EFFECT OF THE STATE OF CAR RUNNING GEARS AND RAILWAY TRACK ON WHEEL AND RAIL WEAR

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ABSTRACT

This paper presents the results of theoretical research and real condition experiments which were conducted with the purpose of determining the influence of the open goods wagon running gear characteristic parameter deviations from their nominal values on the wheel and rail wear. The experimental results has confirmed the reliability of conclusions made on the basis of modelling.

Keywords: wear, open goods wagon, wheel, wheel flange.

1. INTRODUCTION

The problems of wheel and rail wear were important for the railway transport at every stage of its existence. First of all, it is connected with the fact that the area of wheel and rail contact endures rather intensive force actions as well as elastic and non-elastic relative motions. Depending on the wheel lateral displacements in relation to a rail and on the acting forces, the contact between a non-worn wheel and a rail can take place at one point on the wheel rolling surface (one-point contact), at two points which are on the wheel rolling surface and the wheel flange (two-point contact) or, for the worn wheels and rails, it can be in a number of points (multi-point contact). It is quite natural for wheels to have the wear on treads which transmits considerable vertical forces and has considerable slidings with respect to rails in curves and especially in the case of braking and traction power realization. But since the middle of the 1980s on the USSR railways the locomotive and wagon wheel flange wear, as well as rail head side surface wear have sharply increased and it became one of the topical problems on railway transport. It is interesting to note that the flange wear increase is accompanied by the tread wear decrease. As an example we can present the data about the repair of passenger car wheelsets in one of the car depots in Ukraine for the period from 1989 to 1995 (Fig. 1).

The plots in Fig.1 confirm rather exactly the above mentioned tendencies in wear changes in recent years.

2. A MATHEMATICAL MODEL

For the study of the effect of freight car running gear state on wheel and rail wear a mathematical model of the forced vibrations of the system «wagon-track» is used in which the inertial and elastic-dissipative track properties are taken into account in vertical and horizontal planes according to the hypothesis of V.Z.Vlasov, and the real irregularities [1-4] are also taken into account. The motion of an open goods wagon with standard bogies of model 18-100 is considered henceforth.

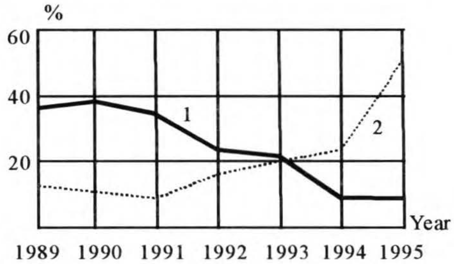

Fig. 1. The change (in %) of deviations in a number of wheelsets which came into repair because of the thread wear (line 1) and of the flange wear (line 2) with respect to the total number of wheelsets which came into repair for the period 1989-1995.

The computational scheme of the system includes eleven solid bodies (a car body, two bolsters four side-frames, four wheelsets) with elastic dissipative connections between them. It is adopted for generality that the wagon has a double springing (the spasers are set between axle-boxes and bogie side-frames).

The following relative motions are assumed to take place:

- yaw and rolling of bolsters relative to a car body;
- all linear and angular in a horizontal plane displacements (yaw) between bolsters and side-frames, and also between side-frames and wheelsets;
- vertical and lateral displacements of wheels relative to the track.

On wheel and rail contact the pseudosliding forces appear which were determined according to the Carter's theory taking into account for physical and geometrical non-linearities.

Henceforth x, y, z denote the body centre of mass displacements along the corresponding axes, and Θ, φ, ψ denote the body rotation angles relative to principal central axes of inertia; the analogous bolster displacements are marked by index ? (? = 1, 2 - a number of a bogie), side-frame displacements - by index *fij* (j = 1 - the left side, j = 2 - the right side of a wagon), wheel set displacements - by index *kim* (m = 1, 2 - a number of a wheel set in the bogie), rail displacements at the points of contact with wheels - by index *pimj* (the displacements of rail lines are assumed only in two directions - along the axes y and z); wheel displacements are marked by index *imj*.

Besides, it is adopted that there is no side-frame rolling, and the angles of wheel set-rotation with respect to a y-axis are determined without taking into account the wheel crippage.

