Safety ensuring of the rail lift and transport facilities at the track ends

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Abstract. The dead-end stops that prevent the descent and destruction of rail lift and transport facilities are used to ensure their safety. They are shock type, shockless type and combined. The most promising are the dead-end stops of unstressed type, since the main task of the dead-end stop is to stop the rail type device at the minimum distance. The probability of the crane overturning in the longitudinal or transverse planes of the track will depend on the "smoothness", "softness" of the braking. To improve the "smoothness" and consequently the braking safety, the dead-end stop design with the self-reinforcement effect of the resistance forces work is proposed and that can be used both for lifting and transport, and for main rail systems. Analytical dependence for the design calculation of the dead-end stop according to its main functional characteristic was obtained. The calculated dependency includes all the basic parameters allowing to determine its dimensions and configurations of the power elements and is the basis for the development of a diagrammatic layout, a general view drawing and a subsequent refined calculation of the dead-end stop design.

Introduction

The work safety and normal operating conditions for rail cranes of such types as bridge, gantry, tower, cranes-loaders, portal and suspended are largely dependent on the of crane runways geometry. Due to the impossibility of laying the crane runway dead horizontally when the crane passes the inclined part of the track and turns the crane arm the with the load, the uneven pressure distribution on the running wheels occurs. Crane way is unevenly elastic deformed (subsided) due to the pressure of the running wheel. In addition, the stability of freestanding cranes is influenced by the inelastic base subsidence, which occurs because of uneven ground compaction, washed away by surface waters, and uneven thawing in spring.

Thus, the unilateral cumulative effect of the above factors creates an additional crane imbalance, which often leads to emergencies. Dead-end stops preventing the descent and destruction of rail truck lift and transport facilities at the truck ends are used to ensure their safety [1-5].

Dead-end stop is a mechanical safety device designed to dampen the residual crane speed and prevent it from leaving the track ends of the crane way in emergencies. They are shock type, shockless type and combined.
Dead-end stops of shock type (with wooden, rubber, spring, spring-friction, hydraulic and combined buffers) are the stops where the crane stops due to the absorption of kinetic energy by elastic elements installed on the crane and the stop. Buffer is a device to soften the blow.

Dead-end stops of shockless (rolling) type (gravity and friction-gravity) are the stops where the crane stops at the expense of absorbing kinetic energy when overcoming the stop roller.

Dead-end stops of combined type include the stops elements of shock and shockless types. The most promising are the dead-end stops of shockless type [6-8]. This is explained by the fact that the main task of the dead-end stop is to stop the rail device at the minimum distance. The probability of the crane overturning in the longitudinal or transverse planes of track will depend on the "smoothness" of braking. To improve the "smoothness" and consequently the braking safety, the dead-end stop design (DES) with the self-enhancing effect of the resistance forces work is proposed [9].

The mode and design description

The proposed design (figure) consists of a block 1, mounted on a rail 2 on both sides of which are the beads 3, movably connected to the sliders 4 of the braking system, moving in the guides 5, while the cone 6 is movably fixed on the slide, coupled with a split cylinder 7, the outer surface is in contact with a fixedly mounted body 8, and the end surface with a spring 9 resting on the body base, in turn, the slider rests on the springs 10, 11 located one inside the other by means of a support washer 12, moreover, the independent stroke $\delta_1$ of the springs 10 and the joint stroke $\delta_2$ of springs 10 and 11 are equal to the slide $\delta_3$ motion and correspond to the joint motion of the springs 9, 10, 11 and the chock. In the complete DES, set includes two chocks, one on each rail, and four brake systems, mounted on the web of each rail on both sides.

The system works as follows. In the first stage, after chock 1 is under the wheel 13, the wheel movement 13 starts the chock 1 and rod 4 as a single system leading to the spring compression 10 by $\delta$. When this occurs, friction forces on the wheel contact surfaces with radius R with a chock and a chock with a rail that enhance the braking power by the force created by the spring 10.

