

Maintenance of working capacity of movement mechanism of load trolley with linear traction electric drive of bridge type crane.

K A Goncharov, I A Denisov

Bryansk State Technical University, 7, 50 years October Boulevard, Bryansk, 241035, Russia

E-mail: ptm_bstu@mail.ru

Abstract. The article considers the influence of the air gap size between the linear motor elements on the stability of the traction drive of the movement mechanism of the trolley of the bridge type crane. The main factors affecting the air gap size and the causes of their occurrence are described. The technique of calculating the magnitude of air gap variation is described in relation to the general deformation of the crane metal structure. Recommendations on the need for installation of additional equipment for load trolleys of various designs are given. The optimal values of the length of the trolley base are proposed. Observance of these values ensures normal operation of the traction drive.

1. Introduction

The development of the construction of lifting machines is closely linked with the improvement of drive systems, which parameters have a direct effect on the technical characteristics of the crane, including the dynamic loading regime of the lifting machine metalwork elements. One of the promising directions for the development of mechanisms for the movement of bridge type cranes and trolleys is the creation of a drive based on a linear electric motor.

In the literature [10], the use of linear asynchronous electric motors for driving bridge cranes is mentioned. Such solutions were not widely used because of low energy indexes of linear electric motors due to the presence of edge effects. To date, this problem can be solved by using frequency converters, since it is proved that to obtain the most favorable operating mode of a linear motor in a reciprocating drive it is necessary to feed it from a variable frequency power supply [10].

The experience in the practical use of linear electric motors has shown that a simple replacement of existing drives with drives with linear motors is impossible or irrational, but the design of the entire production machine is required, taking into account the specific features of the work. When designing bridge type cranes with linear traction drive of a load trolley [3, 4] (Figure 1), it is necessary to take into account the type of the bridge's metal structure and its characteristics, the deformation of the metal structure and the elements of the trolley during operation, as well as the features of the dynamic mode of movement of the trolley with the load.



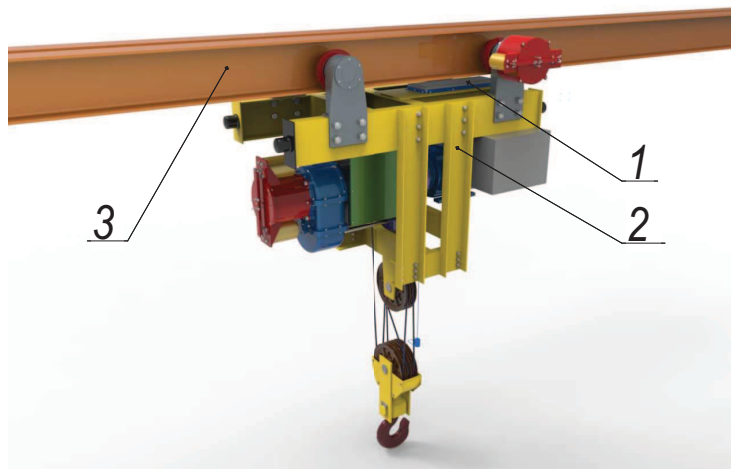


Figure 1. Load trolley of the general purpose bridge type crane with linear traction electric drive

2. Method description

Special attention should be paid to the issue of providing a constant air gap between the primary and secondary elements of the linear electric motor during the movement of the trolley under load. An increase in the gap by an amount Δs above its nominal value s can lead to a drop in the traction capability of the engine and a fluctuation in the effective value of the driving force F_D , which will adversely affect the trolley movement dynamics. The reduction of the gap, on the contrary, increases the probability of contact between the primary element 1 (Figure 1) installed on the load trolley 2 and the secondary element fixed to the metal structure of the bridge crane 3 and contributes to the output of the linear electric motor. Thus, the constancy of the air gap is one of the main conditions determining the operability of the linear drive of a bridge type crane trolley.