It is also adopted that all the wheels move without separation from rails, so vertical displacements of the rails are equal to

$$z_{pimj} = z_{imj} + \Delta r_{imj} - \eta_{\text{в}imj}; \tag{1}$$

$$y_{imj} = y_{kim} - r_{im}\Theta_{kim} - y_{pimj} - \eta_{\text{г}imj}. \tag{2}$$

Here y_{imj} are horizontal wheel displacements relative to rails; $\Delta r_{imj} = f(y_{imj})$ are wheel rolling circle radius changes; η_{vimj}, η_{himj} are current ordinates of vertical and horizontal irregularities; $2l$ is a car base; h is the height of a car body centre of mass above the plane of bolster resting on elastic elements; r_{im} is the im-th wheel rolling circle radius; $2b_2$ is the distance between wheel rolling circles.

Twenty four coupling equations have been derived according to the given assumptions. The system has 58 degrees of freedom. Linear and angular body displacements are taken as a system of generalized coordinates.

Relative body displacements are expressed through the generalized coordinates in the following way:

- body - bolsters:

$$\begin{aligned} &\text{during yaw } \Delta\psi_i = \psi - \psi_i \ (i = 1, 2); \\ &\text{during rolling } \Delta\Theta_i = \Theta - \Theta_i \ (i = 1, 2). \end{aligned} \tag{3}$$

- bolsters - side-frames in longitudinal, lateral, vertical directions and yaw respectively ($2b$ is the distance in lateral direction between the axles of spring assemblies):

$$\begin{aligned} \Delta_{cxij} &= x - (-1)^j b\psi_i - x_{fij} \\ \Delta_{cyij} &= y - (-1)^i l\psi - h\Theta - y_{fij}, \\ \Delta_{czij} &= z + (-1)^i l\varphi + (-1)^j b\Theta_i - z_{fij} \\ \Delta_{c\psi ij} &= \psi_i - \psi_{fij} \ (i, j = 1, 2) \end{aligned} \tag{4}$$

- side-frames - wheel sets in longitudinal, lateral, vertical directions and yaw respectively ($2l_1$ is a bogie base, $2b_1$ - is the distance in lateral direction between the axles of axleboxes):

$$\begin{aligned} \Delta_{fximj} &= x_{fij} - x_{kim} + (-1)^j b_1\psi_{kim}, \\ \Delta_{fyimj} &= y_{fij} - (-1)^m l_1\psi_{fij} - y_{kim}, \\ \Delta_{fzimj} &= z_{fij} + (-1)^m l_1\varphi_{fij} - z_{kim} - (-1)^j b_1\Theta_{kim}, \\ \Delta_{f\psi imj} &= \psi_{fij} - \psi_{kim}, (i, m, j = 1,2). \end{aligned} \tag{5}$$

- wheels - rails in longitudinal and lateral directions

$$\begin{aligned} x_{imj} &= x_{kim} - (-1)^j b_2\psi_{kim}, \\ y_{imj} &= y_{kim} - r_{im}\Theta_{kim} - y_{pimj} - \eta_{himj}, (i, m, j = 1,2). \end{aligned} \tag{6}$$

Wheels crippage on rails in longitudinal (ε_x) and horizontal (ε_y) directions takes place in the track plane which are expressed respectively as

$$\varepsilon_{ximj} = -\left[(-1)^j \frac{b_2\dot{\psi}_{kim}}{V} + \frac{\Delta r_{imj}}{r_{im}}\right], \quad \varepsilon_{yimj} = \frac{1}{V}\left[\dot{y}_{kim} - \dot{y}_{pimj} - r_{im}\dot{\Theta}_{kim}\right] - \psi_{kim}. \tag{7}$$

The tangent frictional forces T_x and T_y were determined from these wheel crippage

$$T_{ximj} = -F_{ximj}\,\varepsilon_{ximj}, \qquad T_{yimj} = -F_{yimj}\,\varepsilon_{yimj}\,. \tag{8}$$

Here F_{ximj}, F_{yimj} are the coefficients of pseudosliding in the directions along (along the x axis) and across (along the y axis) the axis of the track which are determined by the formula

$$F_{ximj} = F_{yimj} = \frac{f_{imj}}{\sqrt{\left(1 + h_{imj}^2\,\varepsilon_{imj}^2\right)}};\ h_{imj} = \frac{f_{imj}}{P_{imj}\,f_T}, \tag{9}$$

where the coefficient f_{imj} depending on the total wheel pressure on rail P_{imj} is determined as in [3, 4] in such a way:

$$f_{imj} = 235\,P_{imj} - 2{,}4\,P_{imj}^2 + 0{,}01\,P_{imj}^3\,. \tag{10}$$

The forces S acting between the wagon bodies having similar indices were determined from the relative motions (3) - (5).

