Analysis of power characteristics in the elements of the active Dead-End stop based on the double wedge system

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Abstract. The dead-end stops that are installed at the truck ends and differ from each other in their design and principle of repayment of the kinetic energy of a moving object are used to improve the operation security of the rail lift and transport facilities. However, the main task of any dead-end stop is to stop the lift and transport facility at the minimum braking distance. Inadequate braking “smoothness” can cause a derailment or destruction of the lift and transport facility, so it is preferable to use dead-end stops excluding the impact during the braking and ensuring a smooth increase of the braking effect. The dead-end stop construction with the effect of increasing resistance forces is proposed, which meets these conditions, that can be used both for lift and transport facilities and for the rail rolling stock. In addition, the analysis of power characteristics in the spot elements was made and analytical dependencies for engineering calculations were developed.

Introduction

During the operation of lift and transport facilities or the rail rolling stock, the situations may arise when the movement of such objects may lead to their derailment or destruction. Uncontrolled movement due to the failure of own braking devices, or erroneous actions of the service personnel controlling the movement of the lift and transport facility or rolling stock may cause this. Dead-end stops preventing the descent and destruction are used to ensure the safety of the transport and lift and transport facilities at the truck ends [1-6].

The dead-end stop is a mechanical safety device designed to dampen the residual velocity of the rail-moving object and prevent it from leaving the track ends in emergency. The dead-end stops are of shock type, shockless type and combined according to the kinetic energy dumping.

When using dead-end stops of shock type, the moving object is stopped by absorbing its kinetic energy by elastic elements mounted on the arrest stop, and sometimes also on the moving object itself (crane or car). The braking process in this case is carried out on a very short track segment and is accompanied by a blow. Elastic elements in such arrest stops (rubber, spring, spring-friction, etc.) play the role of a buffer, softening the stroke.

Dead-end stops of shockless type are used to stop rail lift and transport facilities, due to the rolling wheels on the inclined face of the arrest stop, resulting in the absorption of the kinetic energy of a moving rail lift and transport facility. The advantage of such arrest stops is the absence of a stroke...
during the braking along with a sufficiently small stopping distance. However, when interacting with such an arrest stop, the lift and transport facility tilts that may be dangerous in some cases and makes such arrest stops inapplicable to railway rolling stock because of the derailing danger.

Dead-end stops of combined type include the arrest stops elements of shock and shockless types.

The use of dead-end stops of shockless type [7–9], but with a structure without discussed earlier disadvantages is most preferred. The dead-end stop must stop an object moving along the rails at the shortest possible distance, and at the same time, the braking “smoothness” must be provided to prevent the derailing and damage of a lift and transport facility or a wagon. Active dead-end stop design of power ADS based on the double wedge system is suggested to solve this problem.

The mode and design description

The proposed design (figure) consists of a chock 1 mounted on a rail 4 and having beads 7 on both sides movably connected to the sliders 6. Each slide moves in guides 5 fixed to the rail web and enters the fixedly mounted body 9, where the pusher rests against the spring 10.

Complete dead-end stop set includes two chocks, (one on each rail), and four cylindrical bodies with springs fixed on both sides of each rail.

The system works as follows. When a wheel 3 hits a chock 1, their movement begins as a unified system. At the same time, friction forces arise on the inclined 2 and supporting 8 facets. As the chock moves and the sliders 6 associated with it compress the springs 10 that enhances the braking effect. The wheel and chock movement is limited by the stroke of the sliders until the springs are fully compressed. After the wheel leaves, the chock springs return it to its original position.

![Kinematic scheme of the dead-end stop](image)

**Figure 1.** Kinematic scheme of the dead-end stop

Results

To develop the proposed design of an active dead-end stop, it is necessary to perform an analysis of the power characteristics in its elements and to develop analytical dependencies for engineering calculations, allowing to connect the main functional parameters of the arrest stop and structural elements defining them [10-16]. To solve the problem, some simplifications are accepted. In particular, we will assume that the projected dead-end stop belongs to flat mechanisms, where the leading element is the object to brake and all the links make a translational movement. The proposed dead-end stop is designed for lift and transport facilities or the rail rolling stock, therefore the total mass of the moving elements of the arrest stop itself will be incomparably smaller than the mass of the object to brake, that allows neglecting the weight of the arrest stop elements in the calculation.
According to the functional purpose of the ADS, it must stop the object to brake, i.e. the speed of the object to brake should be zero. This condition will be satisfied in the case when the driving force \( F \) acting on the arrest stop from the wheel side will be balanced by the sum of the resistance forces arising from the interaction of the arrest stop with the wheel. Based on the ADS kinematic scheme (figure), we make the vector equilibrium equation of forces arising when the wheel contacts the arrest stop, as the difference between the driving \( F \) and the sum of resistance forces \( \sum_{i=1}^{n} F_c \), which number varies from \( i = 1 \) to \( i = n \), i.e.

\[
F - \sum_{i=1}^{n} F_c = 0
\]  

(1)

From the analysis of the kinematic scheme, it follows that the following resistance forces will arise in the system. In the place of contact of the wheel rolling surface 3 and the inclined chock facet 2 of the chock 1, the sliding friction force \( F_{fr1} \) will act. Since at the same time the chock will move along the rail, the sliding friction force \( F_{fr2} \) will act in the place of contact the rail surface 4 and the supporting chock facet 8. The wheel friction with respect to the rail surface will be prevented by the friction force \( F_{fr3} \) arising in the contact patch of the wheel with the rail. In addition, finally, another resistance force \( F_s \) will be created by the elastic element 10, which slider 6 contacts. Thus, the number of resistance forces in equation (1) is \( i = 4 \) and, therefore:

\[
\sum_{i=1}^{4} F_c = F_{fr1} + F_{fr2} + F_{fr3} + F_s
\]  

(2)

Substituting (2) into (1) we obtain:

\[
F - \left( F_{fr1} + F_{fr2} + F_{fr3} + F_s \right) = 0
\]  

(3)

Determine the value of the forces included in equation (3). Since the driving force \( F \) and the links displacement of the ADS mechanism are directed along the axis Ox (figure), it is advisable to consider the forces interaction relative to this axis.

