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MANUFACTURING ENGINEERING AND AUTOMATED PROCESSES

МАШИНОБУДУВАННЯ, АВТОМАТИЗАЦІЯ ВИРОБНИЦТВА ТА ПРОЦЕСИ МЕХАНІЧНОЇ ОБРОБКИ

UDC 629.3

ANALYSIS OF KINEMATIC AND FORCE PARAMETERS OF LEANING-AND-SHUNTING DOOR OPENING MECHANISM OF ELECTRIC PUBLIC TRANSPORT

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Summary. *The paper analyses the design and operational peculiarities of the leaning-and-shunting door opening mechanism of electric public transport. The corresponding design of the mechanism is proposed. It is suggested to be driven by electric drive. The simplified diagram of the mechanism is constructed. Structural and kinematic analysis of the mechanism is performed, and the analytical dependencies for describing the motion of its links during the door opening/closing are derived. The main kinematic parameters of the studied mechanism are investigated on the basis of the derived analytical dependencies in MathCAD software, as well as by means of simulating the motion of the solid-state model of the mechanism designed in SolidWorks software. The conclusions about the agreement of the results of theoretical investigations performed on the basis of numerical solving the obtained motion equations and of the virtual experiment (motion simulation in SolidWorks software) are drawn. The analysis of energy efficiency of the investigated leaning-and-shunting door opening mechanism of electric public transport is carried out. Prescribing the resistance force acting on the door leaf and calculating the corresponding door motion speed during its closing, the dependency of the nominal power supply of the mechanism drive as a function of corresponding generalized coordinates was deduced. Analyzing the obtained results, it was established that the necessary nominal power of the leaning-and-shunting mechanism drive is almost twice smaller than the necessary nominal power of the widely used turning-and-shunting door mechanism of electric public transport.*

Key words: *leaning-and-shunting mechanism, electric public transport, structural analysis, kinematic analysis, energy efficiency.*

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Introduction and statement of the problem. Modern living standards cause the growth of the citizens' needs in a faster, convenient and safe transportation. One of the most important problems occurring while designing electric public vehicles is the problem of controlling the mechanisms for opening/closing doors. At the present time, there are dozens of designs of these mechanisms, which differ from each other by the following features: the

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direction of the door movement with respect to the cabin of the vehicle (outside or inside), the drive (hand, electric, hydraulic, pneumatic), the number of moving elements of doors, the type of anti-cramping systems, etc. While developing new designs of door mechanisms, an important task is to maximize their efficiency and to improve their safety and comfortability. Therefore, the designers face quite complicated problems of improving the known or developing new structural and design diagrams of door mechanisms of public transport.

Analysis of the available investigations. The structural and kinematic characteristics, as well as the methods of geometrical synthesis of different mechanisms of actuating doors of urban buses are presented in numerous scientific publications. Most of these works [1−3] consider the kinematic and geometrical properties of the mechanisms, in particular the swept volume, pressure angles, mechanism stalling etc. The other investigations [4, 5] deal with simulation analysis and computer-aided design of the improved door mechanisms of public vehicles. One more group of researchers [6, 7] studies the peculiarities of pneumatically operated door systems of electric vehicles, in particular noise conditions, operational safety, passengers convenience etc.

The analyzed investigations on the subject of the paper do not cover the problems of energy efficiency of door mechanisms. This paper presents the second part of the authors' research related with a new design of leaning-and-shunting door mechanism. Unlike the presented paper, the previous (first) part of the research dealt with turning-and-shunting door mechanism. Thus, on the basis of these parts the comparative analysis of energy efficiency of two widely used door mechanisms of electric public transport is to be performed.

The objective of the present work consists in comparative analysis of power consumption of different door mechanisms of public vehicles on the basis their structural and kinematic analysis and motion simulation using MathCAD and SolidWorks software.

Substantiation of the investigated design of door portal of a public vehicle. One of the most common designs of the door portals of electric public transport is the leaning-and-shunting configuration (Fig. 1). In this case, door leaves 1 are hingely joined by L-shape levers 2 to vertical drive shafts (tubes) 3 placed on each sides of the door portal. Vertical drive shafts 3 have the ability to rotate around their own longitudinal axes and they are driven by the lever mechanisms 4 with a help of the electric geared motor 5. In order to ensure the parallelism of the door leaves to the vehicle body during the process of opening/closing the doors (outside relative to a cabin), the rotary levers 6 are used.