Differential equations of space vibrations of a «wagon - track» system are presented by a system (11):

$$\begin{aligned}
& m\ddot{x} + m_1(\ddot{x}_1 + \ddot{x}_2) + \sum_{i=1}^{2}\sum_{j=1}^{2} S_{cxij} = 0, \\
& m\ddot{y} + m_1(\ddot{y}_1 + \ddot{y}_2) + \sum_{i=1}^{2}\sum_{j=1}^{2} S_{cyij} = 0, \\
& m\ddot{z} + m_1(\ddot{z}_1 + \ddot{z}_2) + \sum_{i=1}^{2}\sum_{j=1}^{2} S_{czij} - (m + 2m_1)g = 0, \\
& I_x\theta + \sum_{i=1}^{2} M_i - 2M_\theta - m_1(\ddot{y}_1 + \ddot{y}_2)h + b_3\sum_{i=1}^{2}\sum_{j=1}^{2}(-1)^j S_{cij} - h\sum_{i=1}^{2}\sum_{j=1}^{2}(-1)^j S_{cyij} = 0, \\
& I_y\ddot{\varphi} + I_{y1}(\ddot{\varphi}_1 + \ddot{\varphi}_2) + m_1(\ddot{x}_1 + \ddot{x}_2)h + m_1(\ddot{z}_1 - \ddot{z}_2)l + h\sum_{i=1}^{2}\sum_{j=1}^{2} S_{cxij} + l\sum_{i=1}^{2}\sum_{j=1}^{2}(-1)^i S_{czij} = 0, \\
& I_z\ddot{\psi} + m_1(\ddot{y}_1 - \ddot{y}_2)l - l\sum_{i=1}^{2}\sum_{j=1}^{2}(-1)^i S_{cyij} + \sum_{i=1}^{2} S_{\psi i} + b_3 f_T\sum_{i=1}^{2}\sum_{j=1}^{2}(-1)^i S_{cij}\, sign\dot{\Delta}_{\psi i} = 0, \\
& I_{x1}\ddot{\theta}_i - M_i - b_3\sum_{j=1}^{2}(-1)^j S_{cij} + b\sum_{j=1}^{2}(-1)^j S_{czij} = 0, \\
& I_{z1}\ddot{\psi}_i + \sum_{j=1}^{2} S_{c\psi ij} - b\sum_{j=1}^{2}(-1)^j S_{cxij} - S_{\psi i} - b_3 f_T\sum_{j=1}^{2} S_{cij}\, sign\dot{\Delta}_{\psi i} = 0, \\
& m_f\ddot{x}_{fij} - S_{cxij} + \sum_{m=1}^{2} S_{fximj} = 0, \\
& m_f\ddot{y}_{fij} - S_{cyij} + \sum_{m=1}^{2} S_{cyimj} = 0, \\
& m_f\ddot{z}_{fij} - S_{czij} + \sum_{m=1}^{2} S_{fzimj} - m_f g = 0, \\
& I_{fy}\ddot{\varphi}_{fij} + l_1\sum_{i=1}^{2}(-1)^m S_{fzimj} = 0,
\end{aligned} \tag{11}$$

$$I_{\mathrm{fz}}\ddot{\psi}_{\mathrm{f}ij} - S_{\mathrm{c}\psi ij} - l_1\sum_{m=1}^{2} S_{\mathrm{f}yijm} + \sum_{m=1}^{2} S_{\mathrm{c}\psi imj} = 0\,,$$

$$m_{\mathrm{k}}\ddot{x}_{\mathrm{k}im} + \frac{I_0\ddot{x}_{kim}^2}{r^2} - \sum_{j=1}^{2}\left(T_{ximj} + S_{\mathrm{f}ximj}\right) = 0\,,$$

$$m_{\mathrm{k}}\ddot{y}_{\mathrm{k}ij} - \sum_{j=1}^{2}\left(T_{yimj} + S_{\mathrm{f}yimj}\right) + \sum_{j=1}^{2} f'\left(y_{imj}\right)S_{\mathrm{f}zimj} = 0\,,$$

