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IMPROVEMENT OF WORKING CONDITIONS OF THE FREIGHT LOCOMOTIVE RUNNING GEAR

Summary. To improve the working conditions of the electric locomotive running part, it is necessary to decrease unsprung masses, lateral loads and the angle of attack and provide the third body with proper tribotechnical properties. A new scheme of the freight electric locomotive running gear is elaborated in the work, which ensures a decrease of unsprung masses, relative sliding of a wheel on the rail and an angle of attack that provides resistance to the destruction of the third body in the contact zone.

УЛУЧШЕНИЕ УСЛОВИЙ РАБОТЫ ХОДОВОЙ ЧАСТИ ГРУЗОВОГО ЛОКОМОТИВА

Аннотация. Для улучшения условий работы ходовой части электровоза зону контакта колеса и рельса необходимо уменьшить необрессоренные массы, поперечные силы и угол атаки и обеспечить наличие третьего тела с необходимыми триботехническими свойствами. В работе приведена новая схема ходовой части грузового электровоза, которая позволяет уменьшить относительную скорость скольжения колеса по рельсу и угол атаки, что повышает устойчивость третьего тела к разрушению.

1. INTRODUCTION

Significant financial resources are expended annually on railway infrastructure and equipment for their efficiency. The freight locomotives are characterized by limited speed, simplicity of the running gear and relatively high power. The steering of wheels on the track is performed by the wheel and rail profiles automatically. Wheel movement on the rail is accompanied by relative sliding and great lateral forces which have a negative influence on the derailment, various kinds of damage intensity of wheels and rails, the rolling resistance and energy consumption, environmental pollution by vibrations and noise [1-4]. Variation of the wheel-rail cohesive and tractive forces stipulates alteration of the stick-slip movement of the tread surfaces with various frequencies. In general, the similar process takes place in the conditions of heavy loaded rolling/sliding frictional contact when the destruction of the third body leads to the alteration of the instantaneous increase of their cohesive force and subsequent sliding. This, in spite of the

limited speed (there has been a number of attempts to increase it), contributes to the alteration of the shearing forces, fatigue and adhesive wear and the damage rate of the locomotive running gear and track. The permanent heavy axial load, increased velocities and torques, steering and lateral forces and creep, as well as the track relief, radii of curves, incompatibility of the wheel and rail profiles, thermal action of the brake shoes on the wheel tread surfaces, etc., have negative influence on the working conditions of the freight locomotive running gear and especially wheels and rails.

Many attempts to improve the freight locomotive running gear are known.

- Decrease of unsprung masses of the electric locomotive running gear is effective, but because of structural complexity it is only used on the high-speed passenger locomotives.
- Improvement of the wheel profiles because of the wear process, plastic deformations and wear, gives a short-term effect.
- Lubrication of the wheels and rails and their thermo-chemical treatment do not always have the expected effect.

As the theoretical, experimental and exploitation researches have shown, among the ways of improvement of the running gear working conditions, the most rational one is decrease of creep, lateral forces, unsprung masses, and modification of interacting surfaces of wheels and rails.

2. PECULIARITIES OF THE MOVEMENT AND DAMAGEABILITY OF THE WHEEL - SETTING THE STRAIGHTS AND CURVES

Locomotive wheels, unlike freight wagon wheels, undergo driving torques. It is the additional source of the relative sliding velocity of interacting surfaces and skidding in extreme conditions. The movement of the wheel is not rectilinear in the straights and it is possible the interaction of the wheel flange with the rail gauge is accompanied with slight creeping (Fig. 1 a). In curves the flange or flange root of the first outer wheel of locomotive is first interacting with the rail gauge and because of great lateral force (which has the order of magnitude as the normal load and mainly depends on velocity and masses) receives the due orientation towards the rail at which the angle of attack, lateral force and movement resistance increases (Fig. 1 b).
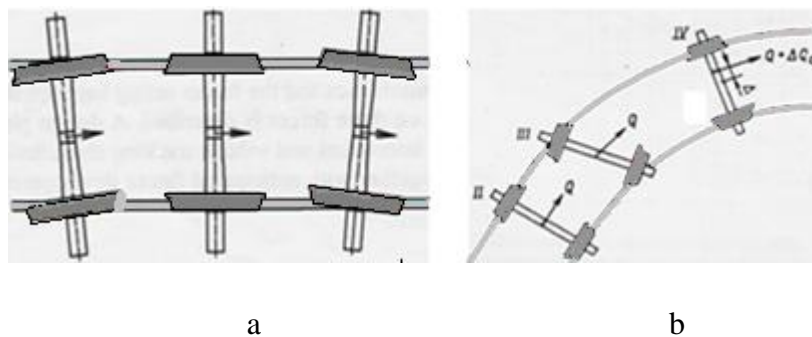

a   b

Fig. 1. Movement of the wheel-set in straight a) and in curve b)
Рис. 1. Движение колёсной пары по прямому a) и кривому b) участкам

