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Improving Locomotive Traction and Adhesion Properties by Improving the Undercarriage

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ABSTRACT

The study considers the issue of increasing traction properties of freight locomotives under the conditions of the Eastern Polygon, the eastern part of the Russian railway network. The analysis results have allowed identifying design factors influencing the adhesion properties of locomotive and suggesting an extended classification of methods for improving the adhesion properties by selecting rational design solutions for the undercarriage.

It is established that for unified three-axle bogies of domestic freight diesel locomotives, the factors leading to a decrease in the adhesion coefficient in operation include a significant base length, unbalanced spring suspension and insufficient rigidity of the body supports resting on the bogie with the upper location of the kingpin, while for bogies of electric locomotives with brushless traction electric motors they include a rigid connection of the engine rotor with the wheelset, leading to the occurrence of high dynamic torques. It is proposed to use balanced pneumatic spring suspension and increase the rigidity of rubber-metal supports as measures for modernisation of diesel locomotive bogies, and for the prospective unified undercarriage of diesel and electric locomotives – to refuse from placing the fuel tank under the body and to use two-axle bogies with a frame-support traction drive with an axial gearbox, or with an axle-support drive of the aggregate type, having an elastic coupling, damping dynamic loads. The obtained results give grounds to believe that these measures will increase the coefficient of adhesion during operation from 0,27 to 0,3… 0,33..

Keywords: *railway transport, locomotive traction, wheel – rail adhesion, undercarriage, adhesion improvent, locomotive traction drive.*

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The original text of the article in Russian is published in the first part of the issue.
Текст статьи на русском языке публикуется в первой части данного выпуска.

BACKGROUND

The main requirement for locomotives for the Eastern polygon [or eastern part of the Russian railway network], which includes the Baikal-Amur and Trans-Siberian Mainlines, is the ability to haul trains weighing 7100 tons, which is possible «in case of implementation of… a traction force of at least 431 kN» [1] in a single locomotive unit, or 862 kN for a double-unit diesel locomotive. A traction force close to this indicator (786 kN) was established for 2TE137 diesel locomotive, built in 1985. Increasing the traction force to 862 kN, or by 10 %, implies bringing the calculated adhesion coefficient to 0,29. However, according to [1], «many years of experience of operating 2TE25A diesel locomotives… with an asynchronous traction drive on the on Baikal-Amur mainline has not confirmed the possibility of implementing a traction coefficient of 0,29, laid down in the technical specifications for this locomotive in all operating conditions of this mainline at a design speed of 18,5 km/h». On the other hand, the calculated adhesion coefficient of 0,29…0,32 was previously implemented on diesel TGM3 and TGM4 locomotives with hydraulic transmission, and currently on the Gravita 10BB diesel locomotives from Voith, Austria and G2000 BB from Vossloh, Germany (0,33). Thus, it is clearly premature to draw a conclusion about the possibility of implementing traction force under adhesion conditions only based on the results of operating one locomotive, without considering its design features (undercarriage, control systems, in particular, anti-skid [anti-slip] systems).

In locomotives with hydraulic transmission, where all wheelsets are mechanically connected to each other, the torque redistribution along the axles during the locomotive's movement occurs in accordance with the value of elastic slip of the wheels on the rails, and, thus, each of the axles, with a total traction force close to the adhesion limits, develops a traction force corresponding to the maximum possible for a given axle. In locomotives with electric axle drive and separate axle control, including those with brushless traction electric motors (TEM), the connection between elastic slip and axle traction force is mediated by a control system that imposes restrictions (slip measurement accuracy, slip threshold values at which the system is triggered, rigidity of the electromechanical characteristics of TEM, inertia of the control system, slip optimisation algorithms, etc.).

Thus, for locomotives with brushless TEM, the problem of choosing solutions for design of the locomotive undercarriage that ensures maximum use of wheel – rail adhesion for all axles remains relevant. The proposed article is an attempt to solve this problem.

RESULTS

Analysis of the Problem

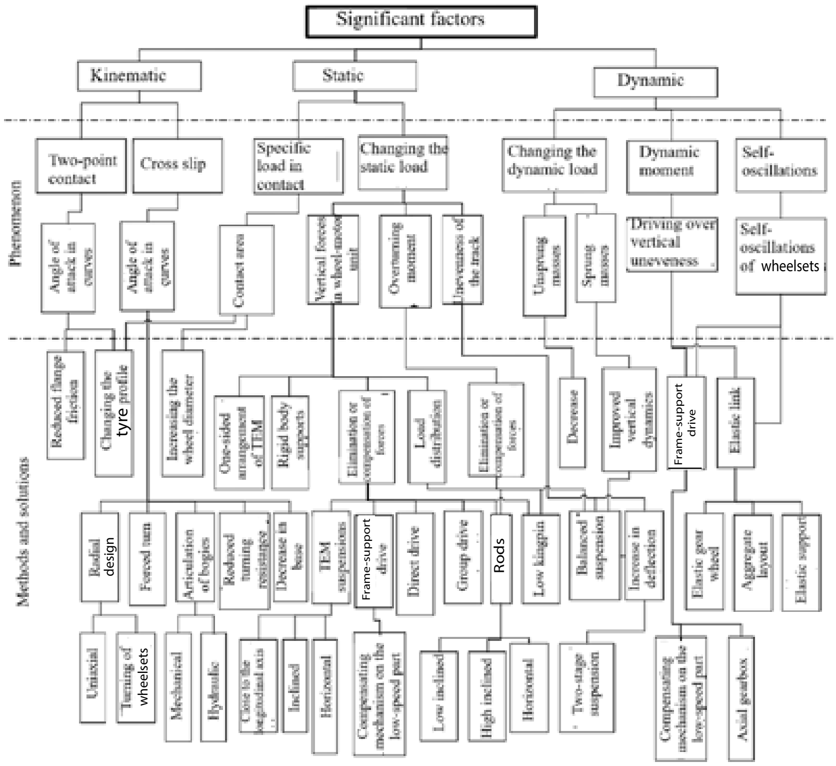
Pic. 1 shows the classification of methods for improving the adhesion properties by changing the design of the undercarriage proposed by the authors. As can be seen from the classification, all factors that affect the adhesion coefficient and at the same time depend on the design of the undercarriage can be divided into three groups: kinematic, static and dynamic factors.

Kinematic factors

Kinematic factors include those that depend on the speeds of the elements of the wheel – rail system when moving in traction mode. This is the presence of a two-point contact of the wheel with the rail and the transverse component of the sliding of the wheelset. Both factors manifest themselves in curved sections of the track, where a decrease in the coefficient of adhesion is observed depending on the radius of the curve. It is known that when starting off in a curve, the decrease in traction can be 40 … 50 %, however, this is explained not only by the two-point contact, but also by the redistribution of the load on the wheels.

Two-point contact of the wheel with the rail is the contact of the wheel with the flange of the side surface of the rail head. Since the contact surface on the flange lies on a wheel radius greater than the tread surface, this increases wheel slippage and, accordingly, reduces the traction limit with regard to adhesion (in practices – when passing curved sections of the track). This problem is partially solved by reducing friction between the flange of the wheel and the rail surface using lubrication, as well as by attempts to change the profile of the wheels. The purpose of these measures is to reduce the wear of the surfaces of the wheel and rai. But as far as the question of changing the profile of the wheel is concerned, the article by I. Pukhov «Between the wheel and the rail. The profile of the wheel affects its wear»[1] denotes a «crisis of ideas», although the possibility of a comprehensive solution to this problem is not

[1] Pukhov, I. Between the wheel and the rail. The profile of the wheel affects its wear. *Gudok*, Iss. 207 (26346) 22.11.2017.



Pic. 1. Classification of methods for improving adhesion properties by changing the design of the undercarriage, proposed by the authors of the article.

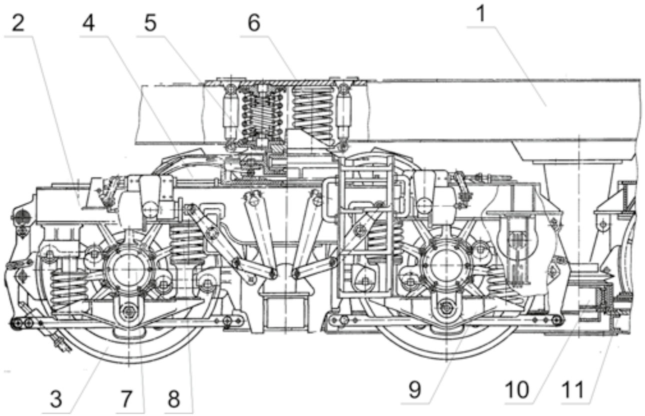
denied. Without going into the complexity of this problem, it can be concluded that when designing the undercarriage, the influence on this factor will mainly consist in providing for the possibility of placing flange lubrication devices, and this usually does not cause difficulties. The reduction in the probability of wheel contact with the flange in curves directly depends on the angle of the attack of the rail by the wheel, which will be discussed below.

According to data [2], a significant decrease in the adhesion coefficient for electric locomotives is observed in curves less than 500 m, and for diesel locomotives – in curves less than 800 m. This difference indicates the influence of the angle of the attack of the rail by the wheel, since the base of a two-axle bogie of cargo electric locomotives is about 3,0 m, and of three-axle bogies of freight diesel locomotives – 3,7…4,4 m. It is known that on the Eastern polygon, curves with a radius of less than 650 m make up 40 %, and less than 350 m – 56 % of the general network values [3]. This means the need to tighten the requirements for locomotives for the Eastern polygon to improve the horizontal dynamics of the vehicle in curves, considering not only compliance with permissible levels of impact on the track, but also the maximum implementation of the adhesion properties of the locomotive.