The change in the air gap Δs includes the following components:

- General and local deformation of bridge steel structures under static loads. The metal structures of the cargo trolleys possess considerable rigidity [2], therefore the influence of the deformation of the frame elements on the size of the air gap can be neglected.
- Oscillation of metal structures under the influence of dynamic loads during the unstable movement of crane mechanisms.
- Tilts caused by inaccuracies in the assembly and installation of a load trolley, wear of bearings, wheels and track. Such components can be compensated by the linear motor position control system and will not be taken into account when developing a primary method for estimating the working gap variation.

Let's consider the change of the working gap between the primary and secondary elements of the linear electric motor of the bridge type crane trolley during the deformation of the bridge metal structure. We take the following initial conditions and simplifications.

- Displacements of metal structure points as a result of local deformations are neglected.
- The trolley is in the middle of the span.
- The metal structure is deformed evenly along the entire span of the crane.
- It is assumed that the base of the trolley does not affect the nature of the metal structure deformation. Such an assumption with respect to the base of the trolley and the span of the crane $a/L \leq 0,2$ leads to an overestimation of the deflection by no more than 5% [9].

As it can be seen from the calculation scheme (Figure 2), the uniformly deformed metal structure of the crane 1 is an arc of a circle with a certain radius R . The length of the chord pulling the arc can be assumed with sufficient accuracy to be equal to the length of the entire span L , since the deflection of the crane bridge is due to plastic deformation of span metal structures. Accordingly, the deflection size is equal to the height of the segment formed by the chord AB and the arc of the circle and can be

determined by the formula:

$$f = R - \sqrt{R^2 - \frac{L^2}{4}}$$

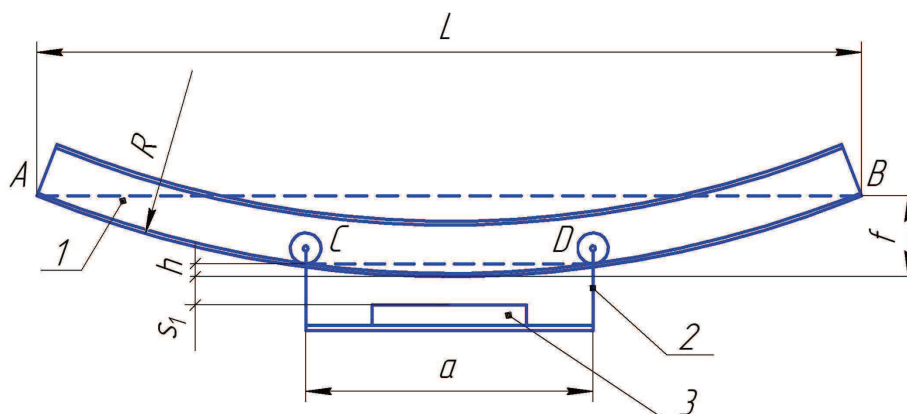


Figure 2. Scheme of deformation of the bridge crane metal structure

Consequently, the radius of the circle is:

$$R = \frac{4f^2 + L^2}{8f}$$

Similarly, the radius can be expressed from a segment that is formed by a portion of the same arc of a circle and a CD chord whose length is equal to the length of the base a of the load trolley 2 with the linear motor 3 installed. Moreover, the height h of the segment between the wheels of the trolley is the component of the working gap variation Δs of the total deformation of the crane metal structure.

Equating the obtained dependences and performing the necessary transformations, we get a quadratic equation, which desired root must also satisfy the condition:

$$h + s_1 < s,$$

where, s – a normative air gap between the primary and secondary elements of the linear motor; s_1 is the actual air gap.

The resulting system of equations has the form:

$$\begin{cases} 4h^2 f - h(L^2 + 4f^2) + a^2 f = 0; \\ h + s_1 < s. \end{cases}$$

Based on the described system of equations and inequalities, under the given air gap size s of the linear motor used, by setting the minimum possible distance s_1 between the primary and secondary elements necessary for the normal operation of the drive, it is possible to determine the rational value of the trolley base a for the known span of the crane L .