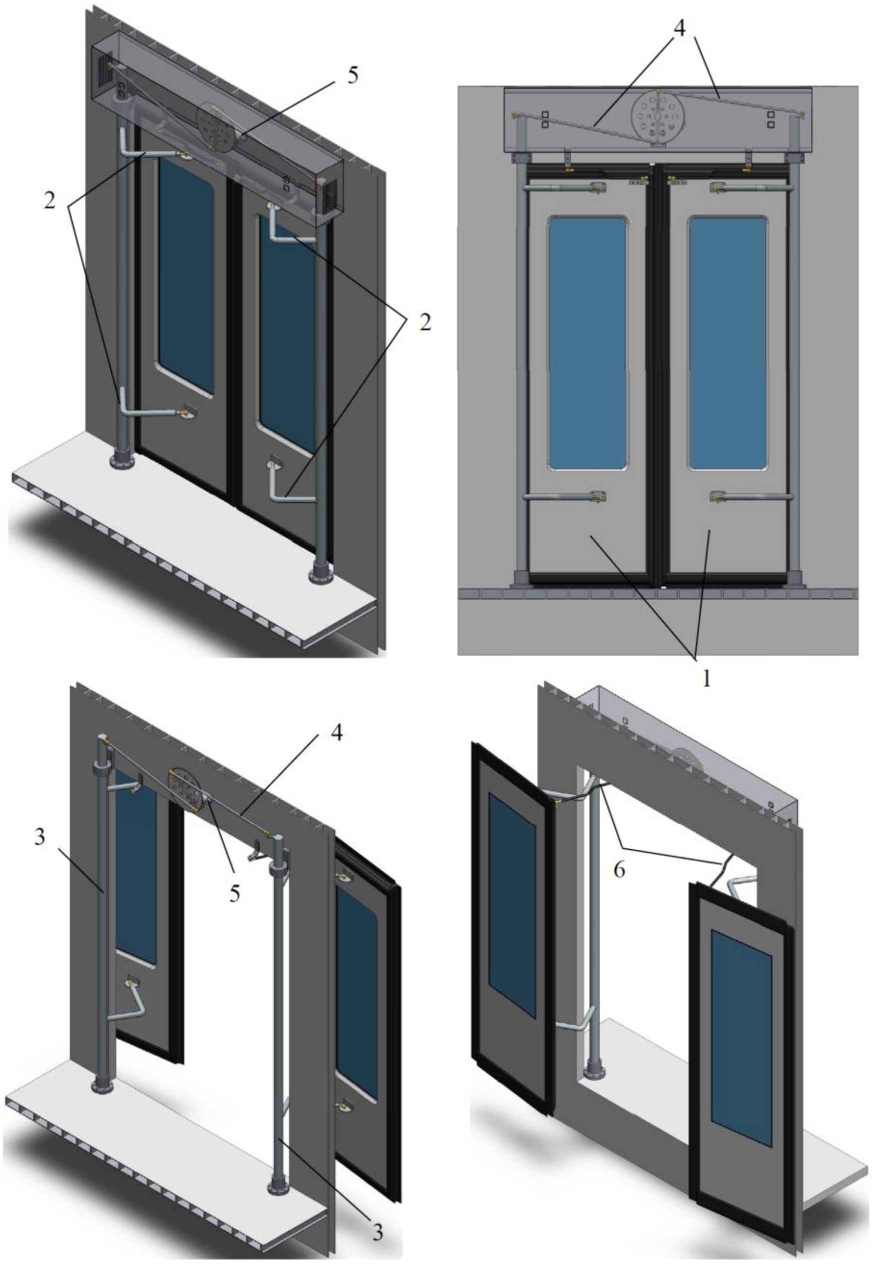


Figure 1. The analysed design of the leaning-and-shunting doors of public transport

Design and kinematic diagrams of the door mechanism. Let us consider the basic design of the leaning-and-shunting door opening mechanism of the electric public vehicle (Fig. 2, a). It consists of the electric geared motor 1 mounted on the body 2 of the vehicle. The driving disc 3 is fixed on the shaft of the motor 1 and is hingely joined with the lever 4. With a help of a spherical hinge, the lever 4 is connected to the rocker 5 of the vertical shaft 6, to which a door leaf 8 is hingely joined by means of the lever 7. In order to provide the leaning-and-shunting method of closing the door, the guide lever 9 is hingely joined with the door leaf 8. The other end of the door leaf 8 is joined with the body 2 of the vehicle by the hinge 10. Thus, during the process of rotation of the driving disk 3 (Fig. 2, a), it turns the lever 4, the rocker 5, the shaft 6, and the lever 7. The lever 7 sets the door leaf 8 in motion. The door leaf 8 rotates around the hinge of its connecting to the guide lever 9 and, at the same time, rotates with the lever 9 around the hinge 10.

On the basis of the considered design of the leaning-and-shunting door opening mechanism, let us develop its simplified kinematic diagram (Fig. 2, b). The mechanism consists of the driving rocker 1, which is hingely joined to the fixed axis of rotation O_1. The coupler 2 is connected to the driving rocker 1 and to the rocker 3 by the spherical hinges *A*. and *B*. The other end of the rocker 3 is hingely joined with the axis of rotation O_2. The coupler 4 (which is a door leaf) is connected to the rocker 3 by the cylindrical hinge *C*. The other end of the coupler 4 is hingely joined with the rocker 5 at the point *D*. The other end of the rocker 5 is connected to the body of the vehicle by cylindrical hinge O_3.

Structural and kinematic analysis of the mechanism. The leaning-and-shunting mechanism for opening the doors of electric public vehicle is considered as an assemblage of two mechanisms: the spatial driving one (including the driving rocker 1, the coupler 2, the rocker 3) and the plane guiding one (including the rocker 3, the coupler 4 and the rocker 5). The first mechanism (O_1ABO_2) is the spatial mechanism consisting of three movable links (Fig. 2, b): crank (rocker) 1, coupler 2, rocker 3, two single-motion turning kinematic pairs (kinematic pairs of the fifth order): O_1, O_2, and two three-motion kinematic pairs (kinematic pairs of the third order − spherical hinges) *A, B*. Thus, in accordance with the corresponding formula for spatial mechanisms [8], we have $n = 3$, $p_5 = 2$, $p_4 = 0$, $p_3 = 2$, $p_2 = 0$, $p_1 = 0$, which allow to determine the degree of freedom of the driving mechanism:

$$\begin{aligned} W &= 6 \cdot n - 5 \cdot p_5 - 4 \cdot p_4 - 3 \cdot p_3 - 3 \cdot p_2 - p_1 = \\ &= 6 \cdot 3 - 5 \cdot 2 - 4 \cdot 0 - 3 \cdot 2 - 3 \cdot 0 - 0 = 2 = 1 + W_m. \end{aligned} \tag{1}$$

It should be noted that the basic degree of freedom of the driving mechanism is equal to $W = 1$. The so-called local degree of freedom [8], which involves the possibility of rotation of the coupler 2 around its own axis $W_m = 1$ (due to the presence of two spherical hinges) does not affect the overall movability of the door opening mechanism.

The second mechanism (O_3DCO_2) is a plane mechanism consisting of three movable links (Fig. 2, b): the rocker 3, the coupler 4, the rocker 5, and four single-motion turning kinematic pairs (kinematic pairs of the fifth order): O_2, O_3, *C, D*. The mechanism does not contain higher (two-motion) kinematic pairs. Thus, in accordance with the Chebyshev formula for plane mechanisms [8], we have $n = 5$, $p_5 = 4$, $p_4 = 0$, which allows to determine the degree of freedom of the guiding mechanism:

$$W = 3 \cdot n - 2 \cdot p_5 - p_4 = 3 \cdot 3 - 2 \cdot 4 - 0 = 1. \tag{2}$$

Consequently, we can state that the leaning-and-shunting door mechanism of electric public vehicle consisting of two single-DOF driving and guiding mechanisms has one degree of freedom, and therefore, to describe the motion of all its links, it is enough to accept one generalized coordinate. As the generalized coordinate, we will adopt the turning angle of the driving geared motor shaft (Fig. 2, a). According to the kinematic diagram (Fig. 2, b), the generalized coordinate represents the turning angle of the crank O_1A.