Reducing the angle of wheel – rail attack in curves refers to measures to reduce the impact on the track in curves and is implemented in the following ways:

- reducing the bogie base;
- reducing the resistance to bogie rotation;
- articulating bogies;
- using a mechanism for radial positioning of wheelsets in curves;
- forced rotation of bogies of wheelsets.

Reducing the bogie base is achieved both by reducing the distance between the axles and by using bogies with a smaller number of axles. The base of two-axle bogies of domestic cargo electric locomotives is 2900 … 3000 mm, while for foreign electric locomotives it is smaller – 2600 … 2900 mm. The dimensions of the base of a two-axle bogie is limited by two factors: the layout conditions (primarily the possibility of placing TEM) and the stability of movement in



Pic. 2. Bogie of TE136 diesel locomotive: 1 – body frame; 2 – bogie frame; 3 – wheelset; 4 – TEM; 5 – support and return devices; 6 – second stage of spring suspension; 7 – axle box; 8 – axle box stage of spring suspension; 9 – balance beam; 10 – kingpin assembly; 11 – low-mounted frame [9].

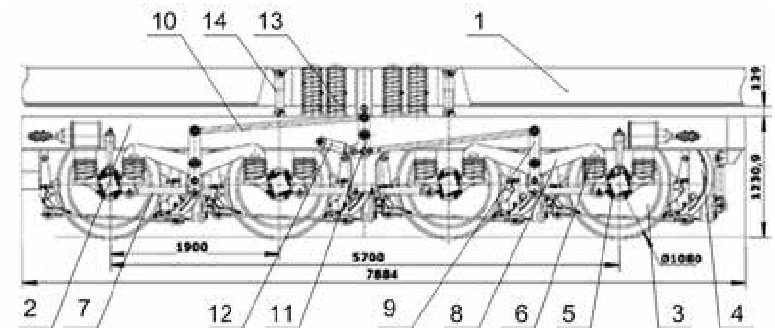
straight sections of the track. The minimum base of two-axle bogies of domestically built locomotives with a design speed of 100 km/h with a wheel diameter of 1250 mm is 2250 mm (TE136 diesel locomotive), with a wheel diameter of 1050 mm – 2100 mm (TEM7 diesel locomotive).

As can be seen from Pic. 2, the bogie of 2TE136 diesel locomotive uses a counter arrangement of six-pole collector TEM with a significant diameter. The reduction in the base under these conditions was achieved due to the connection of the bogie by a kingpin to a low-positioned intermediate frame, and because the TEM rests on the kingpin beam from above. Such a compact arrangement can be achieved by using horizontal or inclined rods to transmit force to the body.

Reducing the resistance to turning of the bogies facilitates the transition from the setting of the greatest tilt to the dynamic setting as the speed increases, which is accompanied by a decrease in the angle of attack. This is achieved by using measures to increase the stability of the bogie in straight sections of the track (introduction of transverse gaps in the kingpin, replacement of friction dampers of transverse oscillations of the bogie with hydraulic ones).

The articulation of the bogies results in the fact that the bogie in front part of the locomotive, turning when entering a curve, turns the rear bogie in the opposite direction, thereby reducing the angle of attack of the wheel of the guide wheelset of the rear bogie. At the same time, the rigid mechanical articulation of the bogies, used on the first domestic electric locomotives (for example, of VL8 series), leads to deterioration of the horizontal dynamics of the undercarriage on straight sections of the track, due to an increased tendency to wobble. The experimental use of elastic articulation of bogies on TE7 diesel locomotive and VL80 electric locomotive [4] showed that on TE3 diesel locomotive the articulation provides a significant effect when moving in a curve of a radius of 300 meters (a decrease in the quasi-static component of frame forces by 58 % at a speed of 10 km/h and setting the greatest skew), in a curve of a radius of 600 meters the articulation does not provide a significant decrease in the quasi-static component of frame forces, and the greatest decrease in the quasi-static component was obtained with hydraulic articulation of bogies. At the same time, the maximum frame forces remained at the same level due to an increase in the dynamic component. On VL80 electric locomotive, mechanical articulation of bogies also showed a decrease in the quasi-static component of frame forces and an increase in the dynamic component. It was concluded that articulation of bogies can be used for locomotives with increased axle load. Later, hydraulic articulation of bogies was experimentally applied on 2TE121 diesel locomotive [5]. As a result of the research, it was established that the hydraulic articulation works satisfactorily only with a kingpin fixed in the transverse direction, therefore, other methods of connecting the bogie to the body were recommended for the serial version.

Radial design of wheelsets («RUKP» / RDWS) was used on the three-axle bogie of JSC VNIKTI, installed on the domestic 2TE25a diesel locomotive. Research using the modelling



Pic. 3. Four-axle bogie with RDWS: 1 – main frame; 2 – bogie frame; 3 – wheelset; 4 – TEM; 5 – axle box; 6 – axle box stage of spring suspension; 7 – axle box guide; 8 – balance beam; 9, 11 – levers; 10 – traction rod; 12, 14 – dampers; 13 – second stage of spring suspension [7].

method, conducted at BSTU [6] showed that due to RDWS, «in a curve with a radius of 600 m, the work of friction forces on the ridge is less by 2,8 … 2,14 times, and on the tread surface – by 3,2 … 2,9 times. Similar results were obtained for the total work of friction forces in curves with radii of 300, 600 and 1000 m». This clearly indicates the possibility of improving the adhesion properties compared to bogies with the same base. Abroad, General Motors and General Electric developed self-steering three-axle bogies with radial design in 1993–95: the HTCR bogie for SD70 series locomotives (GM) and the bogie for AC4400CW series locomotives and later Evolution versions (GE).

JSC VNIKTI also developed a four-axle bogie with RDWS (Pic. 3), which was supposed to be used on the designed 2TE35a diesel locomotive of JSC Sinara for the Eastern polygon. According to JSC VNIKTI[2], the impact of such a bogie on the track in curves should correspond to the impact of the three-axle bogie of the 2TE116 diesel locomotive.

A special case of a bogie with radial design of wheelsets can be a four-axle bogie with slave axles. As indicated in [4], the front wheelset in such bogies in the direction of travel can even have a negative angle of attack, but this requires that the characteristics of the return devices change depending on the radius of the curve and the unquenched centrifugal force, which requires making them controllable. In this regard, three-axle bogies with an intermediate frame are indicated as a preferred design in [4].

[2] Lunin, A. A., Spirov, A. V. On the directions of development of rolling stock. Presentation of JSC VNIKTI 30.08.2019. [Electronic resource]: https://www.railwayexpo.ru/images/docs/2019/presentation/ДЕНЬ 3_30 августа/ЗАЛ 2/10–00_6-й международный форум/Сессия 1/1.7_А_Лунин_А_Спиров_RU.pdf. Last accessed 27.12.2023.

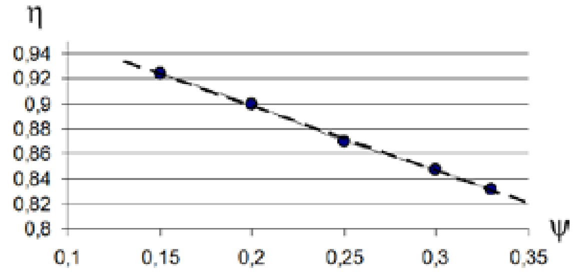
An example of practical implementation of a chassis with forced axle rotation is the Integral diesel train from Jenbacher (Austria) with single-axle bogies. This design has not received further development. In domestic practices, work has been carried out to create systems for forced rotation of industrial locomotives' bogies on steep curves, but it is too early to talk about the possibility of using such systems on mainline locomotives.

From the analysis of kinematic factors it follows that the simplest technical way to reduce lateral sliding in curves and at the same time reduce the probability of two-point contact is the maximum reduction of the bogie base, that is, the use of two-axle bogies with a base of 2250…2600 mm.

Static factors

The static factors in the classification include those that are determined by the interaction forces of the wheel and rail and that change little over a period of time corresponding to the period of the undercarriage's own vertical and horizontal oscillations in motion.

First, the static factors include the specific contact load, which depends on the contact area and can be changed in two ways: by increasing the wheel diameter and by changing the tyre profile. The problematic nature of finding a rational tyre profile has already been discussed above. As for changing the tyre diameter, according to VNIIZhT, in [2] it is predicted that increasing the wheel diameter from 1050 to 1250 mm will lead to an increase in the adhesion coefficient by 4 %, with reference to the research of K. Kraft [7]. K. Kraft's conclusion has not been tested under the conditions of domestic railways. Indirect argument in his favour is associated with the higher average statistical coefficient of adhesion of electric locomotives with wheels of



Pic. 4. Change in the coefficient of use of the adhesion weight η depending on the traction coefficient ψ for the bogie of 2TE116 diesel locomotive, standard version [9].

1250 mm diameter, compared to diesel locomotives with wheels of 1050 mm (which is also explained by the lower probability of pollution of the track with oil products and the smaller base of the bogies), as well as the higher adhesion properties of 2TE121 diesel locomotive with wheels of 1250 mm. In addition, an argument in favour of using wheels of 1250 mm diameter on locomotives with increased traction force per axle is that for wheels with tyres this increases the reliability of the fit of the tyre on the wheel centre (on diesel locomotives in the USA with wheels of 1050 mm diameter, wheels without tyres are used).

A more specific static factor is the change in the static axle load when the locomotive is moving in traction mode. Redistribution of the axle load may occur for the following reasons:

– the presence of vertical components of forces in the wheel-motor units in traction mode;

– the presence of an overturning moment due to the fact that the traction force is transmitted by coupling devices above the level of the rail head;

– moving over vertical irregularities of the track.

