REDUNDANT CONSTRAINTS IN CRANE DISC BRAKE MECHANISM: EFFECT AND DISPOSAL PERSPECTIVES

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Resume

In this paper is considered a promising design of the braking disk mechanism of a construction crane. The analysis of the kinematics of the brake pad drive has shown that due to redundant links in the brake pad drive mechanism the latter are not able to self-align. This leads to uneven braking torque during the smooth braking, overloads and deterioration of the positioning accuracy of the load suspended on the cable. Based on the methods of analyzing the kinematics of complex mechanisms, two variants of modernization without redundant links are proposed. Both variants can be realized without changing the kinematics of the mechanism, but they differ in the complexity of the joint manufacturing technology and depend on the design and dimensional limitations of the brake.

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1 Introduction

Lifting machine brake mechanisms are designed to reduce the kinetic energy of their mechanisms during changes in movement modes and holding loads or crane elements and are responsible elements on which the safety of operation and the lives of personnel depend. Therefore, increasing the reliability of brakes is a reserve for improving the safety of the operation of lifting machines.

Nowadays, the drum-type brakes are widespread in cranes, the characteristic drawback of which is overheating, which determines a certain amount of modern scientific research in the field of their dynamics and materials for manufacturing their elements [1-2]. However, disc-type mechanisms, which cool better and provide high values of braking forces, are more promising. The disc brake elements are highly stressed, which requires ensuring an even distribution of forces between their elements during operation in the presence of manufacturing errors, wear, or thermal grooving.

The presence of redundant constraints in the mechanisms contributes to the increase in the unevenness of the load distribution between the parts [3-4]. This can lead to the destruction of parts [5], increased requirements for manufacturing and assembly accuracy [6], and an increase in mechanical losses [7]. There are not many works dedicated to the study of the structure and elimination of redundant constraints in crane brake mechanisms. For example, the lever mechanisms of crane-type drum-pad brakes with electrohydraulic and electromagnetic actuators were considered in [8], but these elements themselves were not taken into account when compiling the structural diagram of the brake mechanism. In work [9], this gap in the study of crane-type drum-pad brakes is partially filled, but

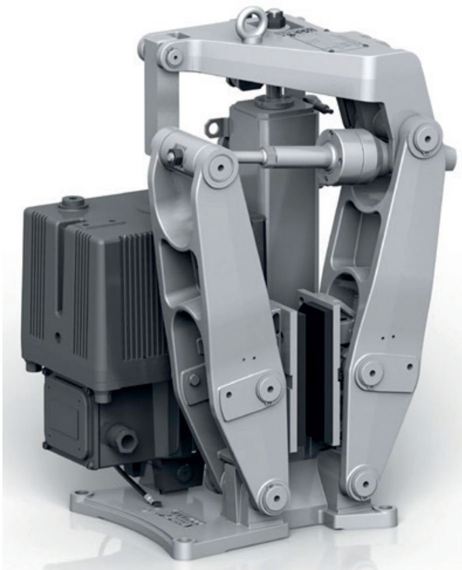


***Figure 1** Disc brake Vulkan general view [10]*

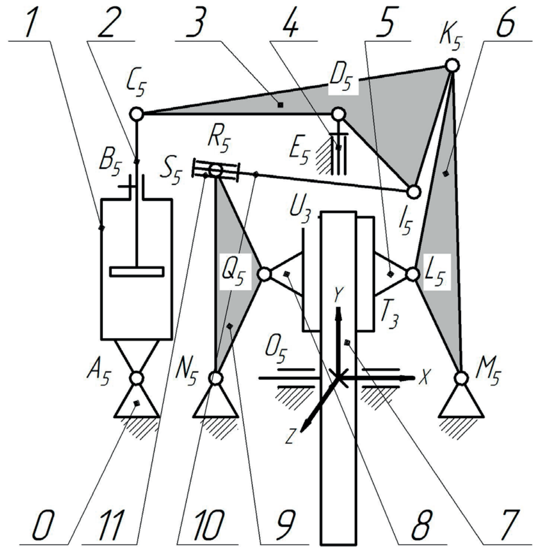


***Figure 2** Structural diagram of basic brake mechanism from Figure 1*

the structure of disc brakes remains underresearched today.

2 Methodology

Consider the structure of the Vulkan disc brake mechanism (Figure 1, Figure 2). It contains cylinder 1 of the electrohydraulic pusher attached to base 0, in which rod 2 is installed with the possibility of axial movement.