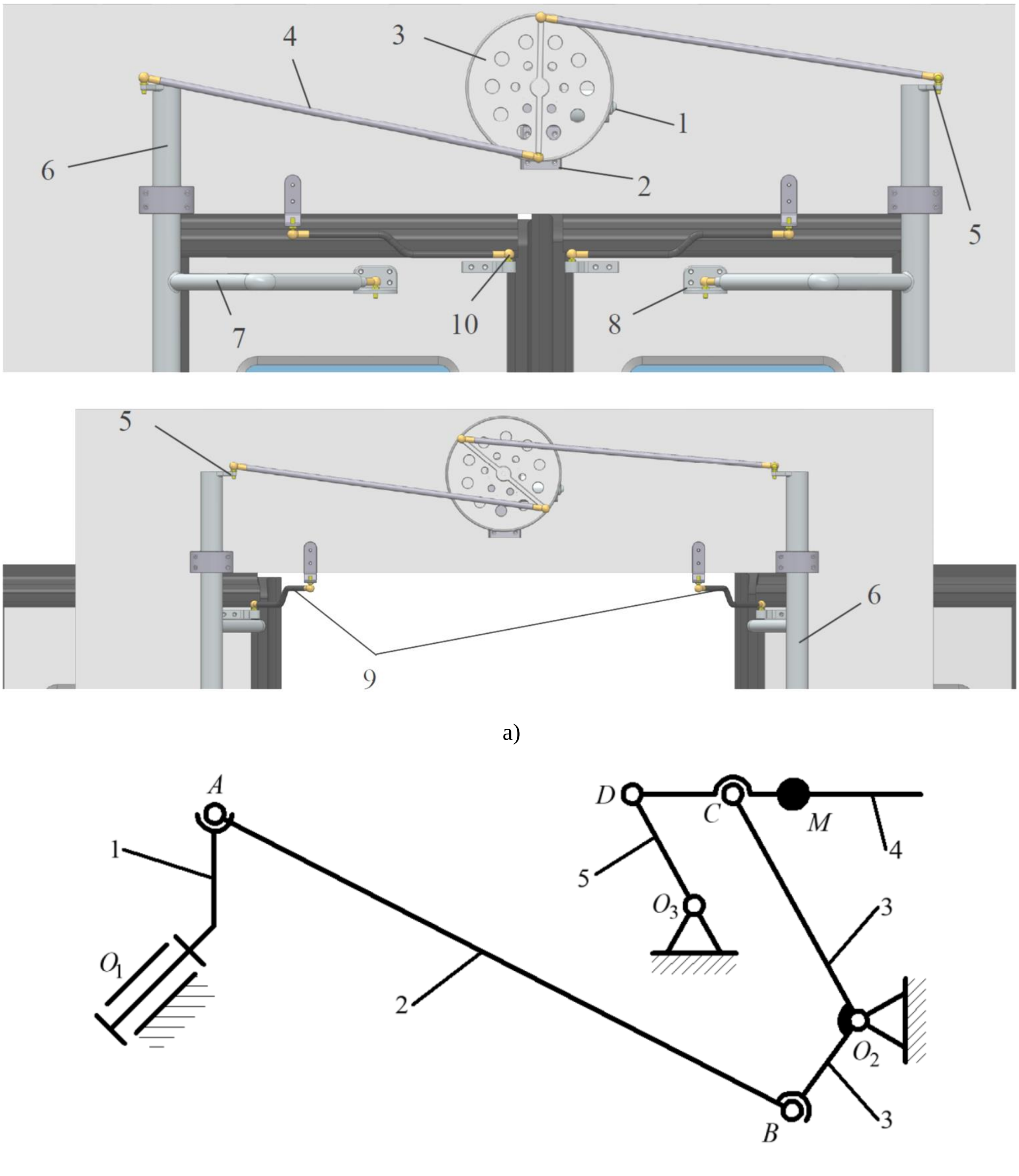


Figure 2. Design (a) and kinematic (b) diagram of the leaning-and-shunting door mechanism of public transport

Let us deduce the formula of the mechanism structure. To do this, we have to divide it into Assur's groups (Norton, 2004). As an input (driving) link we accept the crank 1 that can rotate around the hinge O_1. In this case, the crank 1 is considered as a mechanism of the 1st class. The remaining kinematic chain can be divided into two groups of the 2nd class of the first type (links 2−3, links 4−5). Thus, the formula of the mechanism structure is following:

$$Mech = \mathrm{I}(0,1) \rightarrow \mathrm{II}(2,3) \rightarrow \mathrm{II}(4,5)\,. \qquad (3)$$

In order to perform kinematic analysis of spatial mechanisms of the second class, it is expedient to use the method of closed vector loops [8]. To derive the equations of the mechanism links motion, we arbitrary divide it into four triangles: O_1AB, O_2AB, O_2BC, and CO_3D (Fig. 2, b). In the first two triangles, we know the coordinates of the fixed cylindrical hinge O_2 – x_{O_2}, y_{O_2}, the lengths of the rockers O_1A and O_2B – l_{O_1A}, l_{O_2B}, the length of the coupler AB – l_{AB}, and the adopted generalized coordinate (the angle of the coupler O_1A turning relative to the vertical axis) – φ_1. If we accept the centre of the coordinate system at the point O_1 ($x_{O_1}=0$, $y_{O_1}=0$, $z_{O_1}=0$) and direct the coordinate axes in the following way: O_1z – vertically up, O_1x – horizontally and parallelly to the body to the left, O_1y – horizontally and perpendicularly to the body inside the vehicle's cabin, then the coordinates of the hinge A can be determined by the formulas:

$$x_A = l_{O_1A}\cdot\sin\varphi_1;\quad y_A=0;\quad z_A = l_{O_1A}\cdot\cos\varphi_1. \tag{4}$$