Separately, it is necessary to mention the hypothesis put forward in [2] about the influence of uneven distribution of loads on the wheels of the wheelset with one-sided traction transmission. As shown by the results of studies not included in [8], when the locomotive is moving in the coasting mode, there are tangential stresses in the axles of the wheelset caused by a torque that changes randomly depending on the position of the wheelset in a straight line, the cause of which is the difference in the diameters of the wheels corresponding to the points of contact of the wheel and the tyre. Due to the conicity of the wheel tyre profile, this difference changes depending on the position of the wheel in a straight line. In the traction mode, approaching maximum traction force with regard to adhesion, the magnitude of the tangential stresses becomes constant, which contradicts the hypothesis [2], since otherwise the force would change randomly due to a random change in the tangential rigidity of the wheel – rail contact.

Reduction of vertical forces can be achieved by the following methods:

– elimination of bogie frame rotation in the vertical plane by using rigid body supports on the bogie;

– one-sided arrangement of TEM in three-axle bogies, which leads to contrary directions of forces applied to the bogie frame through the traction motor suspensions located on different sides of the rotation axis in the vertical plane;

– compensation of forces in two-axle bogies by locating TEM suspension points to the rotation centre;

– use of traction drives that do not create uncompensated reaction forces in the vertical plane;

– compensation of vertical forces by reducing the distance between the point of force transfer from the bogie to the body and the level of the rail head, which can also be used to compensate for the overturning moment.

As studies have shown [9], the coefficient of use of the adhesion weight for the three-axle bogie of 2TE116 diesel locomotive in the range of traction coefficient changes from 0,15 to 0,33 changes almost linearly (Pic. 4).

According to calculations [9], for the standard version of 2TE116 diesel locomotive bogie with $\psi = 0,33$, the value of $\eta = 0,832$. Changing the design of the bogie gave the following results:

– reducing the static deflection of the bogie supports from 20 mm to zero (installing rigid supports) increases η from 0,832 to 0,869 due to the elimination of rotation of the bogie frame in the vertical plane;

– the use of balanced spring suspension in combination with rigid supports increases η from

0,832 to 0,922 due to the uniform distribution of loads along the axles of the bogie;

– the use of a frame-support drive with a transmission mechanism on the high-speed part of the transmission, eliminating the vertical component of the forces from the traction drive, but with an unbalanced spring suspension, increases η from 0,832 to 0,902, that is, increases η compared to a axial-support drive from 0,869 to 0,902; when using a frame-support drive with an axial reducer, due to the vertical forces in the reaction rods, η with a three-axle bogie worsens to 0,848.

The need to use a one-sided arrangement of TEM on a three-axle bogie was also confirmed (the counter arrangement reduced η from 0,832 to 0,79 even with rigid supports).

Thus, for a multi-axle bogie in a rigid frame, it is advisable to use a balanced spring suspension, including due to the uneven load on the axles, which is also stated in [9]. In this case, a balanced suspension in the axle box stage is equivalent in its dynamic qualities to an unbalanced suspension with a greater static deflection, since the change in axle load when passing a vertical unevenness is redistributed across the other axles of the bogie. In practice, this makes it possible to abandon the second suspension stage, at least for locomotives with a design speed of up to 120 km/h. In fact, a balanced single-stage suspension was also used on TEP60 passenger diesel locomotive with a speed of up to 160 km/h, since the body rested on the bogie through pendulum supports with rubber cones that had an insignificant static deflection, and the springs between the bogie and the body participated only in the lateral rolling process.

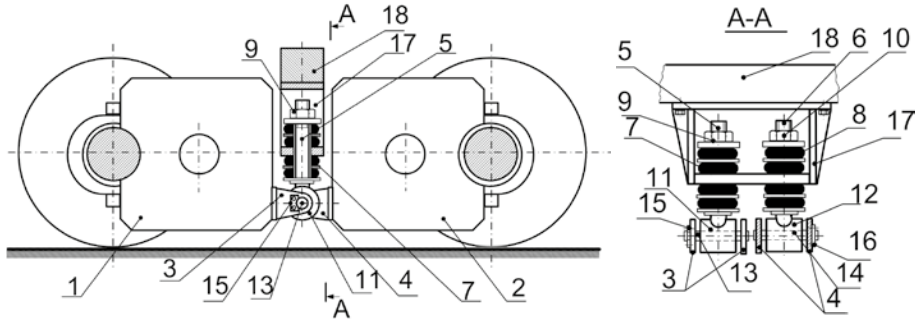
Also, for a multi-axle bogie in a rigid frame, the coefficient of utilisation of the adhesion weight can be increased by using frame-support traction drives with a compensating mechanism on the low-speed part of the drive (which was implemented on the experimental diesel locomotive TE120), as well as a direct traction drive, in particular, an axial-support drive with a horizontal arrangement of the reactive traction, transmitting forces from the stator to the bogie. At the same time, the presence of a radial design of wheelsets complicates the possibility of implementing a frame-support drive due to the occurrence of movements of the outer wheelsets in the horizontal plane, and the use of a direct axial-support drive requires elastic support of TEM on the axle or wheel centres to reduce the unsprung mass.

For two-axle bogies, as a rule, a counter arrangement of the traction motor is used, which allows reducing the moment of inertia of the bogie in the vertical and horizontal planes. In this case, the main way to increase the coefficient of use of the adhesion weight is to compensate for vertical forces and the overturning moment by reducing the level of the point of application of the traction force transmitted from the bogie to the body using a low-positioned kingpin (including a low-positioned intermediate frame), or horizontal or inclined rods.

The use of inclined rods, the direction of which passes through the middle of the longitudinal base of the bogie at the level of the rail heads, allows for full compensation of traction load changes on wheel pairs in one bogie and ensures the value of the coefficient of use of the adhesion weight η > 0,92 with a soft two-stage spring suspension, which determines the widespread use of inclined rods as devices for transmitting traction force from the bogie to the body in modern locomotives. Low-mounted kingpins compensate for the decrease in traction force to a lesser extent.

In modern locomotive designs, inclined rods acting on compression and tension are used. Rods acting only on tension are not used in new locomotives, as are horizontal rods. In this case, the inclined rods are directed towards the nearest automatic coupling device, as a result of which the front bogie of the locomotive is additionally loaded with the vertical component of the reaction force from the traction side, and the rear one is unloaded, and this compensates for the bogie-by-bogie change in load from the action of the overturning moment caused by the traction force applied at the level of the automatic coupling device.

For two-axle bogies of shunting locomotives with single-stage spring suspension, one of the measures to reduce the uneven distribution of the axle load may be the location of TEM suspensions on the axis passing through the centre of gravity of the bogie. The version of such a device proposed by the authors is shown in Pic. 5. During transverse and angular movement of TEMs 1 and 2, rods 5 and 6 of the suspensions tilt due to the mobility of hinges 11 and 12 and sets of rubber washers 7 and 8. When the traction force is applied, in vertically located rods 5 and 6, counter-directed forces arise, which are transmitted to the crossbeam 9. Since hinges 11 and 12 are located on the same axis, the forces



Pic. 5. Arrangement of TEM suspensions on an axis passing through the bogie's centre of gravity. 1, 2 – TEMs; 3, 4 – brackets; 5, 6 – rods; 7, 8 – rubber washers; 9, 10 – nuts; 11, 12 – hinges; 13, 14 – shafts; 15, 16 – strips; 17 – crossbeam; 18 – bogie frame [10].

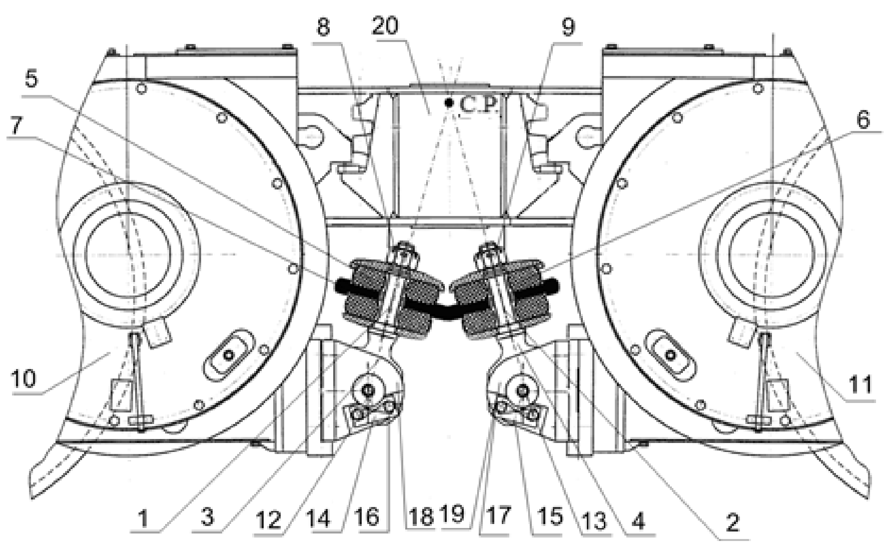
in rods 5 and 6 are directed in opposite directions and are applied to the same crossbeam 9, as a result of which the indicated forces mutually compensate each other and do not create a redistribution of loads along the axes. The authors received a patent for a utility model for the proposed design [10].

For shunting diesel locomotives with asynchronous TEM and rigid gear transmissions (for example, TEM23), the authors proposed a design in which suspensions, including hinges and flat shock absorbers that dampen impacts when passing track irregularities, are attached to a trough-shaped crossbeam (Pic. 6). A patent for a utility model was also received for this design [11].