At the second movement stage, in section $\delta_2$ the springs 10 and 11 begin to contract, together providing a further increase in braking power.

At the third stage, in section $\delta_3$ the slider with a limiter 14 abuts against the cone 6, the split cylinder 7 moves apart, and friction forces arise in contact between the body and the outer surface of the cylinder.

Further, the cylinder moves simultaneously with springs compress 9, 10, 11 and they lead to a maximum increase in braking power. After the decelerated object leaves the chock springs return it to its original position. When this spring 10 removes the cone from the contact eliminating the effect of self-jamming, and spring 9 returns to its original position, the cylinder pressing it to the fixedly fixed cover 15 of the body 8.

Results

To develop the proposed design of dead-end stop, which is a new system, it is necessary to create formulas for the design calculation, allowing connecting the main functional parameters and structural elements determining these parameters [9-14]. The difficulty of compiling such formulas lies in the information lack about the object at the initial design stages, especially if it is a new design that has fundamental differences from the analog and the prototype that is our case.

When solving the task, we will draw on the fact that the projected dead-end stop refers to planar mechanisms, with leading element is the object to brake, and all the links make translational motion.
Since, in accordance with the functional purpose of the stop, the kinetic energy of the object to brake at the end of the working stroke must be zero, then to compile the design calculation formula, the motion equation of the following form can be used

\[ \frac{mV^2}{2} - \sum W_c = 0 \]  

(1)

where \( m \), \( V \) is the mass and speed of the object to brake, \( W_c \) is the total work of the resistance forces created by the spot.

The stop is designed for both lift and transport, and for main rail systems, which mass may be in the range (2000 ... 12000) kg. It is obvious that the total mass of the elements of the stop itself will be one to two orders of magnitude smaller than the mass of the object to brake, and it can be neglected in the calculation.

Since the process of the DES elements movement is carried out relative to the axis \( OX \), it is advisable to determine the work of resistance forces relative to this axis.

If \( L_{AB} \) is the inclined face length of the chock that to simplify is considered as a chord and not as an arc \( AB \), \( \Sigma L_{AB} \) [9] is the total trajectory of the wheel on the face, \( F_{fr1} \) and is the friction force between the wheel rolling surface and the face, then the work \( W_1 \) done by the wheel relative to the inclined face will be equal to:

\[ W_1 = F_{fr1} \cdot \sum L_{AB}, \]  

(2)

and the \( L_{AB} \) projection on the horizontal that is expressed by the inclination angle of the face respectively

\[ L_{AB}^x = L_{AB} \cdot \cos \alpha, \]  

(3)

but according to the working condition, the total trajectory \( \Sigma L_{AB}^x \) is equal to the working stroke \( \delta \) and, therefore

\[ \delta = \Sigma L_{AB}^x = \Sigma L_{AB} \cdot \cos \alpha. \]  

(4)

Horizontal \( F_{fr1} \) projection:

\[ F_{fr1}^x = F_{fr1} \cdot \cos \alpha, \]  

(5)

wherein:
\[ F_{fr1} = F_1 \cdot f_1, \]  
\[ F_1 = \frac{F}{\sin \alpha}, \]  
where \( F_1, f_1 \) is the normal force and the friction coefficient between the inclined 16 chock face and the rail.

Express by the total compression force the springs \( F_s \)

\[ W_1 = F_s \cdot f_1 \cdot \operatorname{ctg} \alpha \cdot \delta. \]

Further we define the \( W_2 \) work performed by the chock along the rail in section 2:

\[ W_2 = F_{fr2} \cdot \delta = F_2 \cdot f_2 \cdot \delta, \]

where \( F_2, f_2 \) is the normal force and the friction coefficient between the inclined 17 chock face and the rail.

