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ASSESSING THE ACCURACY OF MODELING THE TUBBING ERECTOR MANIPULATOR MECHANISM IN SOLIDWORKS MOTION PROGRAM

Purpose. To estimate the calculation error in determining the kinematic and dynamic characteristics of the UT62 tubbing erector manipulator mechanism movement.

Methodology. Modern computer analysis methods are used, which are implemented in a number of computing complexes. Due to the task complexity, a mathematical model for the manipulator mechanism fragment, which is a hydraulic cylinder with two degrees of freedom, has been developed to estimate the error of computer modeling. This model is used as a test model.

Findings. On the example of the model study, it is shown that when simulating the mechanism movement, errors in the calculations are possible due to incorrect problem formulation, as well as to the inaccurate settings of the modules for converting the initial data. Under these circumstances, it has been concluded that when the input link movement is specified as a vector, then when determining the kinematic and dynamic characteristics of the mechanism, it is necessary to use a cubic spline in the "interpolator" module.

Originality. For the first time, the direct problem of the manipulator mechanism dynamics has been solved, which consists in determining the static and dynamic characteristics of the device according to the given motion law of its drives. It is noted that when the motion is specified in the form of smooth analytical functions, then there is a complete coincidence of the calculation results performed by the SOLIDWORKS MOTION program with those obtained by mathematical modeling. In addition, when the mechanism link input movement is modeled as a vector, which is formed from a discontinuous function, then the cubic spline used in the "interpolator" module provides smooth harmonic functions of the movement, acceleration and jerk processes. As a result of modeling the manipulator mechanism parameters, it turned out that it is not expedient to use more than 50 points of discrete time in the research. Thus, the errors in calculating the maximum power values of the manipulator motors do not exceed 20 % for the power hydraulic cylinder and 5 % for the hydraulic motor.

Practical value. The proposed algorithm can be used to model the movement of complex mechanisms in machines.

Keywords: *SOLIDWORKS, SOLIDWORKS MOTION, tubbing erector manipulator, discrete time, cubic spline Akima*

Literature review. In the conditions of modern market relations, most of the technical objects that are produced by domestic enterprises are inferior to foreign analogues in their quality, reliability, and they have a higher cost price. In order to improve them, scientists of Dnipro University of Technology and M. S. Polyakov Institute of Geotechnical Mechanics are developing the latest methods for mathematical and computer modeling of technical objects of any degree of complexity. Thus, under the guidance of Professor I. A. Taran, for the first time, the parameters of new designs of hydro-mechanical transmissions of mine diesel locomotives [1] and the principles of modeling transport routes have been determined and substantiated [2, 3]. Representatives of the scientific school of Professor V. P. Naduty and his doctoral student V. V. Sukharyev study the stress-strain state of vibrating feeders influenced by impact loads [4]. Under the guidance of Academician G. G. Pivniak and Professor V. I. Samus, a complex of studies on the use of heat pumps at mining enterprises has been conducted [5, 6]. Substantiation of the parameters for new designs of mine locomotives was a key topic of research and development works by Professors K. A. Ziborov [7, 8] and V. V. Protsiv [9, 10]. Of great practical importance are the results of scientific research in the field of mechanics of new designs of mine hoisting machines [11, 12]. In particular, the dependences of the dynamic characteristics on the parameters of the technical condition of the hoisting plant individual nodes have been determined [13].

Introduction. By order of JSC Dniprovazhmash, representatives of the scientific school under the guidance of Professor K. S. Zabolotnyi are studying the dynamic and static parameters of tubbing erectors. It should be noted here that tubbing erectors are used in tunneling. They are machines that reinforce horizontal mine workings using metal tubbings and reinforced concrete blocks.