Rod 2 is hingedly connected to rocker 3, which can rotate on axis 4, and is hingedly connected to the right lever 6 of a pad 5. Disk 7 is installed with the possibility of rotation on base 0 and has the possibility of interaction of its ends with the flat surfaces of pads 5 and 8. Block 8 is hingedly fixed on the left lever 9, which is hingedly installed on base 0. The connection of the rocker 3 and the left lever 9 is provided by a rod 11, which is connected to it through a hinge formed by a sleeve 10.

In the disc brake mechanisms, the brake discs have some runout due to manufacturing and installation inaccuracies. The schematic position of the brake disc relative to the pads, at which the runout occurs, is shown in (Figure 3). In addition, some taps may not be used for a certain period of time, as a result of which corrosion occurs on the disc surface, which can also be uneven - less near the brake pads, and more in the most exposed areas. As a result of the above features, in the smooth braking mode, a variable clamping force occurs between the pads and the disc, which leads to an uneven braking torque. In special cases, the runout of the brake disc can be so great that it leads to its surfaces touching the brake pads when the brake is open, (Figure 4). In any case, the runout of the brake mechanism reduces the accuracy and smoothness of the braking process, creates an increased load on its parts and reduces their service life. This problem can be solved in several ways. The first is to increase the manufacturing accuracy of the disc and parts for its installation in the mechanism, as well as to protect the entire brake assembly from dust and moisture. Another option is to install the brake pads on the levers or install the pressure levers on the brake frame in such a way that they are dynamically self-aligned relative to the surface of the disc. To do that, it is necessary to analyze the existing types of joints in the reference mechanism using methods from the theory of machines or kinematic of machines [11-13] to determine the kinematic mobility of its elements - the so-called redundant links. The number of moving links and kinematic pairs is the determined.

The described disk brake mechanism contains eleven movable links (n = 11). Number of 5-class kinematic pairs here is P_5 = 14 (A_5, B_5, C_5, D_5, E_5, K_5, I_5, L_5, M_5, N_5, O_5, Q_5, R_5, S_5), number of 3-class kinematic pairs is P_3 = 2 (T_3, U_3), numbers of 4, 2 and 1-class kinematic pairs are $P_4 = P_2 = P_1 = 0$.

The total kinematic pairs number is:

$$P = P_5 + P_4 + P_3 + P_2 + P_1 = 14 + 0 + 3 + 0 + 0 = 16. \quad (1)$$

The sum of kinematic pairs movabilities is :

$$f = 1P_5 + 2P_4 + 3P_3 + 4P_2 + 5P_1 = 1 \times 14 + 2\times 0 + 3 \times 2 + 4 \times 0 + 5 \times 0 = 20. \quad (2)$$

Number of independent locked circuits by Gohman formula [14] is:

$$k = P - n = 16 - 11 = 5\,. \quad (3)$$

Independent locked circuits are following - $N_5Q_5U_3O_5N_5$; $M_5L_5T_3O_5M_5$; $A_5B_5C_5D_5E_5A_5$; $N_5S_5R_5I_5K_5M_5N_5$; $E_5D_5I_5R_5S_5N_5E_5$.

