INFLUENCE OF FRICTION IN JOINTS ON FORCE CHARACTERISTICS OF SCISSOR ROLLING BRIDGE JACK

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Abstract. The object of the study is the operation of a scissor rolling bridge jack used for vehicle maintenance under service conditions. The subject of the study is the influence of friction in the joints and the geometric parameters of the scissor mechanism of the rolling bridge jack on the force in its drive. The paper analyses the kinematic and force schemes of an automotive scissor rolling bridge jack with a hydraulic drive. The mechanism was calculated taking into account the friction angles in the joints of its links. Based on the force analysis, an analytical relationship was obtained between the load applied to the crossbeam of the scissor rolling bridge jack and the axial force acting on the rod of the hydraulic cylinder. Force characteristics were determined as functions of the friction coefficients in the mechanism joints and the specified geometric parameters. A skeleton and a parameterized three-dimensional model of the scissor rolling bridge jack were developed in the Creo Parametric 12 environment, which made it possible to perform computer simulation of the mechanism operation while taking into account the specified geometric parameters and friction in the joints. The simulation results yielded force characteristics that are in qualitative and quantitative agreement with the theoretical calculations. The parametric model made it possible to assess the influence of the geometric parameters of the mechanism, the position of the drive hydraulic cylinder, and the friction coefficients on the drive force. For the adopted quantitatively defined structure, interferences between the mechanism elements in the folded position occur at a link arm ratio of $L/(2l) > 2.76$ and an offset of $x > 20$ mm. Under unfavorable operating conditions, the contribution of friction ($\mu_k' = 0.42$) in the joints of the scissor mechanism may amount to 25-30% of the total force on the hydraulic cylinder rod.

Keywords: scissor rolling bridge jack, scissor mechanism, joint friction, force analysis, drive force.

Introduction

Lifting and inspection equipment plays an important role in the maintenance of wheeled vehicles. The reduction in vehicle ground clearance characteristics of modern cars requires a minimum jack height when it is located between the longitudinal platforms of four-post automotive lifts or in inspection pits. The lower limit of the overall platform height is determined by the need to accommodate the mechanism and drive elements and by the nonlinear increase in the drive force. It is known that the maximum force developed by the hydraulic cylinder of a lift equipped with a scissor mechanism occurs at the instant when motion starts from the extreme lower position. The study of scissor-mechanism designs and their drives is relevant because such mechanisms are used in lifting devices in various fields [1-5].

Scissor lifting mechanisms are widely used in automotive service equipment due to their compactness in the folded position and sufficient load-carrying capacity. Such devices may be equipped with manual, electromechanical, or hydraulic drives. In this study, attention is focused specifically on scissor rolling bridge jacks with hydraulic drive, since such drives, while having small overall dimensions, are capable of developing high forces, which is particularly important for minimizing the jack height in the fully lowered position.

In kinematic and dynamic studies, mechanisms are considered both without accounting for friction in the joints (in most cases) [1-4] and with friction models of various degrees of simplification [5]. However, analysis of the mechanism operation shows that friction in its joints has a significant effect on the mechanism performance as a whole [5].

The poor condition of friction surfaces due to insufficient lubrication (dry friction), ingress of abrasive dirt particles, corrosion, wear products etc., may lead to a substantial increase in the friction coefficient. Operation of an automotive scissor rolling bridge jack under conditions of excessive friction in the joints leads to increased wear of the connections, higher operating forces, and may result in exceeding permissible loads as well as a risk of mechanism jamming.

To investigate the mechanism operation over the entire range of motion, particularly in critical positions corresponding to the start of lifting, it is advisable to use computer simulation [1; 4].

Materials and methods

For the operating conditions of the adopted scissor rolling bridge jack, the limits of the minimum H_{MIN} and maximum H_{MAX} platform positions are determined. Determining the minimum platform position is a compromise between ensuring the minimum overall height of the structure and the possibility of accommodating the mechanism elements.

The study of the scissor rolling bridge jack operation using computer simulation was carried out in the Creo 12 software environment [6]. For the study, the most common structural scheme of a scissor rolling bridge jack with a quantitatively defined structure was selected (Fig. 1, a); this structure specifies the composition, number, and relative arrangement of the mechanism elements. For the selected quantitatively defined structure, a parameterized sketch was created, which subsequently served as the basis for forming the model skeleton (Fig. 1, b). This approach made it possible to define the key dimensions, joint positions, and force-application points as controlled parameters, thereby ensuring geometric consistency when changing the mechanism configuration, as well as to identify interferences between the parts of the three-dimensional model of the device (Fig. 1, c) [6].