The driving force \( F \) can be expressed in terms of the mass \( m \) and the movement acceleration \( a \) of the lift and transport facility

\[
F = m \cdot a
\]  

(4)

The friction force \( F_{fr1} \) with the friction coefficient \( f_1 \) and the normal force \( F_1 \) in the place of contact of the wheel with the chock will be equal to:

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

(5)

but the force \( F_1 \) normal with respect to the chock, will be the component of the driving force \( F \). Express \( F_1 \) using \( F \) and the sine of the angle \( \alpha \):

\[
F_1 = F \cdot \sin \alpha.
\]  

(6)

We write (5) with (6):

\[
F_{fr1} = f_1 \cdot F \cdot \sin \alpha = f_1 \cdot m \cdot a \cdot \sin \alpha.
\]  

(7)

The force \( F_{fr2} \) can be determined through the projections sum on the axis Oy of two forces:

\[
F_{fr2} = \left( F_{fr2}^v + F_{fr2}^\nu \right) f_2.
\]  

(8)

Moreover \( F_{fr2}^v = F_1 \cdot \cos \alpha \), \( F_{fr2}^\nu = F_{fr1} \cdot \sin \alpha \). Then

\[
F_{fr2} = f_2 \left( F_1 \cdot \cos \alpha + F_{fr1} \cdot \sin \alpha \right).
\]  

(9)

We write (9) with (6) and (7):

\[
F_{fr2} = f_2 \cdot m \cdot a \cdot \sin \alpha \left( \cos \alpha + f_1 \cdot \sin \alpha \right).
\]  

(10)

The strength \( F_{fr3} \) is determined by the ratio:
\[ F_{n3} = f_3 \cdot P = f_3 \cdot m \cdot g, \]  
(11)

Spring force:
\[ F_s = cx, \]  
(12)

where \( c \) is the spring stiffness, N / mm; 
\( x \) is the spring deformation, mm.

Substitute the obtained values into equation (3):
\[ F - f_1 \cdot F \cdot \sin \alpha - f_2 \cdot F \cdot \sin \alpha (\cos \alpha + f_1 \cdot \sin \alpha) + f_3 \cdot m \cdot g - cx = 0, \]  
(13)

or:
\[ m \cdot a \left(1 - f_1 \cdot \sin \alpha - f_2 \cdot \sin \alpha \cdot \cos \alpha + f_1 \cdot f_2 \cdot \sin^2 \alpha \right) + f_3 \cdot m \cdot g - cx = 0. \]  
(14)

Given that:
\[ f_2 \cdot \sin \alpha \cdot \cos \alpha = \frac{c}{2} \cdot f_2 \cdot \sin 2 \alpha, \]  
(15)

we get:
\[ m \cdot a \left(1 - f_1 \cdot \sin \alpha - \frac{f_2}{2} \cdot \sin 2 \alpha - f_1 \cdot f_2 \cdot \sin^2 \alpha \right) + f_3 \cdot m \cdot g = cx. \]  
(16)

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

Option 1. To determine the braking mode acceleration, if the given mass \( m \), angle \( \alpha \), coefficients \( f_1, f_2, f_3 \), spring parameters \( c \). To do this, we transform the equation (16). Given that the spring travel \( x \) is equal to the braking distance \( L \), we get:
\[ a = \frac{c \cdot L - f_3 \cdot m \cdot g}{m \left(1 - f_1 \cdot \sin \alpha - \frac{f_2}{2} \cdot \sin 2 \alpha - f_1 \cdot f_2 \cdot \sin^2 \alpha \right)}. \]  
(17)

Option 2. To determine the allowable weight of the lift and transport facility for which the arrest stop is calculated, if \( a \) acceleration, \( \alpha \) angle, \( f_1, f_2, f_3 \) coefficients, \( c \) spring parameters are specified. In this case, as a result of the expression transformations (16), we obtain:
\[ m = \frac{c \cdot L}{a \left(1 - f_1 \cdot \sin \alpha - \frac{f_2}{2} \cdot \sin 2 \alpha - f_1 \cdot f_2 \cdot \sin^2 \alpha \right) f_3 \cdot g}. \]  
(18)

Option 3. To determine the stiffness of the elastic element, if \( m \) mass, \( a \) acceleration, \( \alpha \) angle, \( f_1, f_2, f_3 \) coefficients are known. Having solved (16) relative to \( c \), we get:
\[ c = \frac{ma \left(1 - f_1 \cdot \sin \alpha - \frac{f_2}{2} \cdot \sin 2 \alpha - f_1 \cdot f_2 \cdot \sin^2 \alpha \right) + f_3 \cdot m \cdot g}{L}. \]  
(19)

Summary
The resulting calculation formula combines the basic parameters of the structural elements of the active dead-end stop and allows associating them with operation conditions, being the basis for the subsequent revised calculation, which will take into account to the fullest extent the functioning peculiarities of the dead-end stop mechanism and its interaction with a specific object to brake.

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