$$m_{\mathrm{k}}\ddot{z}_{\mathrm{k}im} - \sum_{j=1}^{2} S_{\mathrm{f}zimj} + \sum_{j=1}^{2} S_{\mathrm{f}zimj} - m_{\mathrm{k}}g = 0\,,$$

$$I_{xk}\ddot{\theta}_{kim} - b_1\sum_{j=1}^{2}(-1)^j S_{\mathrm{f}zimj} + \sum_{j=1}^{2}\left[(-1)^j b_2 - rf'\left(y_{imj}\right)\right]S_{\mathrm{B}zimj} + r\sum_{j=1}^{2} T_{yimj} = 0\,,$$

$$I_{zk}\ddot{\psi}_{kim} + b_1\sum_{j=1}^{2}(-1)^j S_{\mathrm{f}ximj} - \sum_{j=1}^{2} S_{\mathrm{B}\psi\, imj} + b_2\sum_{j=1}^{2}(-1)^j T_{ximj} = 0\,,$$

$$m_{\mathrm{p}}\ddot{y}_{\mathrm{p}imj} + y_{imj} + S_{\mathrm{f}yimj} - f'\left(y_{imj}\right)S_{\mathrm{B}zimj} = 0\,, \quad (i,\, m,\, j = 1,\, 2).$$

In the equations (11) the interaction forces are equal to:

$$S_{\mathrm{B}z\,imj} = m_z\ddot{z}_{pimj} + \chi k_z\dot{z}_{pimj} + k_z z_{pimj}\,,$$
$$S_{By\,imj} = m_y\ddot{y}_{pimj} + \chi k_y\dot{y}_{pimj} + k_z y_{pimj}\,,$$

where m_z, m_y, k_z, k_y are the given track parameters; χ is a coefficient characterizing the dissipation in the base.

The equations (11) describe the wagor motion along a straight track section. In the case of curved section it is necessary to use the coordinates in a stationary system

$$q^a = q + q^e, \tag{12}$$

where q, q^e are coordinates in relative and transferable motion, the latter (and their derivatives) are determined by a curve equation. In such a case the additions in expressions for relative horizontal lateral displacement appear

$$\begin{aligned} \Delta_{\mathrm{f}yimj} &= y_{\mathrm{f}ij} - (-1)^m l_1\psi_{\mathrm{f}ij} - y_{kim} - \frac{k_0}{2}l_1^2\,, \\ \Delta_{\mathrm{c}yij} &= y - (-1)^j l\psi - h\Theta - y_{\mathrm{f}ij} - \frac{k_0}{2}l^2\,. \end{aligned} \tag{13}$$

Here k_0 is the curvature of a curve.

Expressions for the longitudinal wheel crippages on rails (7) are also to be changed

$$\varepsilon_{ximj} = -\left[(-1)^j\frac{b_2\dot{\psi}_{kim}}{V} + \frac{\Delta r_{imj}}{r_{imj}}(-1)^j b_2 k_0\right]. \tag{14}$$

Terms with the second time derivatives also appear in equations for the lateral play of all bodies, they are centrifugal forces on curves.

Since the biggest side wear appears with a two-point contact [5], so in the paper the description of the second phase of the contact after the clearance in the gauge is taken up, i.e. after the fulfilment of conditions

$$\Delta y^{h}_{kpimj} = (-1)^{j}\left(y_{kim} - y_{pimj} - (r_{im} + \Delta r_{fl})\Theta_{kim} - \eta_{kimj} - (-1)^{j}\delta\right) \geq 0 , \tag{15}$$

where Δr_{fl} is the increment of wheel radius from a middle rolling circle to a flange; δ is half a railway gauge.

It was assumed that by Carter's theory the forces of pseudosliding also appear on the side surface of flange. They are summed up with the corresponding forces on the tread. The vertical wheel pressure on rail is redistributed from tread (index «k») on a flange (index «fl») with increasing the wheel pressure on rail (the contact in a horizontal plane is modelled by a linear spring). The tread wear is usually determined [6] according to the following formula

$$W_{kimj} = \sqrt{\left(T_{ximj}\,\varepsilon_{ximj}\right)^{2} + \left(T_{yimj}\,\varepsilon_{yimj}\right)^{2}} , \tag{16}$$

that is essentially the linear work of friction forces.