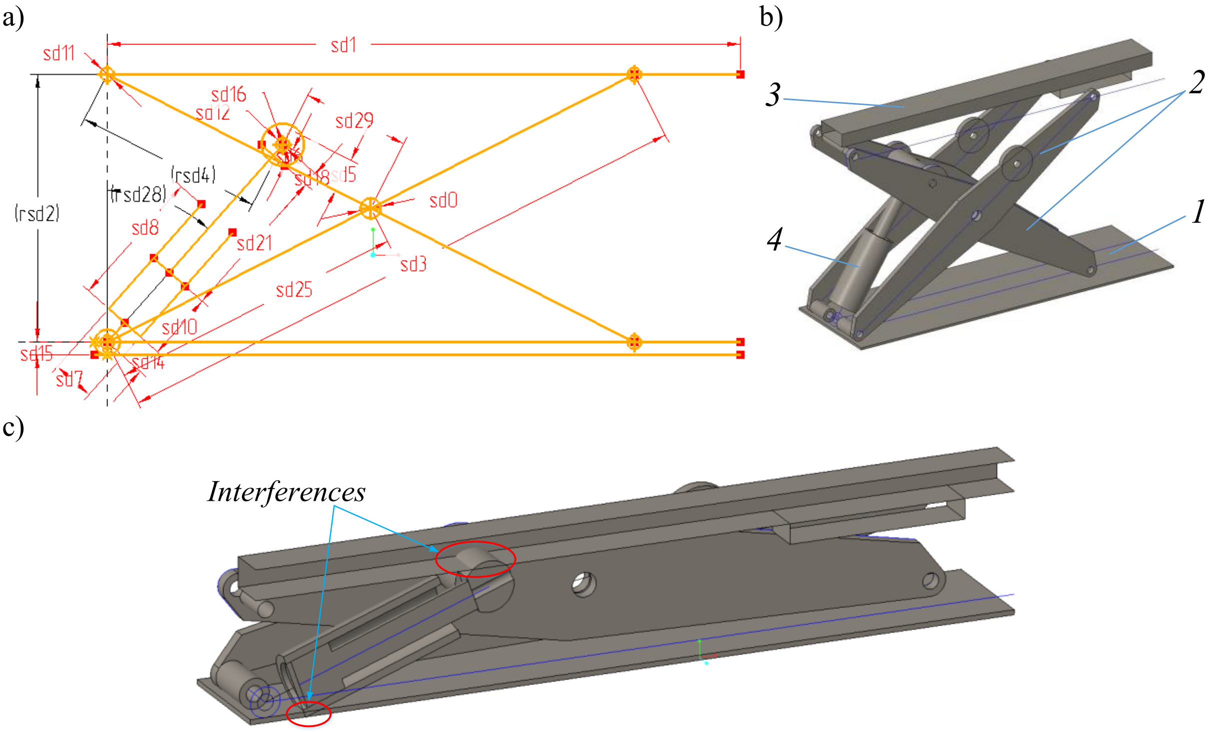

Fig. 1. **Skeleton geometry of the scissor rolling bridge jack:** a – quantitatively defined structure; 1 – base weldment; 2 – scissor assembly; 3 – extension arm tube assembly; 4 – cylinder assembly; b – skeleton geometry of the scissor rolling bridge jack; c – interference zones between components in the folded position

The use of the skeleton model makes it possible to implement the top-down design principle, in which the geometry of individual parts is formed on the basis of a unified reference structure [7]. This considerably simplified the assembly modeling, eliminated geometric inconsistencies, and ensured model stability during parametric studies.

An analysis of jacks of this type shows that the most common arrangement is fixing the lower end of the hydraulic cylinder 4 to paired links at their attachment point to the base (Fig. 1, b) joint 5 (7) (Fig. 2, a), while the upper end is fixed to the other pair of links at a distance l from the central joint of the scissor mechanism (Fig. 1, a, Fig. 2) and with an offset x relative to the axis of link 1-4 (Fig. 2, b). The position of the hydraulic cylinder is determined by the force arms $L/2$ and l and can be characterized by the arm ratio $L/(2l)$, where L is the link length.