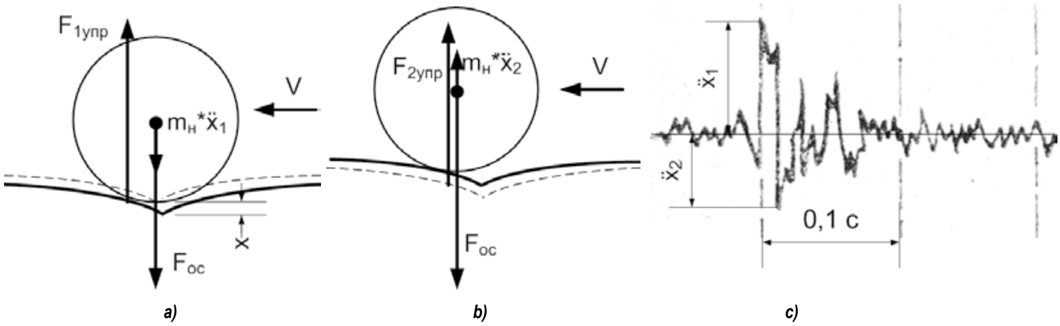
With an unbalanced spring suspension, passing short irregularities without taking into account the vibrations of the over-spring structure can be considered a quasi-static process leading to a redistribution of loads along the axles. A reduction in the influence of this factor can be achieved using the above-mentioned balanced spring suspension or by increasing the overall static deflection of the spring suspension.

It should be noted that both ways (single-stage balanced spring suspension and two-stage suspension with traction force transmission by inclined rod) allow to achieve approximately the same values of η. Thus, for the bogie of TEM21 diesel locomotive with inclined rod and two-stage suspension $\eta = 0,921$, for the bogie of 2TE121 diesel locomotive with balanced single-stage spring suspension $\eta = 0,92$.

A separate issue is the feasibility of using a group [opertion] drive for electric locomotives



Pic. 6. Device for fastening the locomotive traction motor to the bogie frame: 1, 2 – suspensions; 3, 4 – hinges; 5, 6 – sets of flat elastic elements; 7 – crossbeam; 8, 9 – nuts; 10, 11 – traction electric motors; 12, 13 – shafts; 14, 15 – locking strips; 16, 17 – bolts; 18, 19 – brackets; 20 – bogie centre of gravity [11].



Pic. 7. Changes in axial load during oscillations of unsprung masses:
a) deformation of the joint immediately after the wheel passes; b) return to the initial position;
c) approximate graph of accelerations of the unsprung mass [diagram is made by O. V. Izmerov].

and diesel locomotives with electric transmission. On the one hand, mechanical articulation of the axles compensates for uneven distribution of adhesion properties within the bogie, which simplifies the control of traction force. On the other hand, such a drive requires an increase in the power of TEM used and complicates the repair of the undercarriage. In connection with these circumstances, the use of a group drive in the near future makes sense primarily for industrial locomotives with lower TEM power, and therefore the effect of a group drive on increasing the adhesion coefficient is not considered in this article.

Dynamic factors

In the classification, dynamic factors include those that are determined not only by the geometric dimensions, static forces and displacement values of the structural elements, but also by the mass of the elements and the elastic-dissipative properties of the connections between them. Dynamic factors are divided into groups corresponding to the phenomena that cause their manifestation:

– the presence of a dynamic (variable) component of the axial load when moving over uneven track;

– the presence of a dynamic component of torque in the shaft lines of the traction drive;

– the development of frictional self-oscillations when the wheelset slides along the rails.

The dynamic component of the axial load, in turn, consists of dynamic forces during oscillation of unsprung and sprung masses.

We will consider the effect of oscillations of unsprung masses on the adhesion properties using the example of a wheelset passing a joint unevenness (Pic. 7).

After the wheelset passes the bend angle in the joint (Pic. 7 *a*), the vertical component of the unsprung mass speed instantly changes, and the kinetic energy of the unsprung mass must be compensated by converting it into potential energy of deformation of the track superstructure by the value *x* and thermal energy due to friction in the track superstructure at the specified dynamic deformation. In this case, an acceleration of the wheelset $\ddot{x}_1$ occurs, directed upward, as a result of which the maximum total force of the wheelset load on the rails will be (when representing the load of the sprung masses as quasi-static):

$$F_{sum1} = m_s g + m_u \left(g + \ddot{x}_1\right) = F_{ax} + m_u \ddot{x}_1 \,, \qquad (1)$$

where g = 9,81 m/s² – acceleration of gravity;

m_s – the amount of sprung mass per axle,

m_u – the amount of unsprung mass per axle,

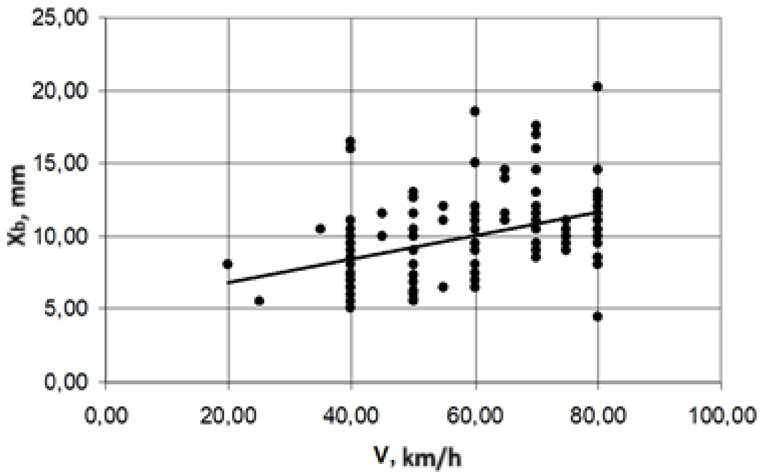
$\ddot{x}_1$ – maximum acceleration upon impact (Pic. 7 *c*),

F_{ax} – static axle load.

After the vertical component of the wheelset speed has been compensated, the track superstructure, under the action of elastic forces, tends to return to its original position, imparting a vertical speed directed upward to the wheelset, and the potential energy of the deformed superstructure is converted into the kinetic energy of the unsprung mass. After the track superstructure returns to the position corresponding to the deformation under static load, the upward movement of the wheelset slows down due to the action of the axial load (Pic. 7 *b*), and the force of the wheelset load on the rail will be:

$$F_{sum2} = m_s g + m_u \left(g - \ddot{x}_2\right) = F_{ax} - m_u \ddot{x}_2 \,, \qquad (2)$$

where $\ddot{x}_2$ – maximum acceleration during the negative half-period of oscillations of unsprung masses after impact (Pic. 7 *c*).



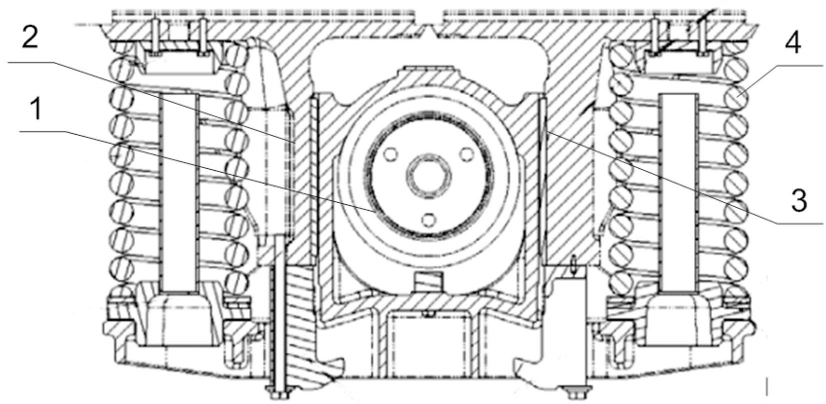
Pic. 8. Maximum vertical displacements of the axle box relative to the bogie frame of the 2TE121–011 diesel locomotive, Vorkuta – Sosnogorsk section [according to data obtained by O. V. Izmerov].

Thus, the dynamic unloading of the wheelset by the value $m_u \bullet \ddot{x}_2$ occurs. Due to the dissipation of energy during deformation of the track superstructure $\ddot{x}_2 < \ddot{x}_1$. The values $\ddot{x}_2$ have not been studied to date, but it can be stated that $\ddot{x}_2$, on average, will be proportional to $\ddot{x}_1$, and its value for a single joint unevenness will depend on the nonlinearity of the rigidity characteristic of the track superstructure, the degree of dissipation of the energy of oscillations of unsprung masses and the presence of other forms of oscillations of unsprung masses. As a first approximation, it can be assumed that $\ddot{x}_2 = 0,75 \bullet \ddot{x}_1$. Thus, for the 2TE25a diesel locomotive at $\ddot{x}_1 = 50$ m/s², m_u = 3900 kg and F_{ax} = 245 kN, we obtain a minimum value of the dynamic axle load of about 100 kN, or a decrease of up to 0,41 from the average value.

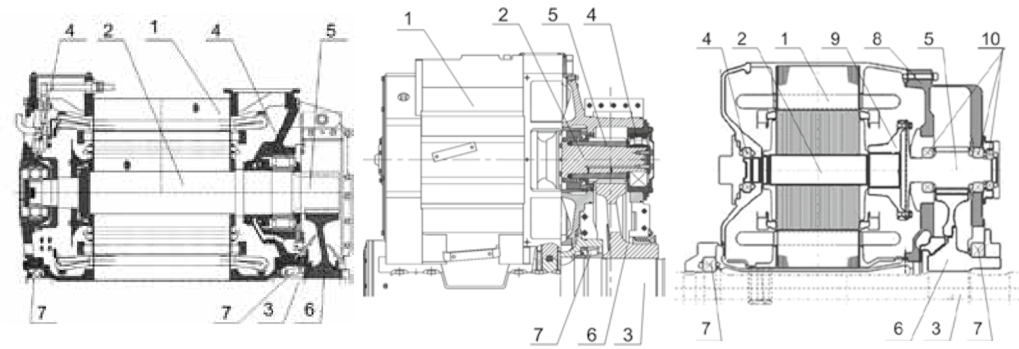
However, when moving on to the effect of dynamic axle unloading on the locomotive's adhesion properties, it is also necessary to take into account the duration of the unloading and the rotational inertia of the wheelset when adhesion is lost, since the frequency of unsprung mass oscillations in the lowest form with a support-axle drive for different locomotives is usually in the range of 20…40 Hz. This issue has not been studied at present, so we can limit ourselves to the conclusion about the advisability of reducing the share of unsprung mass in the total mass of the locomotive, which is primarily achieved by using a frame-support drive.