As the length of an external branch of the rail is more than the internal one and the angular velocities of the wheels are equal, in spite of the conical form of tread surfaces of wheels, the outer wheel will lag from the inner, the angle of attack will increase and the only way of re-establishment of the initial state is sliding of the inner wheel on the rail in the direction of the opposite movement. (As the lateral force and the contact area between tread and flange surfaces for the outer wheel are more than the ones for the inner

wheel, the force of forward sliding of the outer wheel on a rail is more than the force of the backward sliding of the inner wheel). Unless it happens so the outer wheel can elevate on the rail and derail. Because of great lateral forces and unsprung masses, the seizure of the surfaces will also be great and the irrupture will occur instantly, which leads to the generation of vibrations (Fig. 2), causing various kinds of surface destruction [5].
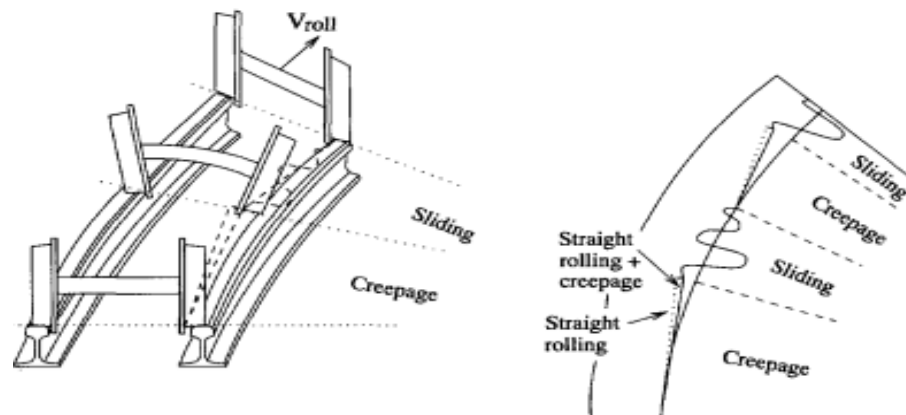

Fig. 2. Movement of wheel-set in curve and slipping of inner wheel
Рис. 2. Движение колёсной пары по кривому пути и проскальзывание внутреннего колеса

For the rail head-wheel tread contact, the sliding velocity and the contact pressure is about 0.1 m/sec and 1.5 GPa, respectively, but for the rail gauge-wheel flange the maximum sliding velocities reach 0.9 m/sec, and the maximum contact pressure was observed up to 2.7 Gpa [6]. In such conditions the main kinds of destruction of wheels and rails—fatigue damage and adhesion wear (and its heavy form—scuffing, leading to vibrations and noise) and plastic deformations are observed. During curving the instant slipping process can periodically repeat, which at approximate equal conditions of movement, can lead to the known kinds of rail destruction—rutting corrugation. In such conditions the wavelength is approximately 100-200 mm, which, at a movement speed approximately 100 km/h, corresponds to a relatively low frequency—50 Hz. The relatively high frequencies (up to 50 kHz) take place in the contact of the wheel flange and rail gauge at the stick-slip movement [7, 8].
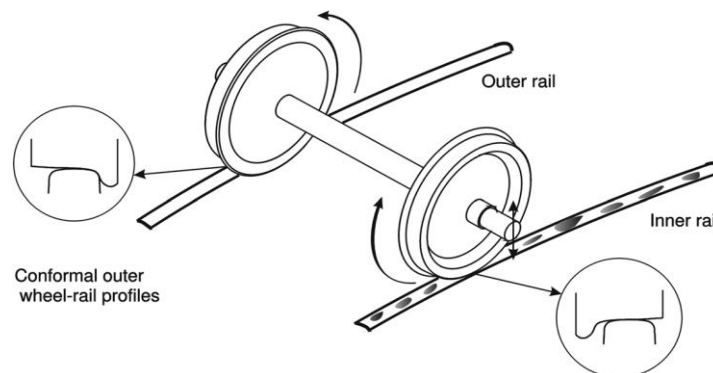