The friction coefficients μ' reported in the literature vary within a wide range (up to 0.78 for steel-steel pairs [8; 9]), since they are determined by many design and operating factors: the quality of mating parts (materials, surface finish, coatings, etc.), the operating conditions of the joints (lubrication, contact pressure, speed, environmental parameters, etc.). The onset of motion in joints during mechanism start-up is accompanied by the transition from static to kinetic friction and is characterized by the need to overcome the peak static-friction forces. This process is characterized by an abrupt decrease in friction forces and is sometimes accompanied by self-excited oscillations.

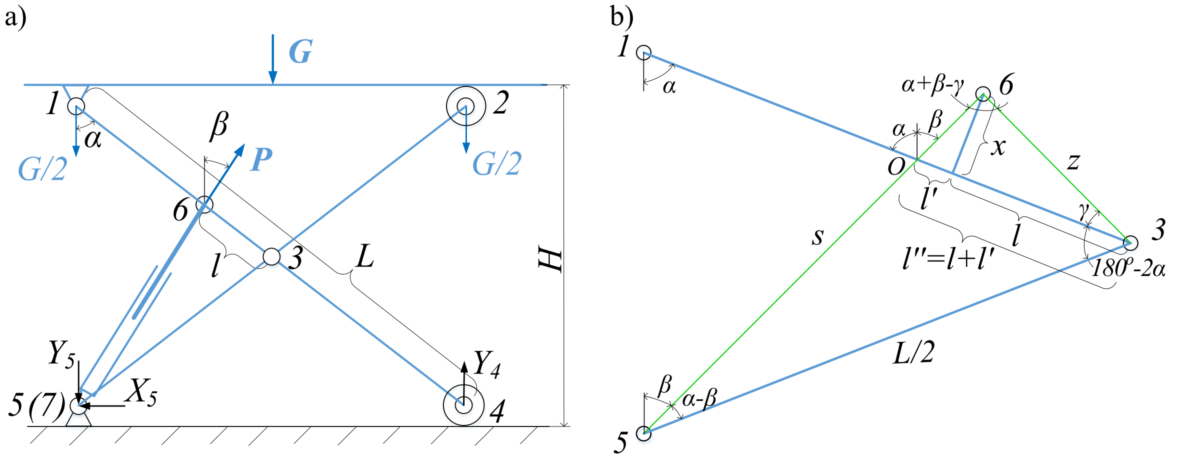

Fig. 2. **Scheme for the kinematic analysis and force calculation of the scissor rolling bridge jack:** a – general scheme of external forces and reactions and determination of the geometric parameters of the mechanism; b – scheme for taking into account the offset (x) of the attachment point of the upper hydraulic-cylinder joint relative to the longitudinal axis of the scissor-mechanism link

According to the existing recommendations for selecting the friction coefficient, one should consider not a specific value established under certain conditions, but a range of possible values determined by the operating conditions of the mechanism. The literature states that in a pin-bushing joint the dry-friction coefficient can be taken as $\mu \approx (1.3\text{-}1.5)\mu'$, where μ' – is the friction coefficient of flat contacting surfaces made of the same material [9].

The position of point 6, which is the attachment point of the upper hydraulic-cylinder joint, is determined using the sine law for the triangle with vertices 3, 5, and 6 (Fig. 2, b).

$$\frac{z}{\sin(\alpha-\beta)}=\frac{s}{\sin(180^\circ-2\alpha+\gamma)}=\frac{L/2}{\sin(\alpha+\beta-\gamma)}. \tag{1}$$

The angle β of inclination of the hydraulic-cylinder axis relative to the vertical and its length s are determined by the following relationships (2):

$$\beta=\arctan\left(\frac{L/2\cdot\sin\alpha-z\cdot\sin(\alpha-\gamma)}{L/2\cdot\cos\alpha+z\cdot\cos(\alpha-\gamma)}\right);\ s=\frac{L/2\cdot\sin(2\alpha-\gamma)}{\sin(\alpha+\beta-\gamma)}. \tag{2}$$

The position of point O, which is the intersection point of the line of action of force P with the longitudinal axis of link 1-3 (Fig. 2, b), is determined by the distance $l''=l+l'=l+x/\tan(180^\circ-(\alpha+\beta))$.