The axle load, when it changes due to oscillations of the sprung masses occurring with a frequency of 1,5…2,5 Hz (body) and 4…6 Hz (bogie), in relation to the dynamic processes in the traction drive when the wheel slides along the rail, can be considered quasi-static. Thus, the change in the axle load can be estimated by dividing the amplitude of the vertical displacement of the axle box relative to the bogie frame, when the locomotive is moving, by the value of the static deflection of the axle box stage. Thus, for 2TE121 diesel locomotive, during tests on Vorkuta – Sosnogorsk section, the average maximum amplitudes of the vertical displacements of the axle box at a speed of 27 km/h were 7,5 mm (Pic. 8) with a static deflection of the axle box stage of 130 mm; accordingly, the axle load can be reduced to 0,942 of the average value. It is possible to reduce the influence of vertical oscillations by improving the vertical dynamics of the undercarriage (using balanced spring suspension, increasing the static deflection).

Also, when the sprung masses of the bogie oscillate, the friction in the pedestal («jaw») axle boxes may affect the adhesion of the wheel to the rail. According to [2], the sliding speed of the driving axles of a locomotive with pedestal axle boxes is 5 % higher than with jawless ones. Pedestal axle boxes continue to be used on diesel locomotives made in the USA, in particular, on the TE33a diesel locomotive, which is operated in some countries neighbouring Russia (Pic. 9). The influence of friction in friction dampers of oscillations of the first stage of spring suspension, widely used on domestic diesel locomotives, has not been studied at present, however, it should be considered that in dampers, unlike jaw axle



Pic. 9. Jaw box of TE33a diesel locomotive: 1 – axle box; 2 – jaws; 3 – frame; 4 – axle box suspension spring [30].



a) b) c)

Pic. 10. Axle-support drive with rigid gear transmission: a) TE33a diesel locomotive; b) BKG-1 electric locomotive; c) 2EV-120 electric locomotive; 1 – TEM; 2 – TEM shaft; 3 – wheelset axle; 4 – rotor bearing; 5 – small gear wheel; 6 – large gear wheel; 7 – axle bearing; 8 – gearbox housing; 9 – diaphragm coupling; 10 – gearbox bearings [29].

boxes, the friction force does not increase proportionally to the traction force.

It is known that the dynamic component of the torque in the traction drive, arising when driving over vertical unevenness of the track and vibrations of the locomotive's superstructure, has a significant effect on the adhesion properties of the locomotive. Thus, it was found that on 2TE10L diesel locomotive, which has a axle-support drive with a rigid gear transmission, when driving track sections with wave-like wear, a decrease in the adhesion coefficient of 20 % or more was observed [2]. Moreover, in [12], the dynamic torque with a rigid gear transmission is considered as the cause of wave-like wear of the rails. Studies [4] have shown that with a rigid gear transmission of the 2TE10L diesel locomotive, the dynamic torque on the shaft of the traction electric motor (TEM) reached 9,4 kNm. With such a torque value and the presence of wheel – rail adhesion, the axial traction force should be 82 kN, and $\eta = 0{,}39$, which indicates the possibility of wheel slippage along the rail, when the resistance to the dynamic torque is created by both the sliding friction force and the inertia of the rotating masses of the wheel.

As indicated in [12], the use of an elastic wheel in the axle-support traction drive of cargo diesel locomotives, leading to a decrease in the dynamic torque by approximately three times, reduced the tendency to skidding and reduced the wear rate of the tyres by 15 %. As a result, by the 80s of the last century, a transmission with an elastic link was introduced on all mainline freight diesel locomotives with a axle-support drive, and an elastic gear wheel (EGW) was created for use in a two-way traction transmission of domestic freight electric locomotives. On the other hand, the use of brushless TEM on foreign freight locomotives has now led to a return to rigid gear transmission (Pic. 10).

The choice of design solutions in the drives shown in Pic. 10 is due to the desire to reduce the diameter of the small gear wheel of the transmission to increase the gear ratio in a single-stage transmission and increase the maximum number of revolutions of the traction electric motor, which in turn allows to reduce the unsprung mass. In the drive of TE33a diesel locomotive (ES44ACi, developed by General Electric, USA), to reduce the diameter of the small gear wheel, the teeth are cut on TEM shaft itself (Pic. 10a), which requires the manufacture of the shaft from high-alloy steel and disassembly of TEM with pressing out of the rotor steel package in case of wear of the shaft-pinion, or restoration of the teeth by surfacing. In the BKG-1 electric locomotive drive (developed by the Datong Plant, China, jointly with Alstom, France, based on the Alstom Prima 47000 electric locomotive), the small gear wheel is fitted onto TEM shaft with tension, but due to the small diameter of the gear wheel, the wheel had to be placed between the rotor bearings to increase the hub (Pic. 10b). In the 2EV-120 electric locomotive drive (developed by Bombardier, Canada), to increase the service life of the bearings, one end of the rotor shaft rests on the pinion shaft of the axial gearbox (Pic. 10c). In this case, the traction transmission is made in all cases with a slight inclination of the teeth to reduce impacts in the engagement, as a result of which transverse forces arise in it, which complicate the creation of EGW. Another factor complicating the use of EGW was reduction of the traction transmission module to 8…9 mm, which increases the influence of the radial clearance in the connection of the gear ring crown with the hub on the durability of the transmission.

Calculation of dynamic torque acting in a traction drive with a rigid gear transmission is complicated by the complexity of calculating vibration and impact loads during tooth shifting and the lack of accurate data on a number of parameters (for example, on the contact rigidity of the teeth in the presence of lubrication on the surface) and taking into account the slippage of the wheel on the rail. In this regard, we will try to approximately estimate the magnitude of the dynamic torque in the drive of the TE33a diesel locomotive, provided that there is no backlash and the adhesion of the wheelset to the rail is maintained.

Based on [11], the dependence of the dynamic torque on the shaft of TEM of the wheelset on the vertical accelerations of the wheelset axle can be determined as follows:

$$M_{\mathrm{d}} = \frac{I_{\mathrm{r}}(i+1)}{c}\ddot{x}_{\mathrm{ws}}\,, \quad (3)$$

where $\ddot{x}_{\mathrm{ws}}$ – vertical acceleration of the axle of the wheelset, m/sec;

c = 0,85 m – distance between the point of suspension to the frame (the turning point of the wheel-motor unit when passing vertical unevenness) and the axle of the wheelset,

I_r – moment of inertia of TEM rotor; i = 5,3125 – gear ratio (85:16).

The vertical acceleration of TEM above the axle of the wheelset is determined based on information about the acceleration of the wheelset of the analogical drive:

$$\ddot{x}_{\mathrm{ws}} = \ddot{x}_{\mathrm{wsa}}\sqrt{\frac{m_{\mathrm{ua}}}{m_{\mathrm{u}}}}\,, \quad (4)$$

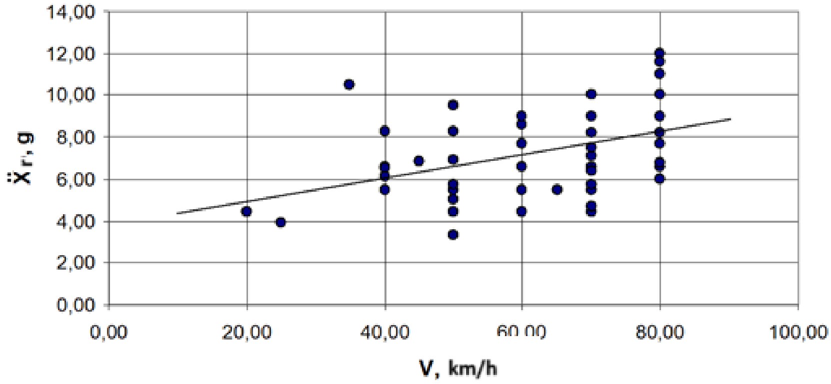
where $\ddot{x}_{\mathrm{wsa}}$ –average maximum values of vertical accelerations of the wheelset of the analogical locomotive at the speed of movement on the ruling grade. As an analogue we will take the 2TE121diesel locomotive (Pic. 11), and the speed of movement on the on the ruling grade will be assumed to be 30 km/h (which is acceptable, since the calculation is made with the aim of analysing primarily the qualitative side of the phenomenon), then from Pic.10 $\ddot{x}_{\mathrm{wsa}} = 55$ m/s². m_{ua} = 4200 kg – the value of the unsprung mass of the analogue, falling on one axle. As for the unsprung mass of TE33a diesel locomotive, we proceed from the fact that the mass of 5GEB30A1TM TEM is 2268 kg against the mass of TEM of 3100 kg on the 2TE10L diesel locomotive with jaw axle boxes and an unsprung mass of 4440 kg, and this gives grounds to accept the value of the unsprung mass for the TE33a as less than for the 2TE10L locomotive by an amount equal to half the difference in the masses of TEM, i.e. m_u = 4000 kg. With the specified values of the parameters, we obtain $\ddot{x}_{\mathrm{ws}} = 56$ m/s².

The moment of inertia of TEM rotor is approximately equal to the moment of inertia of a cylinder in the form of a package of electrical steel with a density of γ = 7650 kg/m², a radius of r_r = 0,225 m and a length of l = 0,5 m:

$$I_{\mathrm{r}} = \gamma\pi\frac{r^4 l}{2}\,. \quad (5)$$

With the specified parameters I_r = 15,4 kg/m².

