Proc. MSTU ) iss 912 pp 23-29
[13] State standard 34093-2017 Carriage of locomotive thrust. Requirements for strength and dynamic qualities
[14] Petrov G I and Tarmaev A A 2018 Modeling of railway vehicles movement having deviations in the content of running parts Proc. Int. Conf. “Aviamechanical Engineering and Transport” (AVENT 2018) (Atlantis Press, Series: Advances in Engineering Research) vol 158 pp 410-415 doi: 10.2991/aevt-18.2018.79
[15] Vershinskij S.V., Danilov V.N., Husidov V D 1991 Dinamika vagona [Dynamics of the car] ed S.V. Vershinskij (Moscow: Transport)

[16] Ushkalov V F , Reznikov L M, Ikkol V S et al 1989 Matematicheskoe modelirovanie kolebanij rel'sovyh transportnyh sredstv [Mathematical modeling of oscillations of rail vehicles] ed V F Ushkalov (Kiev: Naukova dumka)

[17] Hohlov A A 1981 Optimal'nye zakony upravlenija dinamicheskimi processami vagonov [Optimal laws for controlling dynamic processes of cars] Trudy MIIT [Proceedings MIET] iss 679 pp 42-60

[18] Hohlov A A 1982 Postroenie edinoj matematicheskoj modeli kolebanij mnogoosnyh jekipazhej [Building a unified mathematical model of oscillations of multi-axle crews] Vestnik VNIIZhT [Vestnik of the Railway Research Institute] no 3 pp 23-25

[19] Husidov V D, Zaslavskij L V, Chan F T, Husidov V V 1995 Tsifrovoe modelirovanie kolebaniy passazhirskogo vagona pri dvizhenii po pryamym i krivolineynym uchastkam puti [Digital simulation of carriage oscillations while driving along straight and curved sections of the track] Vestnik VNIIZhT [Vestnik of the Railway Research Institute] no 3 pp 18-25

[20] Husidov V V 1995 Komp'juternoe modelirovanie dvizhenija passazhirskikh vagonov po pryamym i krivolineynym uchastkam [Computer simulation of the movement of carriages in straight and curved sections] Theses of the Reports of the Republican Sci. and Methodical Conf. "The use of computers in the educational process and scientific research" (Gomel) pp 81-82.

[21] Chelnokov I I 1975 Gidravlicheskie gasiteli kolebanij passazhirskih vagonov [Hydraulic dampers for carriages] (Moscow: Transport)