To determine the initial hydraulic-cylinder force P required to overcome the weight (mass) of the load G, which is symmetrically placed on the platform, while accounting for friction losses, the system of forces and reactions applied to the scissor mechanism is considered; in the plane scheme it is represented by two links *I* and *II* (Fig. 3).

Friction in joints 1, 2, 4, and 5 of the mechanism links (except for the central joint 3), denoted by the indices i and j, is taken into account as additional angles $\varphi_{ij}+\varphi_{rj}$, determined from the radii ρ of the friction circles [9; 10] (Fig. 4), taking into account the symmetry of the scissor mechanism ($G/2$; $d_i=d_j$; $\rho_i=\rho_j$, etc.).

The friction angle is determined from the equation $\sin\varphi = 2\rho/d$ with the assumption: $\tan\varphi$ ($\sin\varphi$) $\approx\varphi$ for small angles. The friction angles φ_{ij} in the end joints (diameter d_{ij}) of the links (length L_{ij}) and the hydraulic cylinder (length s) are determined as: $\varphi_{ij}=\arcsin(\mu d_{ij}/L_{ij})$, while the equivalent friction angles in the roller supports are $\varphi_{rj}=\arctan(\tan(\varphi_{ij})\cdot d_j/D_j)$.

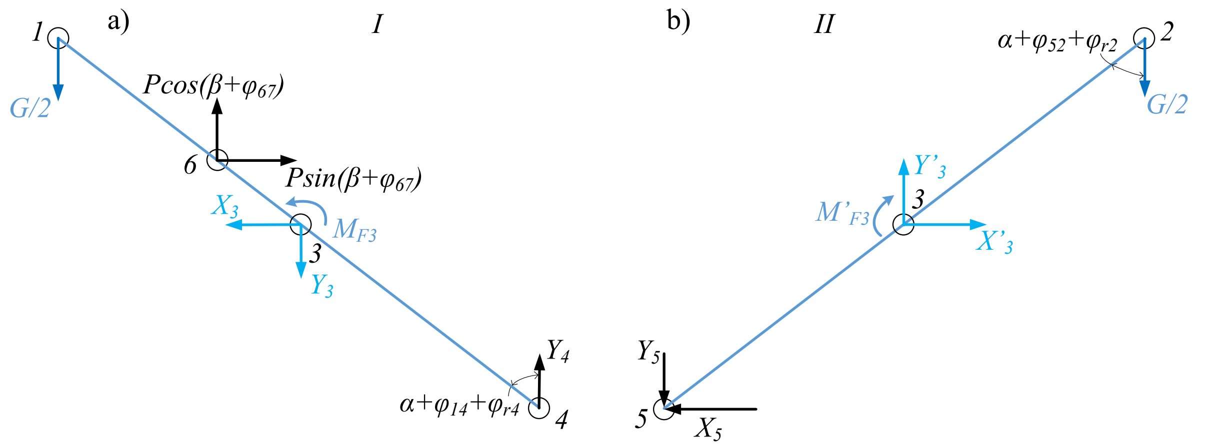

Fig. 3. **Scheme for determining forces in the joints of the scissor mechanism (two-body system):** a – link *I* (1-4); b – link *II* (5-2)