Substituting the obtained values into (3), we obtain the average maximum value M_d = 6,4 kNm, which corresponds to the dynamic component of the traction force on the axle of 65 kN and the traction coefficient ψ = 0,28. This means that the total static and dynamic traction force in this case would exceed the adhesion limit. Thus, the rigid gear transmission on the TE33a diesel locomotive should lead to the same problems with deterioration of adhesion when passing vertical unevenness of the track, which



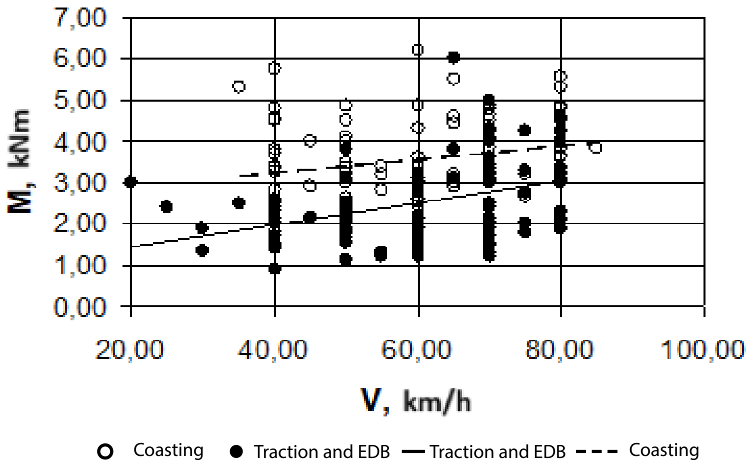
Pic. 11. Maximum vertical accelerations of the axle gearbox of the 2TE121–011 diesel locomotive, Vorkuta – Sosnogorsk section [according to data obtained by O. V. Izmerov].

were previously observed on the 2TE10L diesel locomotive, despite the fact that the use of an asynchronous TEM made it possible to reduce the TEM mass by 1,4 times and the moment of inertia of the TEM anchor by almost half.

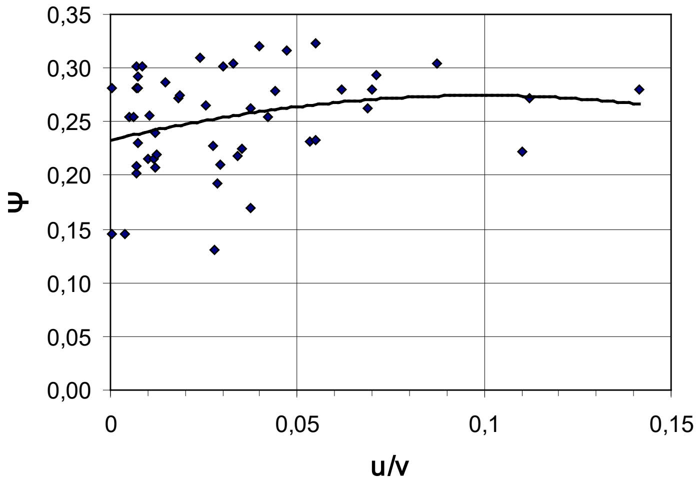
As noted above, the use of elastic links in the shafting of the axle-support drive makes it possible to reduce the maximum values of the dynamic torque by almost three times. An effective reduction of the dynamic torque is also achieved by using a frame-support drive. As shown in Pic. 12, in the frame-support drive with an axial gearbox of the 2TE121 diesel locomotive, the highest maximum torques are observed in the coasting mode; in the traction mode at a speed of 50 km/h (which corresponds to the hourly speed of freight electric locomotives), the value of the average maximum dynamic component of the torque on the shaft of the axial gearbox is approximately 2,25 kNm, which for this locomotive corresponds to a traction force per axle of 15,5 kN and a traction coefficient of ψ = 0,06.

With a hollow shaft frame-support drive of a two-axle bogie of the VL84 electric locomotive in the version with quill driver, the maximum value of the dynamic component of the torque was 6,5 kNm [14], which corresponds to a traction force per axle of 30 kN and ψ = 0,12; for bogies with rigid body supports in a hollow shaft frame-support drive, the dynamic torque is insignificant.

Another way to reduce the dynamic component of the torque in the drive and, at the same time, the unsprung mass is the elastic support of the traction motor on the axle or wheel centre (the so-called centre-support drive). As studies have shown [6], such a drive



Pic. 12. Maximum dynamic torque on the shaft of the axle gearbox of the 2TE121–011 diesel locomotive, Vorkuta – Sosnogorsk section, in different driving modes [according to data obtained by O. V. Izmerov].



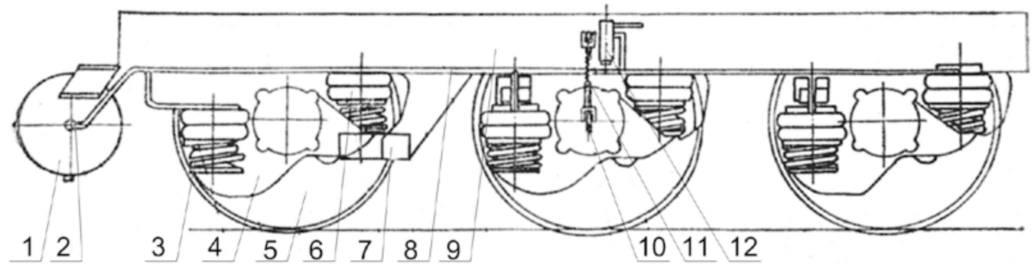
Pic. 13. Empirical dependence of the axle adhesion coefficient ψ of 2TE121–003B diesel locomotive on the relative slip speed u/v in case of development of self-oscillations of the wheelset [according to data obtained by O. V. Izmerov].

can be effective with a wheel diameter of 1250 mm, since with a wheel diameter of 1050 mm there is insufficient space to accommodate elastic elements that provide a sufficient reduction in the vertical accelerations of TEM. In general, the efficiency of elastic support of the traction electric motor on the wheelset decreases with an increase in the specific torque per unit mass of TEM, since the ultimate deformations of the elastic elements from the action of the torque limit the possibility of reducing their rigidity in the radial direction. Thus, the use of a drive with elastic support of the traction electric motor on the axle of the wheelset for freight locomotives makes sense only in the case of electric locomotives with collector traction electric motors and an axial power increased to 1000 kW or more.

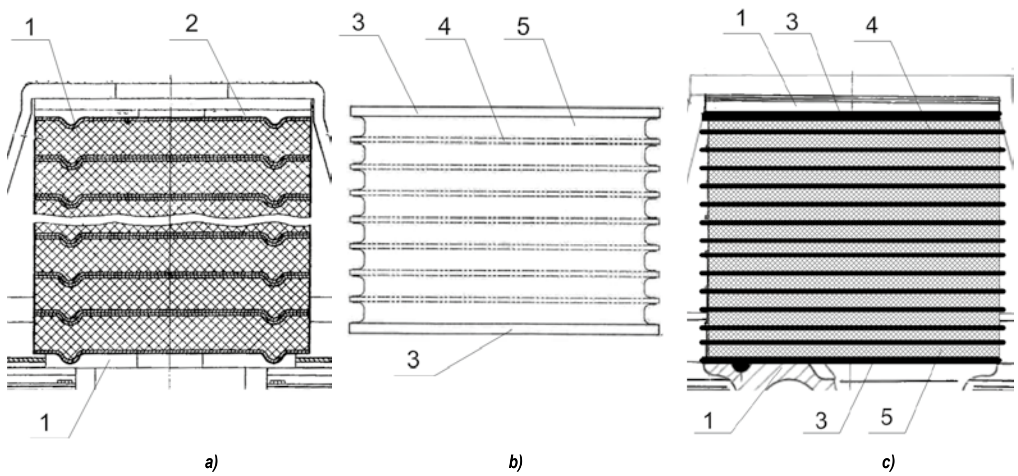
It is known that when skidding begins, the friction coefficient between the wheel and the rail increases if frictional self-oscillations occur in the system of the undercarriage part of the drive [15; 16]. Thus, according to [15], for the industrial 14KR1 electric locomotive in case of intense frictional self-oscillations of the wheelset at an average sliding speed of 12 km/h, the average value of the tangential force of the wheelset increases by 38 % compared to skidding in the absence of torsional vibrations. This phenomenon is explained by the fact that, due to the decrease in the friction coefficient with increasing sliding speed, with an oscillatory change in sliding speed, the integral value of the friction force turns out to be higher than it should be at an average sliding speed. O. V. Izmerov, during tests on VNIIZhT ring railway track of a 2TE121 diesel locomotive, the drive of which is prone to development of self-oscillations of the wheelset, discovered that even with a slip of 10–14 %, the adhesion coefficient could reach a value of 0,25–0,27 (Pic. 13).

In this case, not random but intermittent skidding was observed, despite the absence of separate axle traction control on this locomotive. A. Golubenko in [9] also notes the fact that the operational coefficient of adhesion for the 2TE121 diesel locomotive when using sand is higher than that calculated according to the traction calculation standards for the 2TE10 and 2TE10L diesel locomotives, while the operational coefficient of adhesion for the 2TE10L and 2TE10V diesel locomotives was lower than the calculated standards.

During skidding in the traction mode, the most typical form of self-oscillations of the wheelset is when the wheels on one axle oscillate in antiphase. Conditions for development of such oscillations exist in drives that have elastic links in the shaft lines (support-axle drive with EGW, frame-support periods, centre-support drive). In this case, it is necessary to limit the amplitude of oscillations to avoid the occurrence of unacceptably high tangential stresses in the axle (by limiting the sliding speed or by means of vibration-impact damping of oscillations in the traction transmission).