Similar relationships can be obtained for a bridge type crane with a support trolley. However, the suspension performance of the trolley is the most dangerous case (without the use of additional protective equipment, the probability of contact between the primary and secondary elements is high during the operation of the movement mechanism and under the influence of rapidly changing dynamic loads).

The total deflection of the metal structure f is the sum of the following components:

$$f = z_q + z_b + x,$$

where z_q – deflection of metal structures in the center of the span from the weight of the trolley and the load, mm; z_b – deflection of the metal structure from the weight of the beam, mm; x – is the amplitude of the oscillation of metal structures relative to its deformed state, mm.

For cranes with a span of more than 17 m, the value of z_q should be determined taking into account

the presence of a construction lift. The value of the construction lift for bridge type cranes is mainly governed by the conditions of the trolley movement and the requirement of compensation for the residual deflection. The metal structure of the bridge must be deformed in such a way that the slope of the trolley does not exceed the maximum permissible value $\Theta_{\max} = 0.002$, which is used to calculate the movement mechanism of the trolley [9]. Knowing the boom size of the construction hoist z_0 , the maximum deflection of the metal structure can be determined proceeding from the formula:

$$z_q \approx \frac{0,8z_0\pi - \Theta_{\max}L}{1,55(1 + \beta)}$$

The value of the coefficient β can be described by an approximate dependence [8]

$$\beta = \frac{3}{M} + 0,0002 \left(1 + \frac{20}{M} \right) L^2,$$

where M – crane lifting capacity, t; L – span of the crane, m.

To estimate the vibration amplitude x of the metal structure of a bridge type crane, it is rational to use the known three-mass dynamic model with two elastic connections [1, 5, 8] (Figure 3), which includes the following components: c_m – the rigidity of the crane metal structure, N / m; D_m – coefficient of damping of the metal structure of the crane, Ns / m; $c_k(S)$ – the rigidity of the lifting ropes, which depends on their tension S , N / m; D_k – load suspension damping coefficient, Ns / m; $P(\dot{x}_1)$ – the given driving force, depending on the speed of the rotor of the electric motor of the lifting mechanism, N; x_0, x_1, x_2 are the mass displacements m_0, m_1, m_2 respectively.

We will consider the most loaded mode of operation of a bridge type crane from possible that is lifting of a nominal monolithic cargo "with picking" from a rigid base with a central sling. The dynamical system has a variable structure: with a fixed load lying on the ground (a separating stage), and with a load moving during the ascent (post-breaking stage) [6, 7].

At the pre-breaking stage, the drive is accelerated, the gaps are selected in the elements of the lifting mechanism, and the tension of the load rope [5, 8]. The dynamics of the system under study at the pre-breaking stage is described by the following dependences:

$$\begin{cases} m_0\ddot{x}_0 + c_m\dot{x}_0 + D_m\dot{x}_0 - c_k(S)(x_1 - x_0) - D_k(\dot{x}_1 - \dot{x}_0) = 0; \\ m_1\ddot{x}_1 - P(\dot{x}_1) + c_k(S)(x_1 - x_0) + D_k(\dot{x}_1 - \dot{x}_0) = 0. \end{cases}$$

The initial conditions for solving the system of equations

$$x_0(0) = 0, \dot{x}_0(0) = 0, x_1(0) = 0, \dot{x}_1(0) = v_0,$$

where v_0 – reduced peripheral speed of the motor at idle speed.