Fig. 3. Traces of rail rutting corrugation on an internal rail
Рис. 3. Следы волнового износа по внутреннему рельсу

The periodic advancement of the inner wheel at the movement of the wheel-set in the curve causes the torsion deformations of the wheel-set shaft along with its bending deformations that can lead to the crack propagation in the places of location of wheels and gears (Fig. 4 [9]).
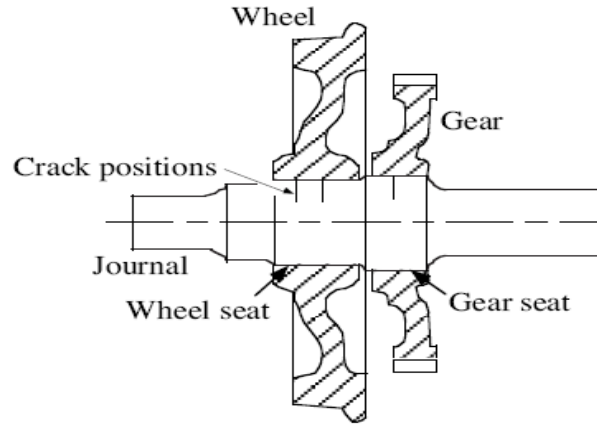

Fig. 4. Crack propagation in the places of location of wheels and gears
Рис. 4. Распространение трещин в месте установки колеса или зубчатого колеса

Therefore, to improve the working conditions of wheels and the rail set movement of the wheel-set in the curve and periodic advancement of the inner wheel and increase of the lateral force, it is necessary to return the wheel-set to the initial radial position and decrease the lateral force.

It is known about positive influence of the third body with necessary properties on the adhesive wear, scuffing and fatigue damage of surfaces. The criterion of destruction of the third body (scuffing criterion) was developed by us for the bodies working in the condition of rolling with sliding at the use of liquid oil lubricant on base of elastohydrodynamic theory of lubrication and experimental results [10-12]:

$$K\left(\frac{R}{\sqrt{R_{a1}^2+R_{a2}^2}}\right)\cdot\left(\frac{\mu_0 V_K}{P_p}\right)^{0,7}\left(\frac{P_p\beta}{R}\right)^{0,6}\left(\frac{\lambda}{\alpha\mu_0 V_{sl}^2 Pe_{1,2}^{0,5}}\right)^e \leq 1 \qquad (1)$$

where: K and e – K and e are determined experimentally and they characterize the stability of the boundary lubricating layers. The following data of K and e are obtained on the twin disk machine for mineral oils K = 0,39; e = 0,4. R—reduced radius of curvature; R_{a1} and R_{a2}—average standard deviation of the profiles of friction surfaces before scuffing; μ—dynamic viscosity of the lubricant oil in the normal conditions; P_l—linear load; β—piezo coefficient of the lubricant viscosity; λ—lubricant thermal conductivity; $\acute{\alpha}$—thermal coefficient of the lubricant viscosity; V_{sl}—sliding velocity; Vr— rolling velocity; Pe—average Pecle number ($Pe_1=bV_1/a_1$, $Pe_2=bV_2/a_2$; where b—half width of Hertzian contact; V_1, V_2—peripheral velocity of the contacting bodies).

As it is shown from the formula, the conditions of destruction of the third body depend on the mechanical and thermo-physical characteristics of interacting surfaces, geometric and kinematic parameters, and thermo-physical and tribotechnical parameters of the third body. Researches have shown also the special sensitivity of the third body on thermal loading and relative sliding, which must be taken into account to improve the working conditions of running gear of electric locomotive.

Different parts of interacting surfaces of wheels and rails need to have different properties. The friction factor for wheel flange and rail gauge face should be as low as possible—less than 0,1. Excessively high friction on tread surfaces cause severe wear, plastic flow, fatigue and low friction, which can cause poor traction and braking. For tread surfaces of a wheel and a rail, the friction factor should not be less than 0.25 and no greater than 0.4; the optimal value of friction factor for tread surfaces

is 0.35. This is one of the required conditions for normal interaction of wheels and rails, which are not properly realized yet.