To determine the abscissa of the hinge B, taking into account the closed vector loops of the triangles O_1AB and O_2AB, we can use the following dependencies:

$$x_B = x_A-\sqrt{l_{AB}^2-\left(z_A-z_{O_2}\right)^2-y_B^2};\qquad x_B = x_{O_2}+\sqrt{l_{O_2B}^2-\left(y_B-y_{O_2}\right)}. \tag{5}$$

Equating two expressions (5), we obtain one equation with one unknown value y_B:

$$x_A-\sqrt{l_{AB}^2-\left(z_A-z_{O_2}\right)^2-y_B^2} = x_{O_2}+\sqrt{l_{O_2B}^2-\left(y_B-y_{O_2}\right)}. \tag{6}$$

The solution of equation (6) has the following form:

$$y_B=\frac{\left[\begin{array}{c} y_{O_2}\cdot\left(l_{AB}^2-l_{O_2B}^2-\left(z_A-z_{O_2}\right)^2+\left(x_A-x_{O_2}\right)^2+y_{O_2}^2\right)+\left(x_A-x_{O_2}\right)\times \\ \times\sqrt{\left(\begin{array}{l}\left[\left(l_{O_2B}+\sqrt{l_{AB}^2-\left(z_A-z_{O_2}\right)^2}\right)^2-\left(x_A-x_{O_2}\right)^2-y_{O_2}^2\right]\times \\ \times\left[-\left(l_{O_2B}-\sqrt{l_{AB}^2-\left(z_A-z_{O_2}\right)^2}\right)^2+\left(x_A-x_{O_2}\right)^2+y_{O_2}^2\right]\end{array}\right)}\end{array}\right]}{2\cdot\left(\left(x_A-x_{O_2}\right)^2+y_{O_2}^2\right)}. \tag{7}$$

Thus, we obtained the analytical expression (7) for describing the ordinate of the hinge B as a function of the generalized coordinate φ_1, since the other parameters in the formula (7) are known. If y_B is determined, the abscissa of the hinge B can be calculated by the formulas (5).

Let us consider the closed vector loop which is described by the triangle O_2BC (Fig. 2, b). In this triangle, we know that the coordinates of the fixed cylindrical hinge

$O_2 - x_{O_2}$, y_{O_2}, of the hinge $B - x_B$, y_B, and the lengths of the sides $O_2B - l_{O_2B}$, $BC - l_{BC}$ and $O_2C - l_{O_2C}$. The horizontal coordinate of the hinge *C* can be determined by the formulas:

$$x_C = x_{O_2} + \sqrt{l_{O_2C}^2 - \left(y_C - y_{O_2}\right)^2}; \qquad x_C = x_B + \sqrt{l_{BC}^2 - \left(y_C - y_B\right)^2}. \tag{8}$$

Equating two expressions (8), we obtain one equation with one unknown value y_C:

$$x_{O_2} + \sqrt{l_{O_2C}^2 - \left(y_C - y_{O_2}\right)^2} = x_B + \sqrt{l_{BC}^2 - \left(y_C - y_B\right)^2}. \tag{9}$$

The solution of equation (9) has the following form:

$$y_C = \frac{\left[\begin{array}{l} \left(y_{O_2} - y_B\right)\cdot\left(l_{BC}^2 - l_{O_2C}^2 - y_B^2 + y_{O_2}^2\right) + \left(y_{O_2} + y_B\right)\cdot\left(x_{O_2} - x_B\right)^2 + \\ + \left(x_{O_2} - x_B\right)\cdot\sqrt{\left(\begin{array}{l} \left[\left(l_{BC} + l_{O_2C}\right)^2 - \left(x_{O_2} - x_B\right)^2 - \left(y_{O_2} + y_B\right)^2\right]\times \\ \times\left[-\left(l_{BC} - l_{O_2C}\right)^2 + \left(x_{O_2} - x_B\right)^2 + \left(y_{O_2} - y_B\right)^2\right] \end{array}\right)} \end{array}\right]}{2\cdot\left(\left(x_{O_2} - x_B\right)^2 + \left(y_{O_2} - y_B\right)^2\right)}. \tag{10}$$

Thus, we obtained the analytical expression (10) for describing the ordinate of the hinge *C* as a function of the generalized coordinate φ_1, since the other parameters in formula (10) are known. If y_C is determined, the abscissa of the hinge *C* can be calculated by the formulas (8).

Let us consider the closed vector loop which is described by the triangle CO_3D (Fig. 2, b). In this triangle, we know the coordinates of the fixed cylindrical hinge $O_3 - x_{O_3}$, y_{O_3}, the hinge $C - x_C$, y_C, the lengths of the links $CD - l_{CD}$ and $O_3D - l_{O_3D}$. In this case, the horizontal coordinate of the hinge *D* can be determined by the formulas:

$$x_D = x_C + \sqrt{l_{CD}^2 - \left(y_C - y_D\right)^2}; \qquad x_D = x_{O_3} + \sqrt{l_{O_3D}^2 - \left(y_{O_3} - y_D\right)^2}. \tag{11}$$

Equating two expressions (11), we obtain one equation with one unknown value y_D:

$$x_C + \sqrt{l_{CD}^2 - \left(y_C - y_D\right)^2} = x_{O_3} + \sqrt{l_{O_3D}^2 - \left(y_{O_3} - y_D\right)^2}. \tag{12}$$

The solution of equation (12) has the following form:

$$y_D = \frac{\left[\begin{gathered}\left(y_{O_3}-y_C\right)\cdot\left(l_{O_3D}^2-l_{CD}^2-y_C^2+y_{O_3}^2\right)+\left(y_{O_3}+y_C\right)\cdot\left(x_{O_3}-x_C\right)^2+\\ +\left(x_{O_3}-x_C\right)\cdot\sqrt{\left(\begin{gathered}\left[\left(l_{O_3D}+l_{CD}\right)^2-\left(x_{O_3}-x_C\right)^2-\left(y_{O_3}-y_C\right)^2\right]\times\\ \times\left[-\left(l_{O_3D}-l_{CD}\right)^2+\left(x_{O_3}-x_C\right)^2+\left(y_{O_3}-y_C\right)^2\right]\end{gathered}\right)}\end{gathered}\right]}{2\cdot\left(\left(x_{O_3}-x_C\right)^2+\left(y_{O_3}-y_C\right)^2\right)}. \tag{13}$$

Thus, we obtained the analytic expression (13) for describing the ordinate of the hinge D as a function of the generalized coordinate φ_1, since the other parameters in formula (13) are known. If y_D is determined, the abscissa of the hinge D can be calculated by the formulas (11).