The separation of the load from the base occurs at the moment when the tension force in the ropes S becomes equal to the weight of the lifted load Q . Studies have shown [8] that an increase in the rigidity of the rope with a decrease in its length practically does not affect the value of the dynamic loads due to the presence of considerable dispartive forces in the lifting mechanism, therefore the rigidity of the rope can be assumed constant when lifting the load. The system of equations that describes the post-breaking stage is as follows:

$$\begin{cases} m_0\ddot{x}_0 + c_m\dot{x}_0 + D_m\dot{x}_0 - c_k(x_1 - x_0 - x_2) - D_k(\dot{x}_1 - \dot{x}_0 - \dot{x}_2) = 0; \\ m_1\ddot{x}_1 - P(\dot{x}_1) + c_k(x_1 - x_0 - x_2) + D_k(\dot{x}_1 - \dot{x}_0 - \dot{x}_2) = 0; \\ m_2\ddot{x}_2 + m_2g - c_k(x_1 - x_0 - x_2) - D_k(\dot{x}_1 - \dot{x}_0 - \dot{x}_2) = 0. \end{cases}$$

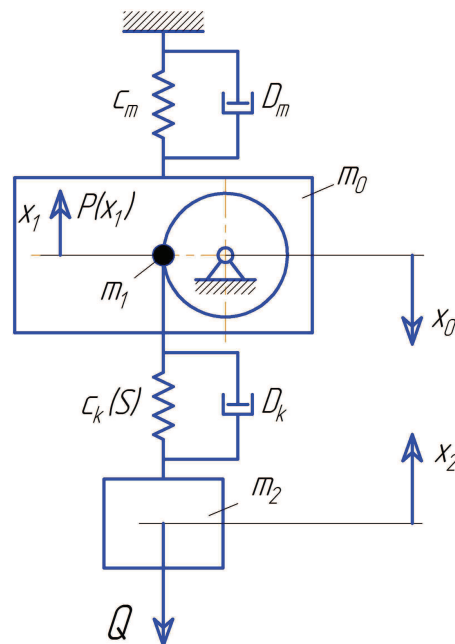


Figure 3. Dynamic scheme of bridge type crane

The initial conditions for solving the system of equations:

$$x_0(0) = x_0(t_{sep}), \quad x_1(0) = x_1(t_{sep}), \quad x_2(0) = 0,$$

$$\dot{x}_0(0) = \dot{x}_0(t_{sep}), \quad \dot{x}_1(0) = \dot{x}_1(t_{sep}), \quad \dot{x}_2(0) = 0.$$

It should be noted that when the dynamic index is varied in the altitudes $D_m = 500 \dots 1000 \text{ Ns/m}$, $D_k = 6000 \dots 10000 \text{ Ns/m}$, the damping factor leads to a decrease in the dynamic coefficients by no more than 2%. Therefore, in some cases, in order to simplify the model, the effect of damping the structure on the maximum amplitude of oscillations of the span structure of the bridge type crane can be neglected [1].

The solution of each of the stages is performed separately in numerical form. In Fig. 4 there is a graph of the movement of the truss metal structure of a bridge type crane with a payload of 15 tons, with a span of 10 meters [3]. One can see that the structure undergoes maximum deformations at the time of the first oscillation with an amplitude of $x = 1.16 \text{ mm}$, after which damping occurs due to the damping properties of the metal structure, suspension and electric motor.

3. Conclusion

The presented dependences do not fully reflect the nature of the metal structure deformation and its connections, but allow in the first approximation to estimate the relationship between individual dimensional parameters and to conclude that the linear traction drive on the bridge type cranes is operational. So in the case of using one type of trolley on bridge cranes with different lengths of span, the deflection of the metal structure f with the corresponding ratio $L / 400 \dots 600$ and the construction rise $z_0 = L / 800 \dots 1000$, the component of the total deformation h will be larger for the crane with a smaller span, so as the wheels of the trolley cut off the arc of the circle with greater curvature.

The calculation of the air gap variation for a number of lifting machines of standard sizes with a trolley span $a = 0.2L$ has shown that at the maximum possible strain rates h is in the range $0 \dots 0.42 \text{ mm}$.