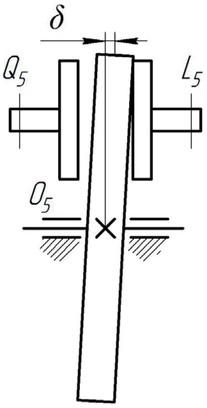


***Figure 3** Braking disk end beating δ influence*

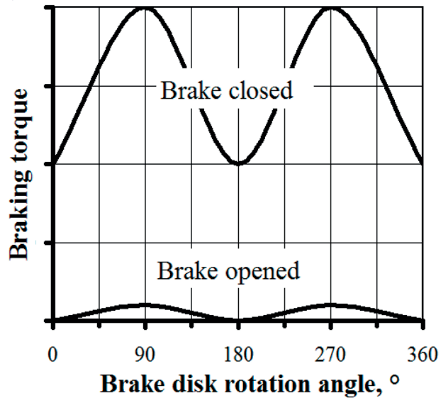


***Figure 4** Approximate influence of the disc beating on the brake friction torque in redundant constraints presence*

***Table 1** Circuit method application to basic brake mechanism (Figure 2)*

Circuit	Planar movabilities f_p			Non-planar movabilities f_n		
	f_x'	f_y'	f_z''	f_x''	f_y''	f_z'
$N_5Q_5U_3O_5N_5$	Ø	U	NQ	W_b U	Ø ↓q_1	U
$M_5L_5T_3O_5M_5$	Ø	T	ML	TO	Ø ↓q_2	T
$A_5B_5C_5D_5E_5A_5$	Ø	B	AC	Ø ↓q_3	E	Ø ↓q_4
$N_5S_5R_5I_5K_5M_5N_5$	Ø	Ø	K I S	R	Ø ↓q_5	Ø ↓q_6
$E_5D_5I_5R_5S_5N_5E_5$	Ø ↓q_7	Ø ↓q_8	D	Ø ↓q_9	Ø ↓q_{10}	Ø ↓q_{11}
$W = 1$, $q = 11$						

Total mechanism mobility by Voinea and Atanasiu equation [15-16] is:

$$W = f - \sum r_i = 20 - (3 + 3 + 4 + 4 + 5) = 1\,, \tag{4}$$

where: r_i - independent locked circuits axis ranks.

Thus, total mechanism mobility is:

$$W = W_b + W_l = 1 + 0 = 1, \tag{5}$$

where: $W_b = 1$ - basic mechanism mobility (disc 7 rotation),
$W_l = 0$ - local links mobilities.

Then, the redundant constraints number in basic variant, by Somov and Malyshev formula is:

$$\begin{aligned} q_{SM} &= W + 5P_5 + 4P_4 + 3P_3 + 2P_2 + P1 - 6n = \\ &= 1 + 5 \times 14 + 4 \times 0 + 3 \times 2 + 2 \times 0 + \\ &+ 0 - 6 \times 11 = 11. \end{aligned} \tag{6}$$

Redundant constraints number, by Ozols formula, [17] is:

$$q_{OZ} = W + 6k - f = 1 + 6 \times 5 - 20 = 11. \tag{7}$$

Thereby, the total redundant constraints number in the analyzed mechanism $q = q_{SM} = q_{OZ} = 11$.

Using the contour method confirms the presented calculations (Table 1)

The identified redundant constraints prevent the self-alignment of brake pads on the ends of the brake discs and make it possible to overload the links of the mechanism (Table 2 and Table 3).

The most dangerous among identified redundant constraints belong to power circuits q_1 and q_2 which initiate the main disadvantage of the brake mechanism - the impossibility of brake pads on disc self-alignment.

3 Results and discussion

The main way to eliminate the detected redundant connections, without intentionally introducing errors into the design and worsening the brake performance, is to add mobilities to the mechanism contours by increasing the classes of kinematic pairs. The first variant is the implementation of the 4-th class pin-spherical pairs L_4 and Q_4 instead of rotary ones (L_5, Q_5) at the points of connection of blocks 5 and 8 with levers 6 and 9, as well as modification of C_3, D_3, K_3, R_3 pairs to the 3-rd class spherical pairs and conversion of I_5 pair to the 4-th class cylindrical pair I_4 (Figure 5).

In this variant, with an unchanged total number of links, kinematic pairs, and circuits, the number of kinematic pairs of the 5-class became $P_5 = 7$ (A_5, B_5,

E_5, M_5, N_5, O_5, S_5), number of 4-class kinematic pairs $P_4 = 3$ (I_4, L_4, Q_4), 3-class kinematic pairs $P_3 = 6$ (C_3, D_3, K_3, R_3, T_3, U_3), 2 and 1-class kinematic pairs number $P_4 = P_2 = P_1 = 0$.