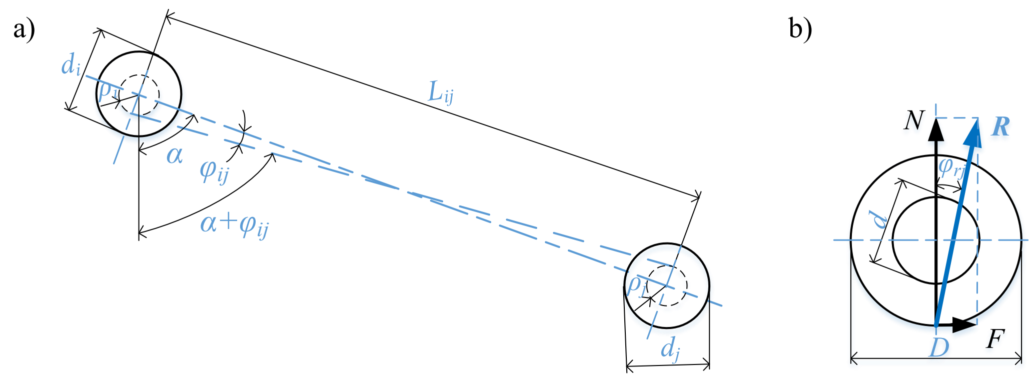

Fig. 4. **Scheme of friction angles in the joints of the scissor mechanism:** a – friction angles φ_{ij} in the end joints *i*, *j* of links 1-4 and 5-2); b – equivalent friction angle φ_{rj} in the roller support (joints 2 and 4)

For the force system shown in Fig. 3 the classical system of equations was written (3):

For link *I*:

$$\sum X_i = 0; \Rightarrow P \cdot \sin(\beta + \varphi_{67}) = X_3\,;$$

$$\sum Y_i = 0; \;\Rightarrow P \cdot \cos(\beta + \varphi_{67}) + Y_4 = Y_3 + G/2\,;$$

$$\begin{aligned}\sum M_{3i} = 0; \;\Rightarrow\; & (G/2 + Y_4) \cdot L/2 \cdot \sin(\alpha + \varphi_{14} + \varphi_{r4}) + M_{F3} = \\ & = P \cdot l'' \cdot (\cos(\beta + \varphi_{67}) \cdot \sin(\alpha + \varphi_{14} + \varphi_{r4}) + \\ & + \sin(\beta + \varphi_{67}) \cdot \cos(\alpha + \varphi_{14} + \varphi_{r4})) = \\ & = P \cdot l'' \cdot \sin((\alpha + \varphi_{14} + \varphi_{r4}) + (\beta + \varphi_{67})).\end{aligned}$$

For link *II*:

$$\sum X_i = 0; \;\Rightarrow\; X_3' = X_5\,;$$

$$\sum Y_i = 0; \;\Rightarrow Y_3 = Y_5 + G/2\,;$$

$$\begin{aligned}\sum M_{5i} = 0; \;\Rightarrow\; & Y_3' \cdot L/2 \cdot \sin(\alpha + \varphi_{52} + \varphi_{r2}) = \\ & = X_3' \cdot L/2 \cdot \cos(\alpha + \varphi_{52} + \varphi_{r2}) + \\ & + G/2 \cdot L \cdot \sin(\alpha + \varphi_{52} + \varphi_{r2}) + M_{F3} \\ \Rightarrow\; & Y_3' = X_3' \cdot \cot(\alpha + \varphi_{52} + \varphi_{r2}) + G + \\ & + M_{F3} / (L/2 \cdot \sin(\alpha + \varphi_{52} + \varphi_{r2})).\end{aligned} \tag{3}$$

The friction moment M_{F3} in the central joint 3 is determined by (4):

$$M_{F3} = R_3 \cdot d_3 / 2 \cdot \mu = (k_1 \cdot X_3 + k_2 \cdot Y_3) \cdot d_3 / 2 \cdot \mu\,, \tag{4}$$

where d_3 – is the diameter of the pin and bushing of joint 3;
k_1, k_2 – coefficients determined for a given ratio X_i/Y_i, as partial derivatives of the function used to represent the function in the form of a linear combination $k_1 \cdot X_i + k_2 \cdot Y_i$, resulting from a first-order multivariable Taylor expansion.

This approach was applied because the above system of equations, when combined with the expression for the friction moment in the form:

$$M_{F3} = R_3 \cdot d_3 / 2 \cdot \mu = \sqrt{X_3^2 + Y_3^2} \cdot d_3 / 2 \cdot \mu ,$$

does not yield a direct solution. Previous studies of the mechanism indicate the ratio of components $X_3/Y_3 \approx 2$. Therefore, for the model the coefficients $k_1 = 0.883$ and $k_2 = 0.463$ were adopted, providing the minimum average error of the resultant R_3, about 1-1.5%, over the entire interval $X_3/Y_3 = 1.5\text{-}2.5$.