Pic. 14. Diagram of pneumatic spring suspension on a unified jawless bogie: 1 – reservoir; 2 – throttle; 3 – spring; 4 – axle box; 5 – wheelset; 6 – pneumatic casing; 7 – axle box leash; 8 – pipeline; 9 – bogie frame; 10 – bracket; 11 – spring; 12 – height control valve [17].



Pic. 15. Rubber-metal supports of the body on the bogie: a) three-axle jawless bogie; b) TE33a diesel locomotive (USA); c) supports proposed by the authors of the article: 1 – support plates; 2 – rubber-metal element; 3 – end reinforcement plates; 4 – intermediate reinforcement plates; 5 – rubber layer [29].

Proposed Solutions

Modernisation of the three-axle bogie of the diesel locomotive

Currently, the 3TE25km diesel locomotives operated on the Eastern polygon mainly have three-axle jawless bogies similar to the bogies of the 2TE116 diesel locomotive, therefore, is advisable to consider the possibilities of increasing their traction properties.

The design of the bogie in question is sufficiently covered in domestic technical literature, therefore, in this article we will consider possible changes:

– use of balanced spring suspension;

– increase in rigidity of the body supports regarding the bogie;

– use of a frame-support drive.

The simplest way to implement a balanced spring suspension for the bogie in question is to use a pneumatic spring suspension in the axle box stage, previously used experimentally on the 2TE116–184 diesel locomotive [10; 18] (Pic. 14). According to [17], the static deflection of such a suspension is 150 mm, which is sufficient for diesel locomotives with a design speed of 100…120 km/h.

Experimental studies have established [9] that with pneumatic spring suspension, the amplitudes of dynamic loads, compared with the standard version of suspension on axle boxes, decreased by 1.5…2.5 times, which also reduces the dynamic unloading of the axle when passing vertical irregularities on the track.

The main problem that arises when using pneumatic suspension in the axle box stage is the need to limit the vertical movements of the axle boxes relative to the bogie frame in the absence of air in the air springs (for example, when moving the undercarriage along factory tracks). It is known that the dynamic movements of the axle boxes of a jawless bogie with individual spring suspension are 30…35 mm [4], and with a balanced single-stage suspension – 22…26 mm [4, 18], which is consistent with the above-mentioned phenomenon of a decrease in the amplitudes of dynamic loads. This allows us to expect that with pneumatic spring balanced suspension, the dynamic movements of the axle boxes relative to the bogie frame will be approximately 25 mm, which makes it possible to limit the vertical movements of the axle box to 35 mm, for example, by installing rubber stops on the axle boxes.

Increasing the static deflection of the axle box stage of the spring suspension allows increasing the rigidity of the body supports on the bogie. In combined body supports of domestic diesel locomotives (Pic. 15a), the support is a set of disk flat single-layer rubber-metal elements with a rubber layer thickness of 30 mm and metal plates 2 mm thick, as a result of which the metal reinforcement is 12,5 % of the package height. This design was adopted in the 60s of the last century to simplify the technological equipment in the manufacture of supports and the possibility of replacing individual damaged elements. In the body supports of US diesel locomotives (Pic. 15b), multilayer rubber-metal packages are used, which complicates the tooling, but allows for a reduction in the thickness of the intermediate reinforcement and ensures the same properties of the rubber in the support.

It should be noted that the issue of using multilayer elements in body supports instead of a set of single-layer ones was raised back in the mid-80s of the last century, however, since the purpose of the proposal was only to save metal, the idea was not implemented. In this case (Pic. 15c), the use of a multilayer element allows, while maintaining the same total thickness of the rubber layer (that is, with the same relative shear deformations when turning the bogie), to significantly reduce the distance between the plates, thereby increasing the rigidity of the support by several times and increasing the coefficient of use of the adhesion weight.

Increasing the rigidity of the body supports in combination with balanced air suspension allows increasing the coefficient of use of the adhesion weight η by approximately 11 % compared to the standard version of the bogie.

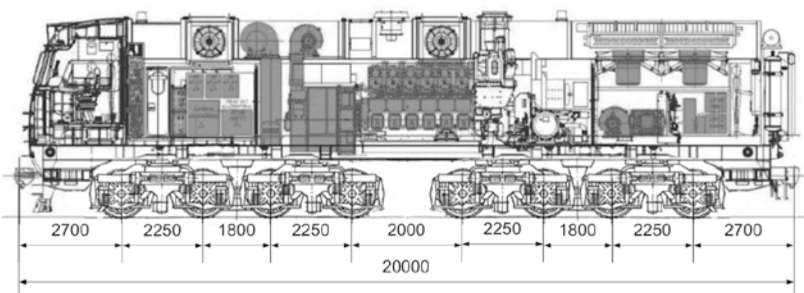
A further increase in this indicator is possible with the use of a frame-support traction drive. However, the use of a traction drive with a hollow shaft on the axle of a wheelset on a diesel locomotive with a wheel diameter of 1050 mm is associated with significant difficulties caused not only by the reduction of the gear transmission centre due to the need to reduce the diameter of the driven gear wheel located on the sprung mass (reducing the number of teeth from 75 to 72 with a module of 10), but mainly due to the placement of a hollow shaft between TEM housing and the axle of the wheelset, while the distance between the hollow shaft and the axle should be about 40 mm. As studies have shown [19], with a single-stage transmission, it is necessary to place an intermediate wheel. Another solution is to use a one-sided hinged-drive mechanism, which will require creation of multi-layer spherical rubber-metal hinges with a load capacity several times higher than existing ones. Now, we can only state the fundamental possibility of creating such a drive in the future and the feasibility of work in this direction, since the frame-support drive will not only allow us to slightly increase the value of η by approximately 4 %, but also reduce the unsprung mass.

Since in any case the disadvantage of a three-axle jawless bogie is a significant base compared to two-axle ones, in the opinion of the authors, for the 3TE25km series diesel locomotives currently operating on the Eastern polygon, it makes sense to modernise them by introducing a balanced pneumatic suspension and more rigid body supports, which does not require significant changes in the bogie design and maintains a high degree of its unification with the serial version. To implement the traction coefficient ψ = 0,3 and higher, it is necessary to create a new undercarriage part that takes into account the specifics of the Eastern polygon.

Unified bogie of diesel and electric locomotive

Currently, there are plans to create two locomotives for the Eastern polygon. Thus, JSC Sinara Transport Machines plans to create double unit 2TE35a diesel locomotives with an eight-axle bogie, having an upper intermediate frame, with a unit length of 24 m. In turn, JSC TMH plans to create a three-unit gas-diesel locomotive 3TE30g with a middle unit, which contains a reserve of liquefied gas or diesel fuel.

Both options have their advantages and disadvantages. The eight-axle bogie of the 2TE35a diesel locomotive allows for an increased adhesion coefficient on curves, but at the same time there is limited space for a fuel tank even with a total unit length of 24 m, which prevents the use of liquefied gas. In addition, the upper intermediate frame is heavy, and the announced



Pic. 16. An example of placement of an eight-axle bogie on the outer unit of a 3TE30g diesel locomotive, as proposed by the authors of the article.

use of TEM of the 2ES8 electric locomotive, which has a two-way rigid gear transmission, leads to a decrease in the adhesion coefficient when passing unevenness in the track, both due to the high unsprung mass (since it includes the rotating masses of the rotor) and due to dynamic loads in the drive. On the 3TE30g diesel locomotive with a unit length of 20 m, the use of a three-axle bogie simplifies and lightens the design, but leads to a deterioration in adhesion properties and an increase in wear of the wheel flanges; the use of an eight-axle undercarriage is unrealistic due to the impossibility in this case of placing a fuel tank under the body or increasing the length of the unit due to an unacceptable increase in the overall length of the locomotive.

Thus, the use of two-axle bogies on the 3TE30g diesel locomotive is hindered by the antagonism of properties: there must be a fuel tank and there must not be one. The antagonism can be removed by placing almost the entire fuel supply in the middle unit, leaving a small tank on the outer units for independent movement of the uncoupled unit along the depot tracks. In this case, it becomes possible to use eight-axle bogies with a lower frame on the outer units of the locomotive, similar to those used on the TE136 locomotive (Pic. 16).

In this case, the bogies of the middle unit can be made unmotorised. If two two-axle motorised bogies are used on the middle unit, then three two-axle bogies can be used on the outer units, similar to how it is done on the EP1 electric locomotive (Pic. 17), using inclined rods to transmit the traction force to the body (not shown in Pic. 15).

The above types of arrangement can also be implemented in the electric locomotive version. In addition, when arranging a diesel locomotive with outer eight-axle units and a motorised four-axle middle unit, the total power of the locomotive can be increased from 6 to 8,8 MW by using a domestic 12LDG500 diesel generator.