If the coordinates of joints C and D are known, we can determine the coordinates of the point M (mass centre of the door leaf) using the following relationships:

$$\begin{gathered}\frac{y_C-y_D}{l_{CD}}=\frac{y_M-y_D}{l_{MD}}\Rightarrow y_M=y_D+\left(y_C-y_D\right)\cdot\frac{l_{MD}}{l_{CD}};\\ \frac{x_C-x_D}{l_{CD}}=\frac{x_M-x_D}{l_{MD}}\Rightarrow x_M=x_D+\left(x_C-x_D\right)\cdot\frac{l_{MD}}{l_{CD}}.\end{gathered} \tag{14}$$

Numerical modelling and simulation of the door mechanisms motion. In order to analyse the adequacy of the analytical relationships for description of the door mechanism motion, let us investigate the displacements of the corresponding links. This stage of research is carried out on the basis of the derived analytical dependencies in MathCAD software and using the solid-state model of the door mechanism in SolidWorks software [9].

On the basis of the proposed design of the door mechanism developed in SolidWorks software, let us write the input data for investigating the motion of the mechanism, in particular, its geometric parameters (Fig. 2, b): $L_{O_2B}=60$ mm, $l_{CD}=242$ mm, $l_{O_1A}=128$ mm, $z_{O_2}=23$ mm, $l_{BC}=406$ mm, $y_{O_2}=-53$ mm, $l_{O_3D}=373$ mm, $l_{AB}=722$ mm, $x_{O_3}=452$ mm, $l_{O_2C}=363$ mm, $y_{O_3}=64$ mm.

Based on the formulas (4−14), taking into account that $l_{MD}=277$ mm, let us plot the graphical dependences of the displacements of the corresponding hinges and of the mass centre of the door leaf on the magnitude of the turning angle φ_1 of the driving rocker (Fig. 3).

To confirm the adequacy of the obtained analytical dependencies, let us present the results of motion simulation of the mechanism received in SolidWorks software (Fig. 4) [9]. During the process of simulation, a constant speed of rotation of the electric geared motor shaft was accepted. This provides a 2 seconds duration of the door closing cycle. Thus, the generalized coordinate varies from the minimum value $\varphi_{1\min}=0$ to the maximum value $\varphi_{1\max}=48°=0.838$ rad in accordance with the linear dependence on time.

Analysing the graphic dependences presented in Figs. 3−4 obtained on the basis of analytical calculation and virtual experiment, respectively, we can state about their satisfactory agreement (convergence), which justifies the adequacy of the derived equations of motion of the leaning-and-shunting door opening mechanism.

Graphical dependencies *b, c, d* in Figs. 3−4 make it possible to draw a conclusion on the translational motion of the door leaf relative to the body of the vehicle, since the ordinates of the three points of the door leaf change according to the similar laws. In the other words, during the processes of the door opening/closing, the door leaf planes do not rotate in relation to the vehicle body remaining parallel to it.

On the basis of the numerical modelling and computer simulation of the motion of the passenger electric transport door, it was possible to substantiate the adequacy of the analytical dependences previously derived to describe the movement of the links of the corresponding mechanisms. These dependences will be further used in the analysis of energy consumptions taking place during the process of opening/closing of the doors.