By analogy the wear of flange surface is determined in the paper as

$$W_{fl\,imj} = \sqrt{\left(T^{fl}_{ximj}\,\varepsilon^{fl}_{ximj}\right)^{2} + \left(\frac{T^{fl}_{yimj}\,\varepsilon^{fl}_{yimj}}{\cos^{2}\alpha_{fl}}\right)^{2}} , \tag{17}$$

where α_{fl} is the angle between side surface of the wheel flange and horizontal plane.

Mr. V.A.Litwin has developed the software for computer modelling in accordance with this model.

3. THE RESULTS OF NUMERICAL EXPERIMENT

By means of modelling it was studied the influence of deviations from nominal sizes of the following parameters of an open goods wagon running gears on the coefficient of wheel flange wear:

- the difference in wheel diameters for the same wheelset, the wheelsets of the same bogie and the wheelsets of different bogies;
- longitudinal and lateral clearances between axle-boxes and slides of sideframes;
- wheelset misalignment in the bogie;
- lateral wheelset displacement relative to longitudinal axis;
- lateral bogie displacement relative to a car body.

The calculations have demonstrated the following.

The wheelset flange wear is considerably influenced by the lateral displacement of wheelset $\Delta_?$ relative to longitudinal bogie axis. So, with the value $\Delta_?$ equal to 4 mm while running in a curve with radius 600 m at speed 20 m/sec the coefficient of flange wear increases almost 1.5 times.

The wheelset misalignment connected with the difference in side-frame bases, depending on the position of curve's curvature centre relative to the direction of vehicle running, can either increase or decrease the value of the angle of attack of the first wheelset on an external rail in a curve. The misalignment's influence on the wear is most essential in the case when it favours the increasing of the angle of attack. With the turn in the plan of each wheelset on the angle $5 \cdot 10^{-4}$ rad that corresponds to the difference of side-frame bases 2 mm with car running in a curve with radius 600 m at speed 20 m/s, the coefficient of wear increases by 15 %.

The difference in wheel diameters of the same wheelset results in increasing of wheel flanges wear. So, with car running in a curve with radius 600 m at speed 20 m/s the increasing of coefficient of wear by 20 % and 30 % corresponds to the wheel diameter difference of 1 and 2 mm respectively.

Lateral bogie displacement relative to a body, caused by the lateral displacement of the pivot unit has a relatively little effect. E.g., in a case of lateral displacement of the pivot relative to a longitudinal body axis by 8.5 mm while car running in a curve with a radius 600 m at speed 20 m/sec the coefficient of wear increases by 10 %.

The decrease of longitudinal and lateral clearances in axle-boxes has practically no influence on the value of flange wear coefficient until the clearances become small enough (longitudinal clearance is 3.5 mm, lateral one is 2.5 mm). With further clearance decrease the wear rises sharply. Thus, during car running in a curve with the radius of 600 m and speed of 20 m/sec with longitudinal clearances decrease from 3.5 to 1.5 mm and lateral clearances decrease from 2.5 to 0.5 mm, the flange wear coefficient increases 1.8 times.

The increase of difference in wheel diameters for different wheel sets in a bogie till 20 mm and wheelsets of different bogies till 40 mm does not canse an evident rise of the wear coefficients.

4. EXPERIMENTAL RESEARCH

The estimation of freight cars wear coefficients was also performed in experiments under natural conditions. The aim of experiment was two fold. First to obtain experimentally the wear coefficients and values characterizing the loading of wheel-rail contact. Secondly, to verify the reliability of the mathematical model of freight car forced vibrations considered above.

The direct measurement of wheel and rails wear characteristics during natural tests performance is difficult. Therefore, the experimental - theoretical method is used to determine the wear coefficients. At the first stage a number of values to be important from the point of view of wear was determined during the experiment. At the second stage wear coefficients were determined by the calculation method from the dynamic values during the first stage experiments.