***Table 2** Redundant constraints presence in basic disc brake mechanism consequences*

Redundant constraint	Redundant Constraint presence influence	Practice way of redundant constraint influence leveling	Leveling absence consequences
1	2	3	4
q_1 q_2	Impossibility of pads self-aligning around *Y* axis $(f_y'' = 0)$.	Control and limitation of the end beating of the brake disc and the thickness of the pads in the plane *XZ*. This leads to an increase in the labor-intensiveness of brake maintenance.	Pads axes loading by sign variable bending moment and its levers by torque in *XZ* plane. Creating periodically changing torque in the brake shaft.
q_3	Impossibility of circuit $A_5B_5C_5D_5E_5A_5$ assembling without tension around *X* axis $(f_x'' = 0)$ in the presence of an angular error in the drilling of hinge holes or errors in of the mechanism aggregation.	Implementation of increased radial gaps in kinematic pairs *A, C, and D*. This leads to delayed brake activation.	Impossibility of mechanism assembling and brake operation without parts deformation in *YZ* and *XZ* plane.
q_4	Impossibility of circuit $A_5B_5C_5D_5E_5A_5$ assembling without tension along *Z* axis $(f_z' = 0)$ in the presence of an error in the manufacture of parts in terms of thickness or displacement of the basic surfaces in the *YZ* plane or errors of the mechanism aggregation.	Implementation of increased axial gaps in kinematic pairs *A, C, D,* or radial gap in *E* kinematic pair. This leads to delayed brake activation.	
q_5	Impossibility of circuit $N_5S_5R_5I_5K_5M_5N_5$ assembling without tension around *Y* axis $(f_y'' = 0)$ in the presence of an angular error in the drilling of hinge holes or errors in of the mechanism aggregation.	Implementation of increased radial gaps in kinematic pairs *N, R, I, K, M*. This leads to delayed brake activation.	Impossibility of mechanism assembling and brake operation without parts deformation in *YZ* and *XZ* plane.
q_6	Impossibility of circuit $A_5B_5C_5D_5E_5A_5$ assembling without tension along *Z* axis $(f_z' = 0)$ in the presence of an error in the manufacture of parts in terms of thickness or displacement of the basic surfaces in the *YZ* plane or errors of the mechanism aggregation.	Implementation of increased axial gaps in kinematic pairs *N, R, I, K, M*.	

***Table 3** Redundant constraints presence in basic disc brake mechanism consequences - continuation*

Redundant constraint	Redundant constraint presence influence	Practice way of redundant constraint influence leveling	Leveling absence consequences
1	2	3	4
q_7 q_8	Impossibility of circuit $E_5D_5I_5R_5S_5N_5E_5$ assembling without tension along *X* and *Y* axes $(f_x' = 0, f_y' = 0)$ in the presence of errors in the location of the hinge holes in the *XY* plane or errors in the mechanism aggregation.	Implementation of increased radial gaps in kinematic pairs *E, D, I, R, S, N,* or axial gap in *E* kinematic pair. This leads to delayed brake activation.	Impossibility of mechanism assembling and brake operation without parts deformation in *XY* plane.
q_9 q_{10}	Impossibility of circuit $E_5D_5I_5R_5S_5N_5E_5$ assembling without tension around *X* and *Y* axes $(f_x'' = 0, f_y'' = 0)$ in the presence of an angular error in the drilling of hinge holes.	Implementation of increased radial gaps in kinematic pairs *N, S, R, I, K, M, N*. This leads to delayed brake activation.	Impossibility of mechanism assembling and brake operation without parts deformation in *XY*, *YZ*, and *XZ* plane.
q_{11}	Impossibility of circuit $E_5D_5I_5R_5S_5N_5E_5$ assembling without tension along the *Z* axis $(f_z' = 0)$ in the presence of an error in the manufacture of parts in terms of thickness or displacement of the basic surfaces in the *YZ* plane.	Implementation of increased axial gaps in kinematic pairs *N, S, I,* or radial gap in *E, R* kinematic pairs. This leads to delayed brake activation.	

The sum of kinematic pairs movabilities:

$$f = 1P_5 + 2P_4 + 3P_3 + 4P_2 + 5P_1 = 1 \times 7 + 2 \times 3 + 3 \times 6 + 4 \times 0 + 5 \times 0 = 31. \quad (8)$$

Total mechanism mobility, by Voinea and Atanasiu equation, is:

$$W = f - \sum r_i = 31 - 5 \times 6 = 1 \quad (9)$$

Total mechanism mobility is:

$$W = W_b + W_l = 1 + 0 = 1, \quad (10)$$

where: $W_b = 1$ - basic mechanism mobility (disc 7 rotation),
$W_l = 0$ - local links mobilities (the lack of local mobility of the sleeve 10 can be ensured by boring a hole under the axis S_5 outside the center of the sphere R_3).

Then, redundant constraints number in basic variant, by Somov and Malyshev formula, is:

$$\begin{aligned} q_{SM} &= W + 5P_5 + 4P_4 + 3P_3 + 2P_2 + P_1 - 6n = \\ &= 1 + 5 \times 7 + 4 \times 3 + 3 \times 6 + 2 \times 0 + \\ &+ 0 - 66 \times 11 = 0. \end{aligned} \quad (11)$$

Redundant constraints number by Ozols formula

$$q_{OZ} = W + 6k - f = 1 + 6 \times 5 - 31 = 0. \quad (12)$$

The application of the circuit method for the variant 1 structural diagram is shown in Table 4, confirms the obtained results.