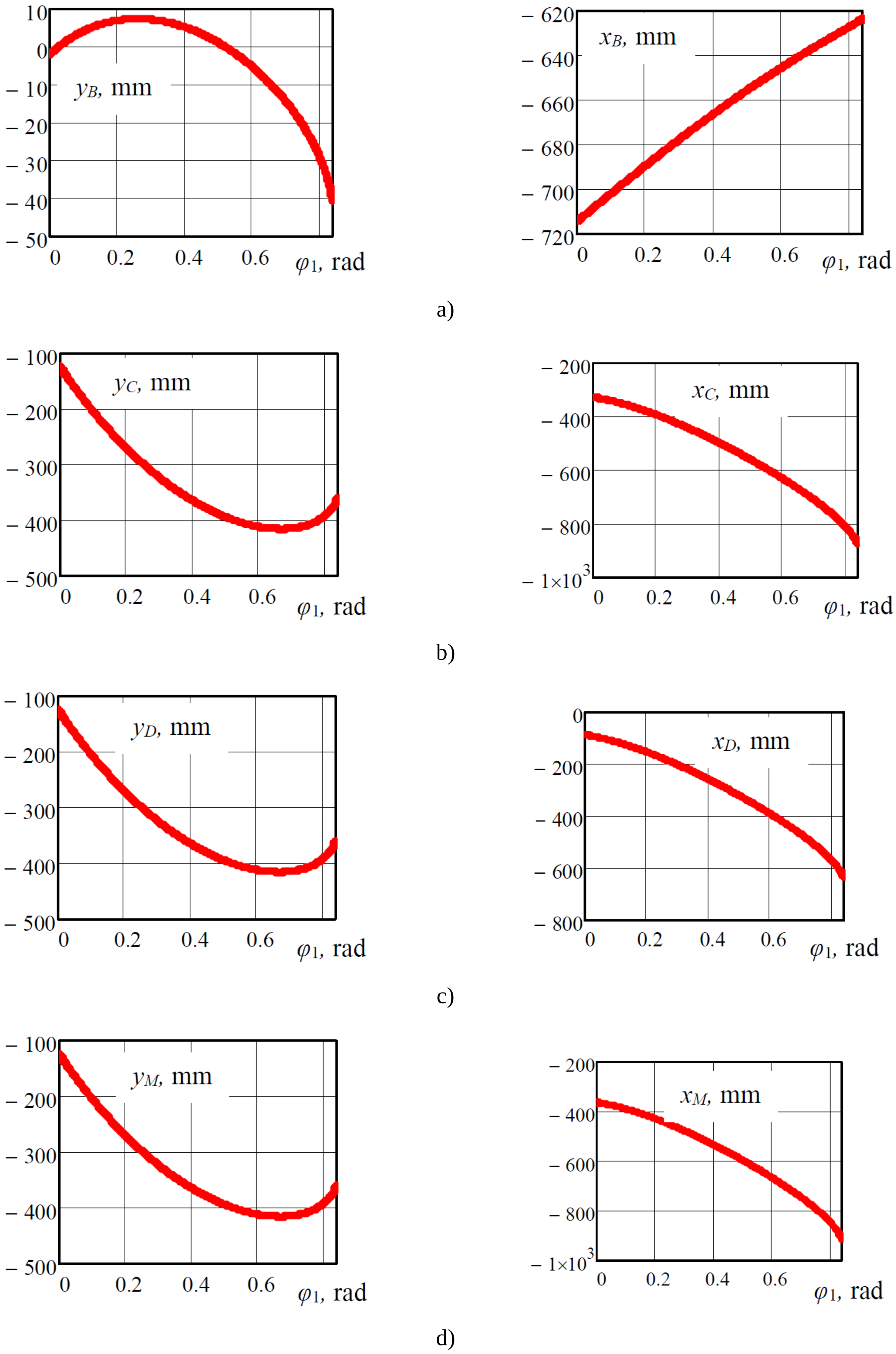
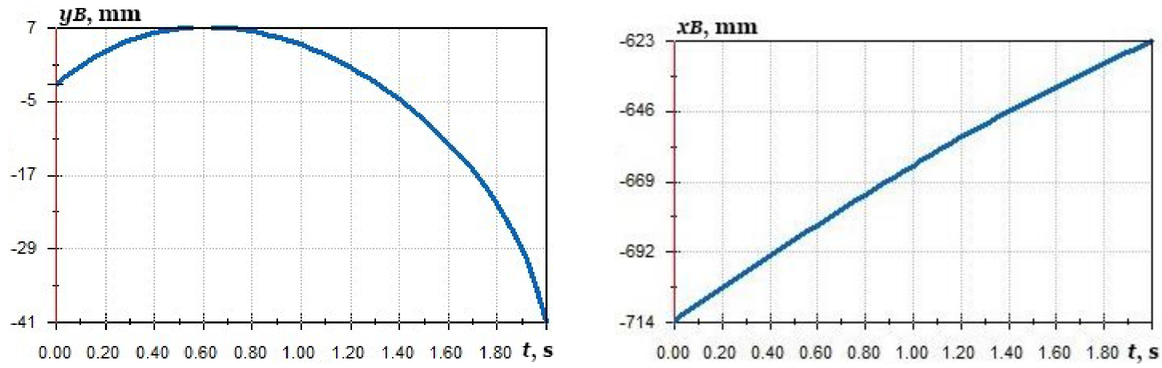
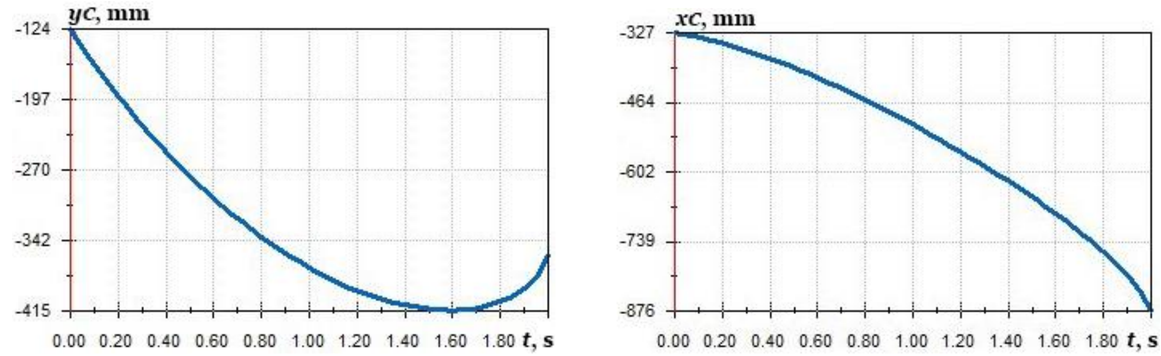


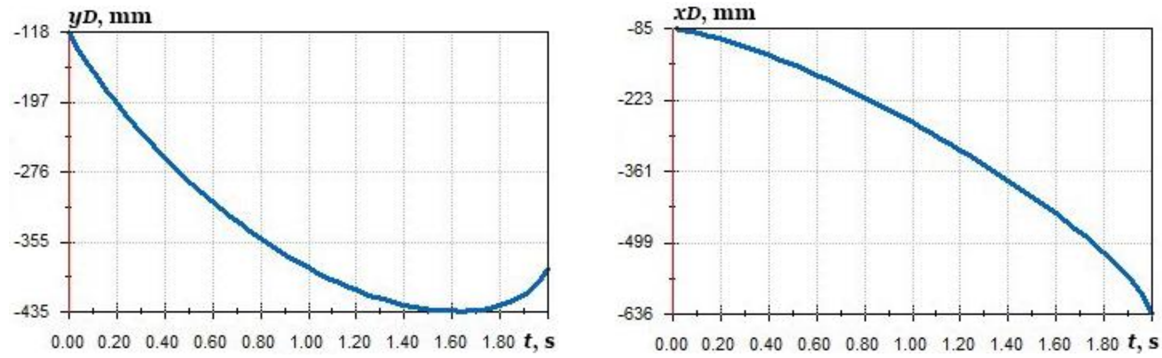
Figure 3. Results of numerical modelling of motion of the leaning-and-shunting mechanism: a – hinge *A*, b – hinge *C*, c – hinge *D*, d – mass centre of the door leaf



a)



b)