To prevent the above-mentioned undesirable phenomena, it is important to control the friction factor and protect the third body against destruction. But until recently, despite a considerable quantity of works devoted to the investigations of dependences between wheel/rail and wheel/brake shoe friction forces and their durability, as well as rail cant and wheel profiles, expected results are not obtained yet. Besides, power and thermal loads on the contact zone are very high, which cause high damage rate of the third body and interacting surfaces of wheels and rails, their damage and variation of the friction factor in wide range. For example, the contact stress at initial point contact of wheels and rails (or at initial linear contact of worn-out wheels and rails) can reach 3 GPa and the contact area—1 square centimeter, the average temperature on the tread surfaces—400°C, and on the flange surfaces can exceed 800°C [9, 13], and on the brake shoe—melting temperature of metal.

3. INFLUENCE OF THE UNSPRUNG MASSES ON THE WHEEL/RAIL CONTACT LOADING AND THE SCHEME OF THE FREIGHT ELECTRIC LOCOMOTIVE RUNNING GEAR

The supporting elements of the freight locomotive traction engine are the unsprung wheel-set and the sprung part of the bogie or engine (Fig. 5). Partial leaning of the traction engine on the wheel-set increases its unsprung mass and contact loading of the wheel and rail.
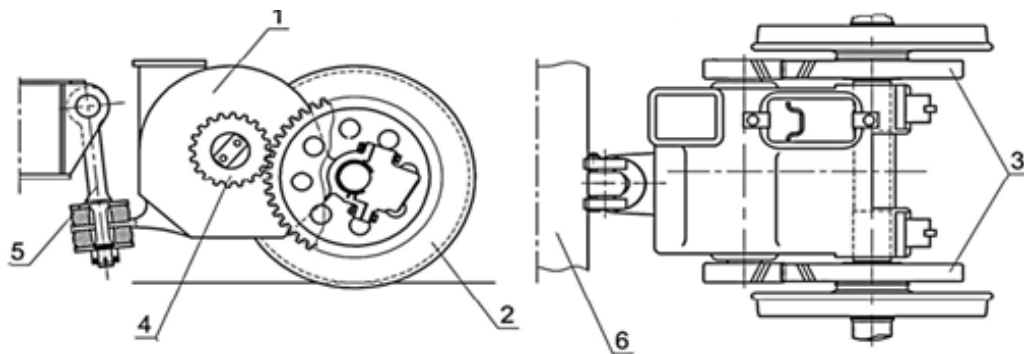

Fig. 5. The freight locomotive running gear 1 – traction engine; 2 – wheel-set; 3 – gear wheel; 4 – pinion; 5 – suspension of the traction engine; 6 – transversal beam of the bogie

Рис. 5. Ходовая часть грузового локомотива. 1 – тяговый двигатель; 2 – колёсная пара; 3 – зубчатое колесо; 4 – шестерня; 5 – подвеска тягового двигателя; 6 – поперечная балка тележки

Therefore, it is necessary to elaborate such a running gear of the freight locomotive, which will ensure to decrease the unsprung masses of the freight locomotive running gear, the lateral forces and relative slipping in the contact zone.

Such scheme is shown in Fig. 6, where the traction engine is completely sprung, and when the lateral force exceeds the allowable value in the curves, the wheel moving on the inner rail will be automatically disengaged from the running gear kinematic chain, and at a decrease of the lateral force the wheel will again be engaged in the kinematic chain.

At a movement in the curve, the flange of the outer wheel 8 will contact with outer rail (not shown) gauge and produce the axial force. This force will be transmitted to the bushing 12, which will move in the direction of this force together with the wheels 8 and 9. The teeth of half-coupling 11 will enter deeper in the teeth of the half coupling of the gear wheel 5 and the teeth of the half coupling of wheel 8 will be disengaged from the teeth of half-coupling 13 of gear wheel 1. Springs 10 and 21 return wheels 8 and 9 to