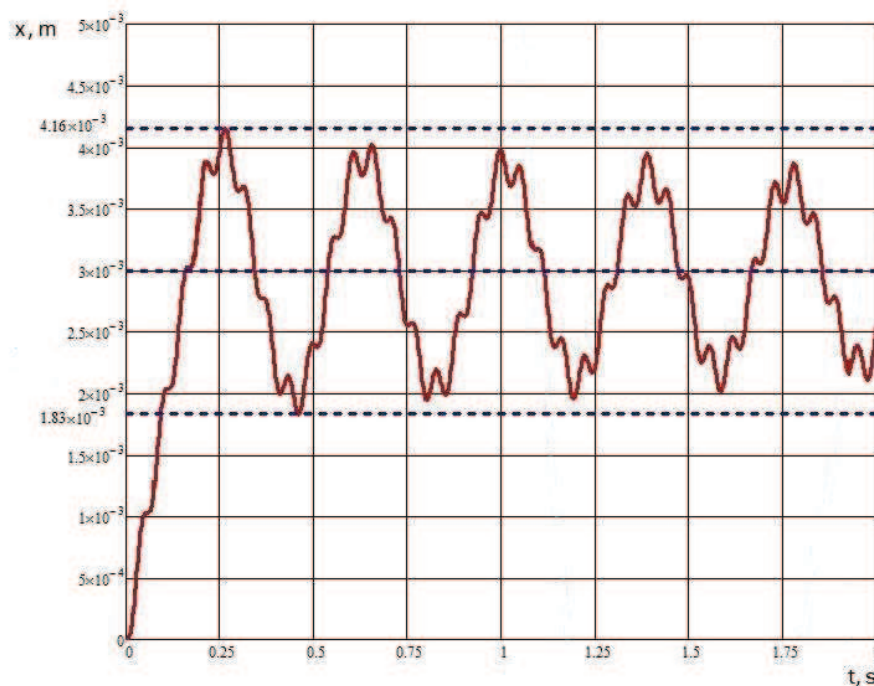


Figure 4. The graph of the relocated steel structure of the bridge type crane when lifting the nominal load

The obtained data confirm the possibility of using a linear traction electric drive for load trolleys of bridge type cranes. The existing regulatory requirements for the rigidity of metal structures in most cases ensure the normal operation of a linear electric motor with an air gap of 3 ... 5 mm. On load trolleys of suspension design, it is additionally necessary to install safety rollers preventing accidental contact between the primary and secondary elements. The trolley base providing the value $h \leq 0.2$ mm is the most rational, since the traction characteristics of linear electric motors in this case change minimally.

References

- [1] Budikov L Y 2003 *Multiparametric analysis of the dynamics of load-lifting cranes of bridge type* (Lugansk: publishing house of the VNU. After V.Dal)
- [2] Denisov I A and Prostavkov D A 2016 *Innovative development of hoisting and transport equipment: materials of all-Russian. scientific-practical. conf.* ed K A Goncharov (Bryansk: BSTU) pp 36-42
- [3] Denisov I A 2015 *Innovative development of hoisting and transport equipment: materials of all-Russian. scientific-practical. conf.* ed K A Goncharov (Bryansk: BSTU) pp 15-19
- [4] Dunaev V P, Kuleshov D Yu, Goncharov K A and Denisov I A *Tal elektricheskaya kanatnaya* [Electric hoist rope]. Patent RF, no. 159563, 2016
- [5] Lagerev I A 2015 *Scientific and Technical Herald of the Bryansk State University* **1(1)** 59-54
- [6] Lagerev A V, Lagerev I A and Milto A A 2015 *International review on modelling and simulations* **8(2)** 223-226
- [7] Lagerev A V, Lagerev I A and Milto A A 2014 *International review on modelling and simulations* **7(4)** 644-652
- [8] Lobov N A 1987 *Dynamics of load-lifting cranes* (Moscow: Mechanical Engineering)
- [9] Sokolov S A 2005 *Metal constructions of hoisting-and-transport machines* (Saint-Petersburg: Politechnica)
- [10] Sokolov M M 1974 *Electric drive with linear asynchronous motors* (Moscow: publishing house "Energy")