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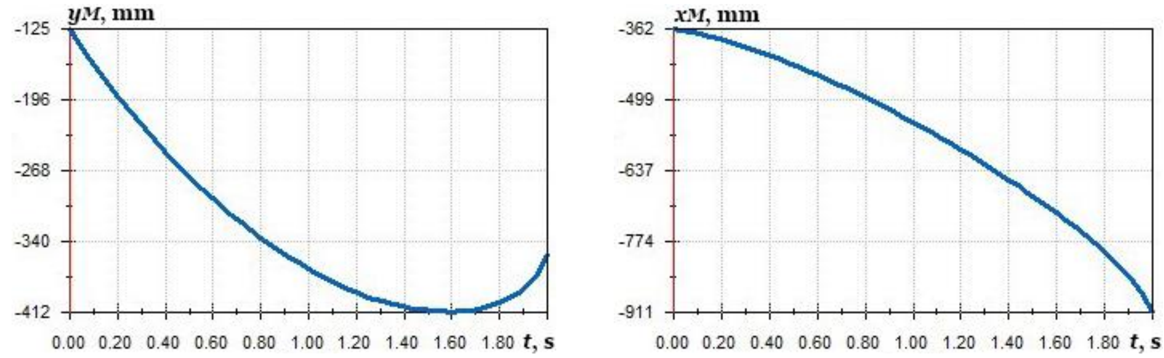


d)

Figure 4. Results of motion simulation of the leaning-and-shunting mechanism: a – hinge *A*, b – hinge *C*, c – hinge *D*, d – mass centre of the door leaf

Analysis of energy efficiency of the mechanism. The leaning-and-shunting door mechanism of electric public transport (Figs. 1–2) is currently considered as one of the safest and most comfortable mechanisms, since the doors do not occupy extra space in the vehicle cabin, provide high-tightness closing and high convenience of passengers boarding/disembarking. However, among its disadvantages we can mention the necessity to use the drive with a significant power store, when it is required to close the doors in the case of overcrowded passenger cabin.

Let us consider the problem of providing the necessary power of the mechanism drive while closing the doors of the electric public transport in the presence of obstacles on their way. The obstacles are simulated in the form of a constant forces applied to the ends of the door leaves and directed oppositely to the direction of their closing. The value of the resistance force is $F_r = 100\ \text{N}$. To find the nominal power of the drive, which is required to overcome the given force, taking into account the specified duration of the door closing cycle $T_c = 2\ \text{s}$, we need to determine the speed of the guide hinge *D* parallel to the body of the vehicle, that is, the speed $\dot{x}_D$. To find this speed, it is necessary to differentiate the dependency (11) considering the function $\dot{x}_D$ as a complex function, which depends on the turning angle of the drive shaft of the electrical geared motor, which in turn is a function of time. Therefore, the general expression for calculating of the speed $\dot{x}_D$ is following:

$$\dot{x}_D(t) = \frac{d\left(x_D(\varphi_1)\right)}{d(\varphi_1)} \cdot \frac{d\left(\varphi_1(t)\right)}{d(t)}. \tag{15}$$

As mentioned above, the speed of turning of the geared motor shaft, which provide a two-second cycle of the door closing, is constant:

$$\frac{d\left(\varphi_1(t)\right)}{d(t)} = const = 24\ \text{deg/s} = 0,419\ \text{rad/s}. \tag{16}$$

Considering the expression (16), we can state that:

$$\varphi_1(t) = 0,419 \cdot t\ (\text{rad}). \tag{17}$$

Taking into account the awkwardness of the derivative expression $\dfrac{d\left(x_D(\varphi_1)\right)}{d(\varphi_1)}$, it is inexpedient to present this expression in the paper, but it has been deduced and accepted for modelling with a help of MathCAD software.

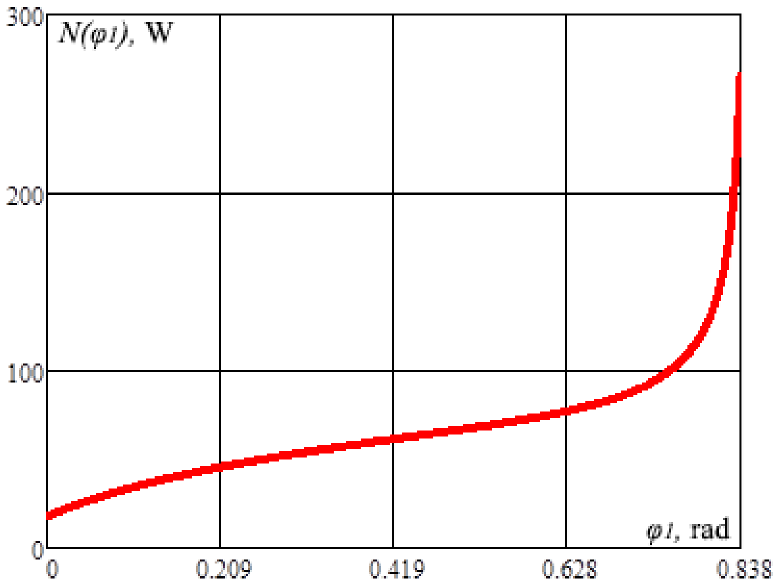
Adopting the resistance force while closing the doors $F_r = 100\ \text{N}$ and calculating the corresponding speed of the door motion (formula 15), we can determine the dependence of the nominal power of the doors drive on the corresponding generalized coordinate:

$$N(l\varphi_1) = F_{on} \cdot \frac{d\left(x_D(\varphi_1)\right)}{d(\varphi_1)}, \tag{18}$$