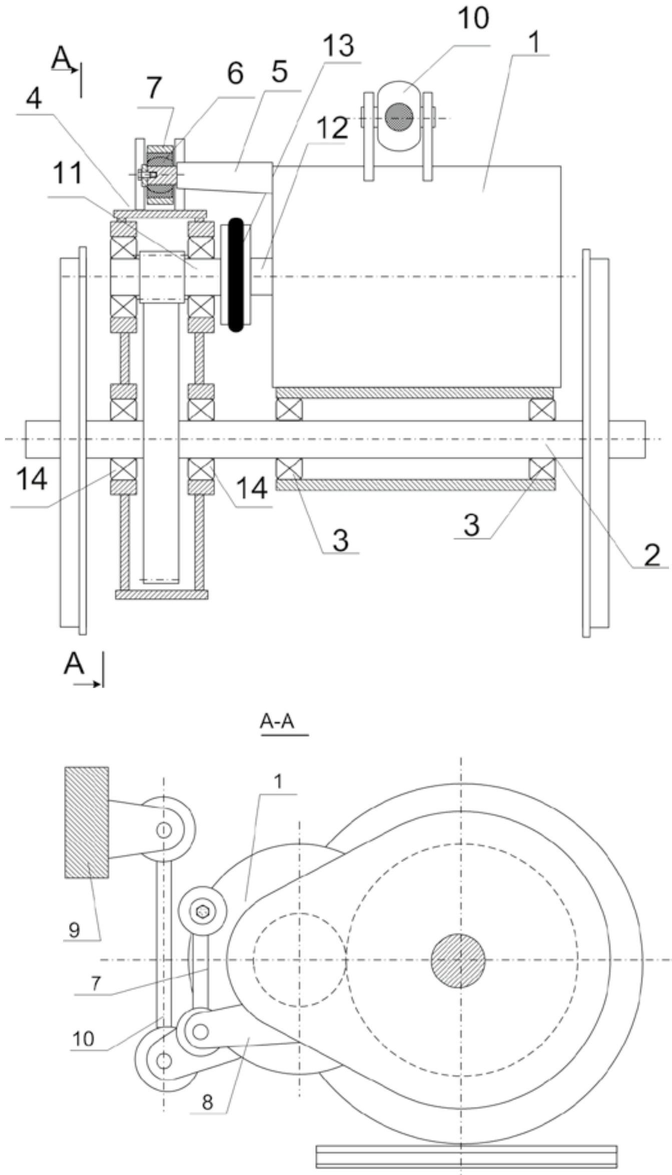
The use of inclined rods or a low-lying frame, compensating for redistribution of weight along the axles, allows free selection of the type of traction drive. Since the implementation of high traction forces requires the creation of new design elements for the transmission mechanism located on the low-speed part of the drive (for example, two-layer spherical rubber-metal hinges, previously described by the authors in [20]), and the terms of development and production of new locomotives are limited, two drive options are possible, which, on the one hand, do not require the creation of fundamentally new design elements, and on the other hand, ensure a reduction in the unsprung mass of the locomotive and the dynamic component of the traction force:

- drive with a frame-support suspension of TEM and an axial gearbox;
- drive with an axial-support suspension of TEM of an aggregate arrangement.

The use of a frame-support drive with an axial gearbox is advisable when using a traction electric motor with a maximum rotation speed below 3000 min^{-1}, since this allows the torsion shaft of the transmission mechanism to be placed inside the hollow rotor. With a continuous traction coefficient of ψ = 0,3 and an axial load of 245,25 kN, the drive must provide a traction

Pic. 17. A variant of the layout of a diesel locomotive with motorised axles of the middle unit and six-axle outer units, as proposed by the authors of the article.

a)



b)

Pic. 18. The aggregate arrangement of the axial-support drive proposed by the authors of the article: a) general view; b) view from the side of the axle gearbox; 1 – TEM; 2 – wheelset; 3, 14 – bearings; 4 – axle gearbox; 5 – bracket with pin; 6 – spherical joint; 7 – suspension; 8 – bracket of the axle gearbox; 9 – bogie frame; 10 – TEM suspension; 11 – TEM shaft; 12 – shaft of the axle gearbox; 13 – elastic-compensation coupling [28].

force on the wheel rim of F_∞ = 73,575 kN. To implement such a traction force, the gear ratio of the gearbox must be:

$$u = \frac{F_\infty D_w}{2M_\infty \eta_g}, \qquad (6)$$

where M_∞ = 10,4 kNm – torque of the TEM in continuous mode (by analogy with the TEM STA-1200U1 of the DS3 electric locomotive, which has a drive of this type), D_w = 1,25 m is the wheel diameter; η_g = 0,98 is the efficiency of the gear transmission with rolling bearings. For the specified parameters, u = 4,51. With a gear transmission module equal to 10 and a number of teeth on the driven gear equal to 95, the closest number of teeth on the small gear is 21, i.e. only 1 tooth less than in the axial gearbox on the 2TE121 diesel locomotive (with u = 4,52). Since the radial dimension of the STA1200U1 TEM in the plane of the traction transmission is about 800 mm, which is significantly less than the outer diameter of the ED126AUHL1 TEM (1036 mm), the possibility of placing the STA1200U1 TEM with this gear ratio can be considered proven.

The maximum rotation frequency of the TEM rotor with this gear ratio will be:

$$n_{max} = \frac{60 v_w u}{3{,}6 \pi D_w}, \quad (7)$$

where v_d = 120 km/h – design speed of the locomotive. With the above values, n_{max} = 2304 min^{-1}, which is significantly lower than the calculated maximum rotation frequency of STA1200U1 TEM (2304 min^{-1}), so it can be assumed that significant problems with the rotor bearings will not arise in the case under consideration.

When using TEM with a maximum rotation speed of about 3500 min^{-1}, it is advisable to use an axial-support drive of the aggregate type, the advantages of which, compared to the integrated drive used in foreign locomotives (Pic. 10c), were previously described in detail by the authors in [21]. According to [22], it is proposed to classify aggregate traction drives as traction drives that contain several units with the same type of support, maintaining operability separately from each other, and connected to each other by movable connections. Due to the mobility of the connection of the axle gearbox and TEM, it becomes possible to connect the shafts of the axle gearbox and TEM with an elastic-compensating coupling, while the misalignment of these shafts, unlike the frame-support drive with an axial gearbox, does not depend on vertical movements of the axle box, but is determined only by manufacturing errors of the parts. Minor shaft misalignment makes it possible in principle to use a single compensating coupling and increase its rotation frequency to values determined by the service life of the drive gear and input shaft bearings. In addition, the presence of an elastic connection between TEM rotor and the wheelset contributes to development of self-oscillations of the wheelset during slippage, which increases the adhesion coefficient.

The authors have developed and patented several design schemes for such drives (for example, [23–26]). One of the versions of such a drive proposed by the authors is shown in Pic. 18.

In the proposed drive, the axle gearbox is connected to TEM, which rests on the axle of the wheelset, by a hinged suspension, which does not allow the axle gearbox to move relative to TEM in the vertical direction, but this does not require such precision in the manufacture of the housing parts as in the integrated type drive described above. As can be seen from Pic. 17a, the aggregate arrangement leads to a slight increase in the axial dimensions of the drive to accommodate the elastic coupling, however, as previously proven by the authors in [27], for 1520 mm gauge locomotives this does not prevent the placement of a traction motor of the required power. The authors received a patent for a utility model for the drive shown in Pic. 17 [28].

Based on the fact that for the TE136 diesel locomotive, which used an eight-axle bogie with a frame-support drive and traction force transmission through a low-positioned intermediate frame, without the use of separate axle traction force control systems, the traction coefficient ψ was increased to 0,24 against 0,18 for the 2TE116 diesel locomotive, i.e., by 33 %, while equalising the axle loads theoretically made it possible to improve the traction properties in relation to the 2TE116 diesel locomotive by no more than 15 %, there is reason to believe that the proposed version of a unified bogie for freight diesel locomotives and electric locomotives using brushless TEM with separate axle control will make it possible to increase the traction coefficient in the design mode to values of 0,3…0,33.

CONCLUSIONS

The existence of a problem of rational selection of design solutions for the undercarriage of locomotives, ensuring the most complete use of adhesion properties in the conditions of the Eastern polygon, characterised by the presence of a significant number of curves with a radius of 350 m and less, was revealed.

Based on the analysis of factors affecting adhesion properties, an extended classification of methods for improving adhesion properties by improving the undercarriage was proposed.

Factors that can lead to deterioration of the adhesion properties of domestic locomotives in the conditions of the Eastern polygon were established. For freight diesel locomotives with standardised three-axle bogies, deterioration of adhesion properties is caused by a significant bogie base, insufficient rigidity of the rubber-metal body supports resting on the bogie with the upper location of the kingpin and unbalanced axle box suspension. The use of radial wheelset bogies on diesel locomotives only partially eliminates these shortcomings, since the axle box suspension remains unbalanced, and the low kingpin is less effective compared to the transmission of traction force to the body using an inclined traction rod, while the radial wheelset mechanisms complicate the design of the bogie, its maintenance and repair. For freight electric locomotives, a factor that worsens the traction properties is the use of a rigid transmission in the support-axle drive, which leads to the occurrence of total static and dynamic torques exceeding the adhesion limit when passing unevenness in the track.

It has been established that the undercarriage of foreign freight locomotives also cannot be considered completely rational in terms of maximising adhesion properties, primarily due to the rigid gear transmission. For the TE33a diesel locomotive developed by General Electric, the adhesion properties are also worsened by the significant base of the three-axle bogie and the use of jaw axle boxes.

As measures for modernisation of the unified three-axle bogie of freight diesel locomotives, it is proposed to use balanced pneumatic spring suspension in the axle box stage and increase the rigidity of the rubber-metal body supports resting on the bogie through the transition from a set of individual rubber-metal elements to a multilayer rubber-metal support with an increased number of intermediate plates.

It is proposed to create a unified undercarriage of promising diesel and electric locomotives for the Eastern polygon with a brushless drive in the diesel locomotive version to abandon the idea to place fuel tanks under the bodies of the outer units and place the main fuel supply in the middle four-axle unit. This will allow, with an outer unit length of 20 m, to use on the outer units either two four-axle bogies in the form of two-axle with a base of 2250 mm, united by a low-lying beam, or three two-axle with traction force transmitted to the body using an inclined traction rod. It is proposed to use structures with an elastic connection of TEM rotor and the wheelset as a traction drive, in the form of a frame-support drive with an axial gearbox and a traction motor with a maximum rotation frequency below 2500 min^{-1}, or a axial-support drive of an aggregate scheme with an elastic coupling. It is assumed that these measures will increase the operational coefficient of adhesion from 0,27 to 0,3 … 0,33, including due to development of self-oscillations of the wheelset in the skidding mode.

The authors received seven patents for utility models on the topic of the article.

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