The disadvantage of the first variant is the use of pin-spherical pairs L_4 and Q_4 since it is structurally quite difficult to do this, and it is not possible to use spherical pairs here, since it can give pads 5 and 8 local mobility, which in turn can cause their rotation around the X axis and partial release from contact with the ends of the brake discs along their entire length. A good option in this case is to leave pairs L_5 and Q_5 as 5-class rotary ones.

That is why variant 2 of mechanism improvement can be spherical pairs C_3, K_3, R_3, M_3, N_3, and cylindrical pair I_4 implementation (Figure 6).

In this version, with an unchanged total number of links, kinematic pairs, and circuits, the number of kinematic pairs of the 5-class became $P_5 = 8$ (A_5, B_5, D_5, E_5, L_5, O_5, Q_5, S_5), number of 4-class kinematic pairs $P_4 = 1$ (I_4), 3-class kinematic pairs $P_3 = 7$ (D_3, K_3, M_3, N_3, R_3, T_3, U_3), 2 and 1-class kinematic

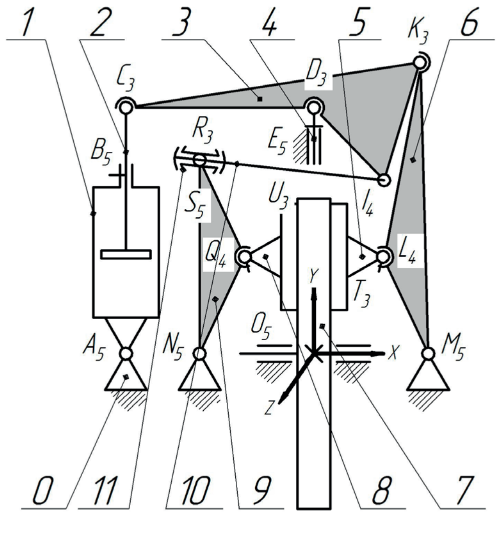


Figure 5 *Modified by variant 1 brake mechanism structural diagram*

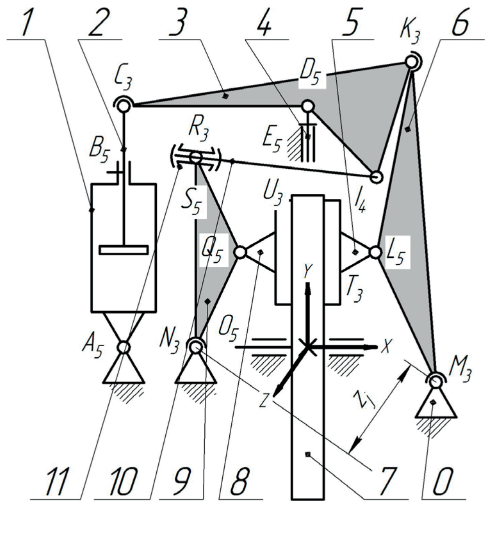


Figure 6 *Modified by variant 2 brake mechanism structural diagram*

Table 4 *Circuit method application to modified by variant 1 brake mechanism (Figure 5)*

Circuit	Planar movabilities f_p			Non-planar movabilities f_n		
	f_x'	f_y'	f_z''	f_x''	f_y''	f_z'
$N_5Q_4U_3O_5N_5$	Ø	U	NQ	U	Q	U
$M_5L_4T_3O_5M_5$	Ø	T	ML	TO	L	T
$A_5B_5C_3D_3E_5A_5$	Ø	B	AC	C	CE	Ø
$N_5S_5R_3I_4K_3M_5N_5$	Ø	Ø	IK	K	K	I
$E_5D_3I_4R_3S_5N_5E_5$	Ø	Ø	DR	DR	DR	Ø
W = 1, q = 0						

pairs number $P_4 = P_2 = P_1 = 0$.

The sum of kinematic pairs movabilities is:

$$f = 1P_5 + 2P_4 + 3P_3 + 4P_2 + 5P_1 = 1 \times 8 + \\ + 2 \times 1 + 3 \times 7 + 4 \times 0 + 5 \times 0 = 31. \tag{13}$$

Total mechanism mobility, by Voinea and Atanasiu equation, is:

$$W = f - \sum r_i = 31 - 5 \times 6 = 1 \tag{14}$$

Total mechanism mobility is:

$$W = W_b + W_l = 1 + 0 = 1, \tag{15}$$

where: $W_b = 1$ - basic mechanism mobility (disc 7 rotation),
$W_l = 0$ - local links mobilities.