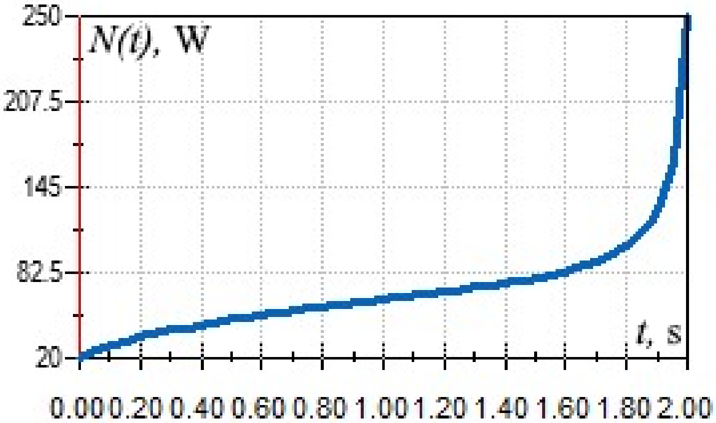
and time dependence of changing the nominal power of the doors drive during the process of their closing under the action of constant resistance force:

$$N(t) = F_{on} \cdot \frac{d\left(x_D\left(\varphi_1\right)\right)}{d\left(\varphi_1\right)} \cdot \frac{d\left(\varphi_1\left(t\right)\right)}{d\left(t\right)}. \quad (19)$$

The results of calculations performed by the formula (18) are shown in Fig. 5, a, and the results of the virtual experiment carried out in SolidWorks software are presented in Fig. 5, b [9].



a)



b)

Figure 5. Results of analytical and experimental analysis of the nominal power of the door opening mechanism drive of electric public transport: a – obtained on the basis of the derived analytical dependencies in MathCAD software; b – obtained by means of the mechanism motion simulation in SolidWorks software

Analysing the obtained graphical dependences (Fig. 5), for the constant resistance force opposing the motion of the door leaves $F_r = 100$ N and for the constant duration of the door closing cycle $T_c = 2$ s, we can state that the required nominal power of the electric drive is approximately 250 W and is more than twice smaller than the required nominal power of the pneumatic drive of the leaning-and-shunting doors that are widely used in modern electric public transport [1−3].

Conclusions. The design and operational peculiarities of the leaning-and-shunting door mechanism of electric public transport were considered. On the basis of the constructed simplified diagram of the mechanism, its structural and kinematic analysis was performed. As a result of the carried out analysis, the analytical dependencies for describing the motion of the mechanism links during the door motion were derived. Using the obtained expressions, the main kinematic parameters of the studied mechanism were investigated in MathCAD software. The virtual experiment was carried out by means of simulating the motion of the solid-state model of the mechanism designed in SolidWorks software. The results of theoretical investigations and of the virtual experiment justified the adequacy of the derived analytical expressions. The energy efficiency of the investigated leaning-and-shunting door opening mechanism of electric public transport was analyzed. While performing energy efficiency analysis, the resistance force acting on the door leaves was prescribed ($F_r = 100$ N) and the corresponding door motion speed during its closing was calculated. Based on the obtained results the dependency of the nominal power supply of the mechanism drive as a function of corresponding generalized coordinate was deduced. Comparing the obtained results with the investigations presented in modern scientific publications, it was established that the necessary nominal power of the leaning-and-shunting mechanism drive more than twice smaller than the required nominal power of the pneumatic drive of the leaning-and-shunting doors that are widely used in modern electric public transport. Further investigations on the subject of the paper can be carried out in the direction of developing different control systems providing safe and reliable operation of the considered door opening mechanism driven be electric and pneumatic actuators.

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АНАЛІЗ КІНЕМАТИЧНИХ І СИЛОВИХ ПАРАМЕТРІВ ПРИТУЛЬНО-ЗСУВНОГО МЕХАНІЗМУ ВІДКРИВАННЯ ДВЕРЕЙ ПАСАЖИРСЬКОГО ЕЛЕКТРОТРАНСПОРТУ

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Резюме. *Одна із найскладніших проблем, які виникають у процесі проектування транспортних засобів для перевезення пасажирів, полягає в забезпеченні енергоефективного, комфортного і безпечного процесу відкривання/закривання дверей. Основні методи проведення досліджень за даною тематикою передбачають проведення структурного й кінематичного аналізу дверних механізмів із подальшим імітаційним моделюванням їхнього руху в прикладних програмних продуктах. У даній статті проаналізовано конструктивні та функціональні особливості притульно-зсувного механізму відкривання дверей пасажирського електротранспорту. Запропоновано відповідне конструктивне виконання механізму з використанням електричного привода та побудовано його спрощену розрахункову схему. Проведено структурний і кінематичний аналіз механізму та виведено аналітичні залежності для описування руху його ланок у процесі відкривання/закривання дверей. Досліджено основні кінематичні параметри запропонованого механізму на основі виведених аналітичних залежностей у програмному продукті MathCAD та з використанням його твердотільної моделі, розробленої у середовищі SolidWorks Motion. Зроблено висновки про узгодження результатів теоретичних досліджень, проведених на основі чисельного розв'язування отриманих рівнянь руху, та віртуального експерименту (імітаційного моделювання у програмному продукті SolidWorks). Проведено аналіз енергоефективності досліджуваного притульно-зсувного механізму відкривання дверей пасажирського електротранспорту. Задаючи зусилля опору закривання дверної стулки та розрахувавши відповідну швидкість руху дверей у процесі їх закривання, встановлено залежність номінальної потужності електричного привода механізму як функції відповідних узагальнених координат. Аналізуючи отримані результати, встановлено, що необхідна номінальна потужність привода притульно-зсувного механізму практично вдвічі менша, ніж необхідна номінальна потужність привода дверей, в яких використовується поширений на даний час поворотно-зсувний механізм. Подальші дослідження за тематикою даної статті можуть бути проведені в напрямку розроблення чи удосконалення систем керування дверними механізмами, побудованими за запропонованою структурою, з метою їх енергоефективного, комфортного й безпечного функціонування в громадському електротранспорті.*

Ключові слова: *притульно-зсувний механізм, пасажирський електротранспорт, структурний аналіз, кінематичний аналіз, енергетична ефективність.*

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