From the system, the force on the hydraulic-cylinder rod is determined (5):

$$P = \frac{2 \cdot G}{(1 - A_2) \cdot (A_3 + \cos(\beta + \varphi_{67}) - A_1 \cdot \sin(\beta + \varphi_{67})) - (1 + A_2) \cdot \sin(\beta + \varphi_{67}) \cdot (\cot(\alpha + \varphi_{14(52)} + \varphi_{r4(2)}) + A_1)}, \quad (5)$$

where $A_{1(2)} = k_{1(2)} \cdot \mu \cdot d_3 / A_4$;

$A_3 = 2 \cdot l'' \cdot \sin(\alpha + \varphi_{14(52)} + \varphi_{r4(2)} + \beta + \varphi_{67}) / A_4$;

$A_4 = L \cdot \sin(\alpha + \varphi_{14(52)} + \varphi_{r4(2)})$.

Results and discussion

The minimum lowering height of the jack was determined using the skeleton model, which provided a parametric description of the limiting lower position of the mechanism, in particular with account taken of the ratio $L/(2l)$, the offset x, and the cylinder diameter (Fig. 5).

Thus, for the adopted quantitatively defined structure, interferences between the mechanism elements in the folded position arise at $L/(2l) > 2.76$ and $x > 20$ mm. For a specified platform load of $G = 20$ kN and typical geometric dimensions of the structural elements of a jack with a scissor mechanism (link lengths $L_{ij} = 700$ mm; joint diameters $d_{(1,2,4,5)} = 16$ mm, $d_3 = 24$ mm, $d_{(6,7)} = 28$ mm, $D_{(4,2)} = 60$ mm), analytical calculations were performed using the proposed mathematical apparatus, along with computer simulation taking into account the interaction of the links and the conditions of their connections, for the arm ratio in the interval $L/(2l) = 2.59\text{-}2.76$ and the offset of the hydraulic-cylinder attachment point relative to the link axis in the interval $x = 0\text{-}20$ mm. The force was obtained without considering friction (Fig. 6) and for two values of the kinetic-friction coefficient for unlubricated steel surfaces: $\mu_k' = 0.20$ (Fig. 7, a) and $\mu_k' = 0.42$ (Fig. 7, b).

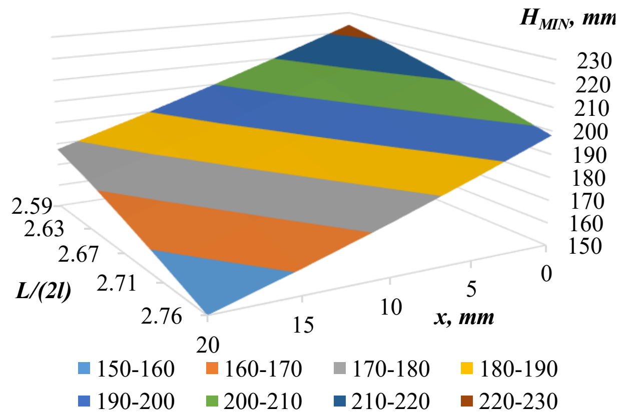

Fig. 5. **Dependence of the minimum jack platform height on the offset *x* and the ratio *L*/(2*l*) of the arms of the scissor-mechanism link**

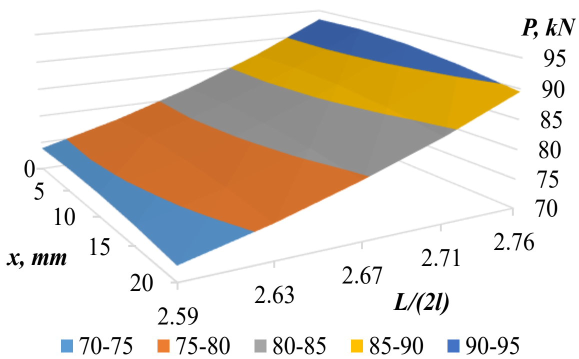

Fig. 6. **Dependence of the force *P* on the jack hydraulic-cylinder rod on *x* and *L*/(2*l*) without accounting for friction in the joints of the scissor mechanism**

Under the specified unfavorable operating conditions of the jack, the contribution of kinetic friction ($\mu_k' = 0.42$) in the joints of the scissor mechanism may reach 25-30% of the total force on the hydraulic-cylinder rod. In addition, at the initial instant of mechanism motion, an additional force must be developed by the hydraulic cylinder to overcome the resistance during the transition from static to kinetic friction in the joints and in the elements of the hydraulic cylinder itself.