Then redundant constraints number in basic variant, by Somov and Malyshev formula, is:

$$q_{SM} = W + 5P_5 + 4P_4 + 3P_3 + 2P_2 + P_1 - 6n = \\ = 1 + 5 \times 8 + 4 \times 1 + 3 \times 7 + 2 \times 0 + \\ + 0 - 6 \times 11 = 0. \tag{16}$$

Redundant constraints number, by Ozols formula, is:

$$q_{OZ} = W + 6k - f = 1 + 6 \times 5 - 31 = 0. \tag{17}$$

The application of the circuit method is shown in Table 5, confirms the obtained results.

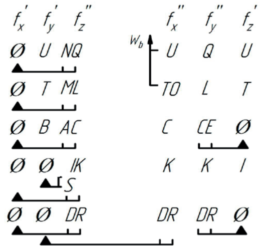
The stiffness of the structure can be increased by installing the kinematic pairs M and N at a distance z_j along the Z axis.

4 Conclusion

Theoretical studies of the existing crane disc brake mechanism have been performed, and the following conclusions have been drawn:

1. Structural analysis of the kinematics of the brake pad drive mechanism showed that the mechanism has 11 redundant constraints. These constraints do not allow the brake pads to self-adjust relative to the brake disc surface. Due to that, in the event of a brake disc beating, the braking torque will not be constant.
2. The analysis identified angular constraints as the predominant challenge in the kinematics of the mechanism's constraints (joints). To mitigate the risk of jamming (or to ensure guaranteed mobility), it is necessary to manufacture these joints with either high accuracy of the mating parts, which results in an increase in price, or, conversely, to create an increased gap in the joint, which in turn increases the brake response time due to the movement of the mating parts in the joints with a gap from one extreme position to another. Concurrently, the presence of these gaps can diminish the service life of the joints, thereby increasing the impact loads they experience. That, in turn, can lead to a reduction in their fatigue strength, particularly in the drive levers.
3. A particular group of restrictions imposed on the drive mechanism can impede the ability of the brake pads to self-adjust relative to the brake disc surface. Such contact can lead to inconsistent braking torque within a single revolution, resulting in diminished smoothness and accuracy in load positioning. Furthermore, in instances of substantial brake disc beating or malfunctioning of the brake pad retraction mechanism, the pads may inadvertently come into contact with the disc surface. This phenomenon contributes to a decline in the smoothness and accuracy of movement of the load suspended on the cable, thereby reducing the service life of the brake mechanism. It also leads to accelerated wear, overheating, and increased dynamic loads.
4. In this paper, the authors proposed two design solutions for the kinematics and, consequently, the design of the brake pad drive mechanism. These solutions allow for elimination of excessive restrictions that prevent the pads from self-installing relative to the pressure levers and the surface of the brake disc. The first option proposes the use of pin spherical pads. The primary benefit of this approach

Table 5 *Circuit method application to modified by variant 2 brake mechanism (Figure 6)*

Circuit	Planar movabilities f_p	Non-planar movabilities f_n
$N_3Q_5U_3O_5N_3$		
$M_3L_5T_3O_5M_3$		
$A_5B_5C_3D_5E_5A_5$		
$N_3S_5R_3I_4K_3M_3N_3$		
$E_5D_5I_4R_3S_5N_3E_5$		
$W = 1$, $q = 0$		

is that it necessitates only a modification to the pad mount to the pressure levers. However, a notable drawback is the complexity of implementing such a mount, particularly with regard to its strength and manufacturing technology. The necessity of a compact, durable joint underscores the challenges associated with this approach. The second option involves a substantial redesign of the brake mechanism, entailing the installation of the pressure levers on spherical joints, necessitating a corresponding modification to the brake mechanism's frame. This approach offers the benefit of preserving the original design of the replaceable brake pads.

5. Both proposed design solutions can be implemented on the disk brake of the crane without altering the mechanism's operational principle. It is noteworthy that the Vulcan crane disc brake utilizes solely cylindrical joints, which, in addition to their manufacturability, offer the benefit of a two-support design. Conversely, the implementation of a spherical joint is only feasible as a cantilever design. From the perspectives of strength, fatigue, tribology, manufacturing technology, and installation procedure, this type of joint necessitates a series of research and development endeavors in either of the two design solutions.

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Conflicts of interest

The authors declare that they have no known competing financial interests or personal relationships that could have appeared to influence the work reported in this paper.

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