the initial position. This will cause a movement of axle 17 towards its initial position, a decrease of the angle of attack, axial force, rolling resistance and wear rate of the wheel flange and rail gauge.
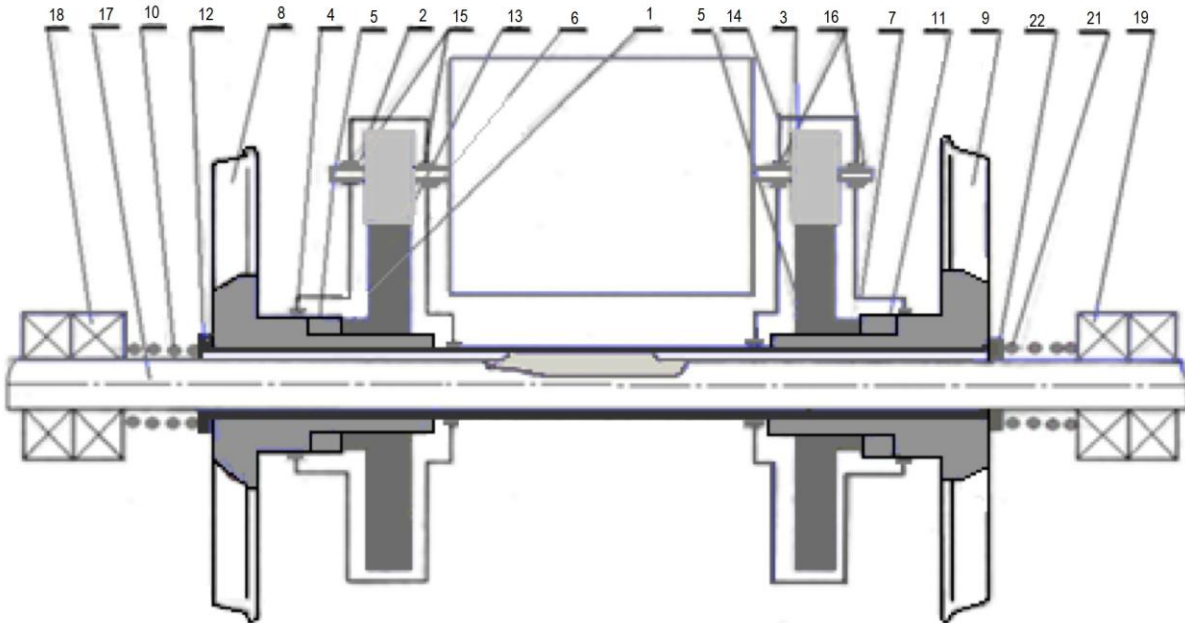

Fig. 6. A scheme of the electric locomotive running part
Рис. 6. Схема ходовой части локомотива

Teeth gears 1 and 5 together with wheels 8 and 9 and shaft 17 will roll up on driver pinions 2 and 3 (mounted on the engine shaft) at the action of the vertical loading from rails (not shown on the scheme), so this loading will not be transmitted directly to engine 1, sprung on the bogie or body. It will considerably decrease the unsprung mass (approximately by 2.5 tons). If we assume that the mass of the engine is 5 t, which is half-sprung and the mass of the wheel-set is 1.4 t, then the unsprung masses per one wheel will be $m_1 = 2$ t and in the case of the sprung engine – $m_2 = 0.7$ t. At a train velocity of 100 km/h (27.7m/s), when the rail sinusoidal wave length is 0.2 m and height is 3mm (the amplitude 1.5mm), variation of the wheel vertical displacement can be expressed as x=0,0015sinwt. Since the period of impulses is t=l/v=0.2/27.7=0,0072 s, the cyclic frequency will be w=2π/t=2.3.14/0.0072=872 1/s and coordinate x and its derivatives will be: x=0,0015sin872t; x'=0,0015*872cos872t; x''=-0,0015*872²sin872t. Maximum acceleration and maximum periodic forces acting on the masses m_1 and m_2 will be: x''=1140,6 m/s². F=x''$_{max}$*m=1140.6*2000=2281 kN, F_1=x''$_{max}$*m_1=1140.6*700=798 kN. Thus, the periodic force acting on the wheel is decreased by 1483 kN (or 2.85 times). At the same masses and velocities, if we assume that diameter of the wheel is 1 m and the eccentricity is 3 mm, then its maximum acceleration will be x''=0,0015*(55,2)² =4,57 m/s², and the maximum periodic forces acting on the wheels for unsprung and sprung masses will be: F=x''$_{max}$*m=4.57*2000=9140 N, F_1=x''$_{max}$* m_1=4.57*700=3199 N. Thus, the periodic force acting on the wheel is decreased by 5940 N (or 2.85 times).

4. CONCLUSIONS

1. A new scheme of the locomotive running gear is offered in the paper in which the relative sliding, angle of attack and lateral force are decreased by disengagement of the inner wheel from the kinematic chain at a limiting value of the lateral force.
2. The impact of the traction motor mass on the wheel-set is limited in the offered scheme due to the possibility of rotation of the wheel-set gear wheel around the motor pinion.
3. The mentioned factors decrease the wear rate of the wheel and rail, energy consumed, vibrations and noise.

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