In the experimental part like in the theoretical calculations, wear coefficients were determined by the value of work due to frictional forces acting along the surface of the wheel flange and the rail head contact. To determine this work, a well-known expression for the wear coefficient was used [6] $\Phi = Y\alpha$, where Y is a directive force, α is an angle of attack of a wheel on a rail. Taking into account that in deriving the above mentioned expression, the momentary radius of the contact point on the flange was taken as equal to «incidence» b (or prelimination) of the contact point relative to vertical plane of the wheel set simmetry, it was corrected so that momentary radius $\rho = \sqrt{b^2 + t^2}$, where b is an incidence of a contact point, t is a vertical displacement of the contact point on the flange relative to contact point on tread. Then, the expression for work due to frictional forces Δ? at the contact point on the flange corresponding to wheel rotation on angle $\Delta\varphi$, takes the following form

$$\Delta A = \mu Y \sqrt{t^2 + \left(tg\alpha \cdot tg\beta \cdot r\right)^2} \cdot \Delta\varphi ,$$

where r is a wheel radius; β is a pitch angle of working flange surface to the horizon (for car wheels $\beta = 60°$); μ is a friction coefficient during the flange sliding along the side surface of a rail head.

For adopted values t = 0.01 m, r = 0.475 m, $tg\alpha = \alpha$; $tg\beta = 1.73$, $\mu = 0.25$ the work is

$$\Delta A = 0{,}25Y\sqrt{10^{-4} + 0{,}68\alpha^2} \cdot \Delta\varphi .$$

Taking into account that $\Delta\varphi = \Delta\ell / r$, the frictional forces work referred to the passed distance $\Delta\ell$ (the wear coefficient ΔW) is

$$\Delta W = Y\frac{0{,}25}{r}\sqrt{10^{-4} + 0{,}68\alpha^2} = 0{,}53Y\sqrt{10^{-4} + 0{,}68\alpha^2} . \quad (18)$$

In this formula a directive force Y and an angle of attack are to be determined. A guide force can be determined from the relationship

$$Y = H_p + T_{y1} + T_{y2},$$

where H_p is the frame force measured directly during experimental runnings; T_{y1} and T_{y2} are projections of friction forces between rolling surfaces and rail heads under wheel set rotation in horizontal plane on horizontal lateral axis in the process of fitting relative to the pole of rotation. These forces can be determined from the relationship

$$T_{y1} = \mu P_{1j}\frac{x_1}{\sqrt{x_1^2 + s^2}}, \quad j = 1,2 \quad (19)$$

where P_{1j} is the vertical load from left (j = 1) and right (j = 2) wheel (in the direction of test car running) of wheel set; x_1 is the pole distance for the leading wheelset.

In such a case, as far as a bogie frame is composed and there is incidence of side-frames, it is necessary to determine the pole distance of wheel set which is $x_1 = \alpha R$, where R is curve radius. Then

$$T_{y1} = \mu P_{1i}\frac{(\alpha R)^2}{\sqrt{(\alpha R)^2 + s^2}} . \quad (20)$$

Here s is half a distance between rolling circles of wheel set in the middle position (s = 0.79 m). Vertical loads P_{1i} can be determined from the test data by the following relationships:

- for right curve, when the left wheel flanges

$$P_{11} = P_{st} + \frac{\Delta P_{11} b_4 - \Delta P_{12} b_3}{2s}; \tag{21}$$

- for left curve, when the right wheel flanges

$$P_{12} = P_{st} + \frac{\Delta P_{12} b_4 - \Delta P_{11} b_3}{2s}, \tag{22}$$

where P_{st} is the static load from the wheel of the first wheelset on rail

$$b_4 = \frac{2b+2s}{2} = b_1 + s = 1.81 m,\ b_{43} = \frac{2b+2s}{2} = 0.225\ m; \tag{23}$$

ΔP_{11} and ΔP_{12} are dynamic additions of vertical loads, which have an effect on left and right axle-boxes of the first wheelset.

Angle of attack α was determined from the following relationships:

- for right curve

$$\alpha_{rt} = \frac{\ell + a}{R} - \left(\psi_f + \frac{x_{11} - x_{12}}{2b}\right); \tag{24}$$

- for left curve

$$\alpha_{lf} = \frac{\ell + a}{R} + \left(\psi_a + \frac{x_{11} - x_{12}}{2b}\right), \tag{25}$$

where 2ℓ and $2b$ are base of body and base of bogies respectively (2ℓ = 8.66 m, $2b$ = 1.85 m); ψ_f is a yaw angle of a bolster designated as positive if the bolster rotates clockwise relative to body (for right curve $\psi_f > 0$, for left curve $\psi_f < 0$); x_{11} and x_{12} are longitudinal displacements of left and right axle-boxes of the first wheel set relative to their side-frames designated as positive in the direction of car motion.