Expressing \( F_2 \) by \( F_s \):

\[ W_2 = F_s \cdot f_s \cdot \operatorname{ctg} \alpha \cdot \delta. \]

The \( W_3 \) work performed by the friction force \( F_{fr3} \) when moving the cone relative to the split cylinder will occur within the technological gaps and its value can be neglected \( W_3 = 0 \).

The \( W_4 \) work in the area \( \delta_4 = \delta_1 \) from the friction force \( F_{fr4} \) arising between the body and the elements of the split sleeve, which is determined by the friction coefficient \( f_4 \) and the normal force \( F_4 \), created by the compression force \( F_s \) of the spring 9 is equal to:

\[ W_4 = F_{fr4} \cdot \delta_4 = F_4 \cdot f_4 \cdot \delta_4 = F_s \cdot \operatorname{tg} \gamma \cdot f_4 \cdot \delta_4. \]

Further, we will present the work of the resistance forces along the axis OX in general, as. To do this, let us describe the operation of the brake chocks, if their number is equal to "K":

\[ \sum F_{fr1} = \sum F_1 \cdot f_1, \]

then the brake systems work, if the number is equal to \( \Delta \):

\[ \sum F_{fr1} = \sum F_1 \cdot f_1 \cdot \operatorname{ctg} \alpha \cdot \delta. \]

and finally, we get the total work of the resistance forces in general:

\[ \sum W_s = \sum (W_1 + W_2) \cdot K + \sum (W_4 + W_5) \cdot \Delta \]

Since in our case we are talking about the design calculation for simplicity we introduce the average stiffness index \( c \) and the length of the working sections will be the same \( \delta_1 = \delta_2 = \delta_3 \), i.e. \( \delta = 3 \delta_0 \). After that, we determine the total force \( F_s \) of the spring compression:
\[ F_s = F_{s1} + F_{s2} + F_{s3} = c\cdot(\delta_1 + \delta_2 + \delta_3) + c\cdot(\delta_1 + \delta_2) + c\cdot\delta_3 = \]
\[ = 3\cdot\delta_0 \cdot c + 2\cdot\delta_0 \cdot c + \delta_0 \cdot c = 6\cdot\delta_0 \cdot c = 2\cdot\delta \cdot c \]

(17)

and we make up the equations of the work \( W_1, W_2, W_4, W_5 \), taking into account (16, 17) and the adopted restrictions:

\[ W_1 = 2\cdot\delta^2 \cdot c \cdot f_1 \cdot ctg \alpha; \]
\[ W_2 = 2\cdot\delta^2 \cdot c \cdot f_2 \cdot ctg \alpha; \]
\[ W_4 = \frac{1}{9} \delta^2 \cdot c \cdot f_4 \cdot tg \gamma; \]
\[ W_5 = 2\cdot\delta^2 \cdot c. \]

Substituting (16) into (1) with (18) and then transform

\[ \frac{mV^2}{2} - c\cdot\delta^2 \left[ 2K \cdot ctg \alpha (f_1 + f_2) + \left( \frac{1}{9} f_4 \cdot tg \gamma + 2\right) \Delta \right] = 0. \]

(19)

Solving (19) with respect to the velocity \( V \) of the object to brake, which can be canceled to zero by DES, depending on the object mass and its design characteristics:

\[ V = \delta \sqrt{\frac{2c}{m}} \left[ \frac{2K \cdot ctg \alpha (f_1 + f_2) + \left( \frac{1}{9} f_4 \cdot tg \gamma + 2\right) \Delta}{\delta^2} \right] \]

(20)

As a result, we obtain the analytical dependence for the design calculation of a dead-end stop according to its main functional characteristic.

Summary

Obtained calculation formula includes all the main parameters characterizing the operating conditions of the dead-end stop, allowing to determine its dimensions and configuration of the power elements depending on the object mass to brake, its speed, etc.

The resulting dependence is the basis for the development of the diagrammatic layout, general view drawing and subsequent refined calculation of the dead-end stop design for specific operating conditions.

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