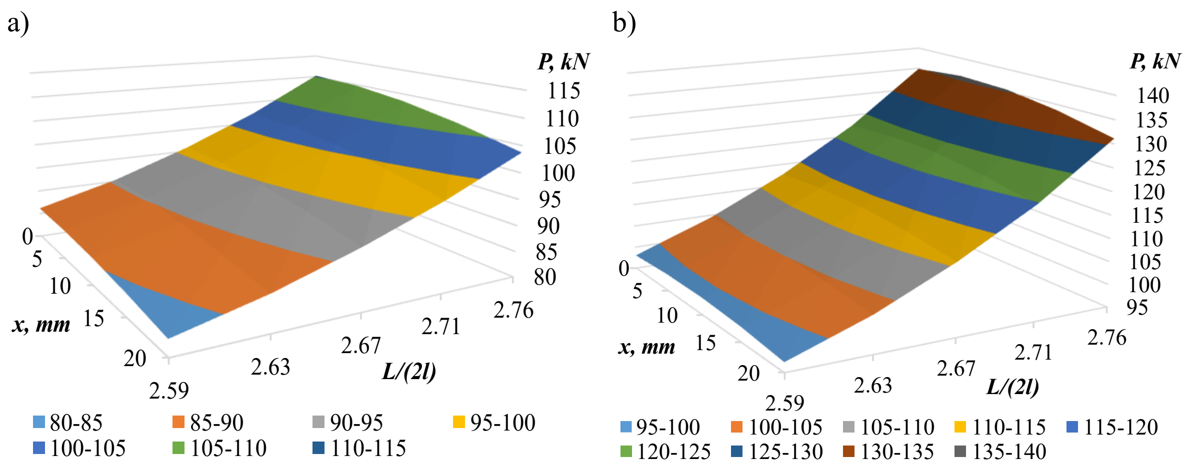

Fig. 7. **Dependence of the force *P* on the jack hydraulic-cylinder rod on *x* and *L*/(2*l*) taking into account friction in the joints of the scissor mechanism:** a – $\mu_k' = 0.2$; b – $\mu_k' = 0.42$

The parametric nature of the model made it possible to vary the geometric parameters of the mechanism, the position of the drive, and the friction coefficients in the joints, which allowed the dependence of the drive force on the mechanism configuration and operating conditions to be obtained. Thus, for the adopted geometric parameters, a 6% change in the ratio $L/(2l)$, even without considering friction in the joints, leads to an increase in the rod force by up to 26%, whereas shifting the attachment point of the upper hydraulic-cylinder joint by up to 20 mm reduces the force by up to 4%. Accordingly, when friction in the joints is taken into account ($\mu_k' = 0.42$), the same change in the ratio $L/(2l)$ leads to an increase in force by up to 38%, while the offset x to 20 mm reduces the force by up to 3%.

Conclusions

A comprehensive approach to the analysis of a scissor rolling bridge jack is proposed, combining analytical modeling based on a system of equations for calculating the hydraulic drive forces taking into account friction in the mechanism joints and computer simulation in the Creo environment. The implementation of the methodology using top-down design technology and skeletons of quantitatively defined structures ensures consistent formation of the mechanism geometry, variation of its parameters, and investigation of the jack performance characteristics.

Based on the proposed methodology, graphical relationships were obtained that make it possible to determine the minimum lowering height of the jack platform and the force in the most heavily loaded position. It is shown that accounting for friction in the mechanism joints is necessary, since the friction-force component has a significant effect on the total drive force.

The computer simulation results confirmed the analytically obtained relationships and the determined values of the maximum forces for the specified geometric parameters. This confirms the expediency of using three-dimensional parametric modeling and top-down design technology for the study and improvement of scissor lifting mechanisms.

Author contributions

Conceptualization, Y.B.; methodology, Y.B., M.T., V.P. and N.T.; software, Y.B.; validation, Y.B.,V.P., M.T. and N.T; formal analysis, M.T. and N.T.; investigation, Y.B., V.P., M.T. and N.T.; data curation, Y.B., M.T and V.P.; writing – original draft preparation, V.P.; visualization, Y.B., V.P. All authors have read and agreed to the published version of the manuscript.

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