It was adopted that the wear coefficient is not equal to zero, if $\alpha > 0$ and $Y > 0$ that holds when $H_p + T_{y1} + T_{y2} > 0$.

Thus, according to this techniques for wear coefficient determination on the base of test data in the process of tests it is necessary to measure the following quantities: frame force H_p, dynamic additions of vertical forces acting on wheelset axle-boxes, angle of yaw of bolster relative to the body of that bogie where the measuring wheelset is situated, longitudinal displacements of axle-boxes of measuring wheel relative to side-frame.

The mean value of coefficient of wear while running on the sections with length $L = n\Delta\ell$ is

$$W = \frac{\ell}{L}\sum W_s \Delta\ell = \frac{\ell}{n}\sum \Delta W. \tag{26}$$

Processing of records was performed only for sections of the track with constant radius curves R_{300} = 300 m and R_{600} = 600 m, taking into account their lengths L_{300} and L_{600}. The time of recording in curves R_1 = 300 m and R_2 = 600 m is determined respectively as

$$T_{300} = \frac{L_{300}}{V}, \quad T_{600} = \frac{L_{600}}{V}. \tag{27}$$

Number of ordinates n under digitization of processes with the frequency of quantization ω:

$$n = \frac{L}{V}\omega . \tag{28}$$

Usually the frequency of quantization is $\omega = 100$ Hz.

Completely loaded open goods wagon on standard bogies (model 18-100) were used for tests. The following deviations were implemented in the test cars for determining the effects of running gears size deviations from the nominal ones in the process of testing on wear coefficients:

a) diameter difference of one wheelset was 3 mm; b) diameter difference of wheelsets in a bogie was 20 mm; c) diameter difference of wheelsets from different bogies was 40 mm; d) base difference of bogie side-frames including total longitudinal clearances between axle-boxes and guide (16 mm) was 6 mm; e) with the base difference mentioned above the total longitudinal clearance of axle-boxes in guides of side frames was 3 mm. As a reference-wagon a wagon with minimal running size deviations from the nominal ones was tested.

The pivot units and sliders surfaces of support were tested while being lubricated and non-lubricated. The experiments were conducted in curves of 300 and 600 metres in radius with the train having the test wagons.

The tests resulted in the increase of coefficient of wear with the increase of wagon speed and curve radius decrease. Wheelset misalignment corresponding to the base difference of the side-frames 6 mm, realised during the test, essentially influences the increase of the coefficient of wear with the minimum considered total longitudinal clearance of axle-boxes (3 mm). In this case the coefficient of wear under the motion of a wagon in curve with the radius of 300 mm and speed rate 10...20 m/sec increases about 40% as compared to the data obtained for the reference wagon. It is necessary to note that the value of side base differences, adopted in the tests was three times as much as the maximum admissible after the car repair in a depot.

Diameters inequality of the wheels of a wheelset also results in the essential increase of flange wear. For instance, with the difference of diameters in 3 mm while running on the curve with the radius of 300 m and speed 20 m/s, realized in the test, the flange coefficient of wear was 40 % more than that for the reference car.

The tests show that there are no appreciable changes of the coefficient of wear under the diameter differences of wheel sets in a bogie as mich as 20 mm and the diameter differences of wheel sets of different bogies equal to 40 mm.

The analysis of these data shows that the results of the test in real conditions correspond the results of theoretical calculations.

It is necessary to note that the pivot lubrication do not practically affect the coefficient of wear during the running of cars in curve with radius 300 m in speed range 10...15 m/sec. However under the high speeds the lubrication of pivots decreases coefficient of wear (under the speed 20 m/s approximately by 25...30%).

While the cars running in curve with radius 600 m the decrease of coefficient of wear is observed after the pivot lubrication even at lower speeds and reached 20...25 %.

5. CONCLUSIONS

Wheel and rail wear is considered to depend on many reasons including constructive peculiarities of the track as well as rolling stock running gears. But it is affected more by wheel and rail conditions in the process of operation.

On the whole the present paper describes the changes of wheel and rail wear being affected by one of the parameters. However the wear is generally depends on the set of parameters characterizing the conditions of rolling stock running gears. That's why the minimization of wheel and rail wear must be solved as multiparameter problem at further stages of research.

Great investigations on exposure of reasons of intensive wheel and rail wear were conducted. But this problem not considered to be completed because of its many-sided character. Not only the theoretical research but also the specially arranged tests in real conditions should be continued.

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