Literature sources [14, 15] present the analysis of technical and technological parameters of existing tubbing erectors: lever with one and two levers, annular on internal and external supports, cassette, arc with a fixed and swinging arc, arrow-shaped. As evidenced by the research results, the proposed designs of tubbing erectors have increased metal consumption and drive power. This is conditioned by the fact that at present there is no scientific-based method for determining the rational parameters of such machines. It should be noted that the development of such a method is complicated, since during operation, a stress-strain state occurs in the parts and nodes of the tubbing erector mechanism, which varies over a wide range [14]. For this reason, the analytical methods proposed in the literature [14, 15] do not make it possible to determine the internal forces in the manipulator nodes with sufficient accuracy.

The scientific idea of this research is to use a combination of modern computer analysis methods operating in a number of computing complexes developed by Dassault Systèmes SOLIDWORKS Corp.

Dnipro University of Technology has in its use a license to work with SOLIDWORKS Education Edition software package, which includes software applications for performing calculations of dynamic [16] and static [17, 18] loads that occur in machine elements.

The purpose of this research is to substantiate the application of the computer analysis method of the kinematic and dynamic characteristics of the manipulator mechanism movement in the process of erecting tubbings. It is necessary to verify the method during operation of the UT62 tubbing erector manipulator mechanism, which has the most complex design compared to other tubbing erector types.

Problem statement. The tubbing erector manipulator mechanism is a system of elements with several degrees of freedom (Fig. 1). Here, the links of Lever arm *2*, Shoulder *3* and Section *4* are connected to the hoisting device using rotational kinematic pairs. The latter performs the function of assembling the Pin *5* tubbings in the lining tunnel. Lever arm *2* link with counter-weight *7* is set on the drive shaft of the hydraulic motor *1* (Support). Two power hydraulic cylinders *6* (Engine) control the links *3* (Shoulder) and *4* (Section).

In the process of erecting, the manipulator must deliver the tubbing to the specified location of the processing ring assembly along the optimal trajectory of movement, applying minimal force and spending as little time as possible.

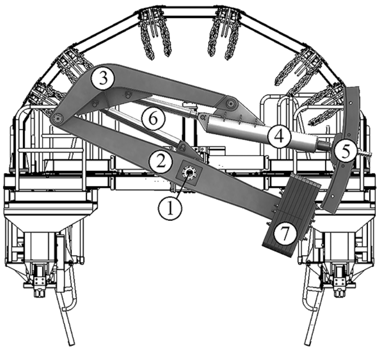


Fig. 1. Tubbing erector manipulator design

The task set is to estimate the error of modeling performed by the SOLIDWORKS MOTION program when solving the direct problem of the dynamics of the tubbing erector manipulator mechanism, which consists in determining the static and dynamic device movement characteristics according to the given motion law of its drives.

The error is the difference between the modeling result obtained in SOLIDWORKS, SOLIDWORKS MOTION program, and the values of modeling the quantities found analytically.

In the manipulator mechanism (Fig. 2, *a*), we distinguish its fragment (link) – a hydraulic cylinder, which has a body with a fixed hinge (Rocker) and with a fixed axis (Shaft), and a rod (Slider) (Fig. 2, *b*). This fragment has two degrees of freedom. The slider performs a complex displacement, in which the relative motion is concentrated in relation to the rocker and the translational motion together with the rocker.

Analytical solution. The calculation scheme of the model problem is shown in Fig. 2, *a*, where the following designations are used: M is a material point representing the slider; $\varphi(t)$ – the rocker rotation angle; $r(t)$ – the slider relative motion; L – the distance from the rotation axis to the center of the slider mass in its lowermost position.

The translational motion law of the rocker is specified in the following form

$$\varphi(t) = \pi(1 - \cos\omega_1 t), \quad (1)$$

where ω_1 is cyclic frequency of the rocker oscillations during the translational motion.

The relative motion law of the slider is written in the form of the following dependence

$$r(t) = L + h(1 - \cos\omega_2 t), \quad (2)$$

where h is the amplitude of oscillatory motion with cyclic frequency ω_2 during relative motion.

By differentiating these formulas according to the time parameter, expressions are obtained to determine the following characteristics:

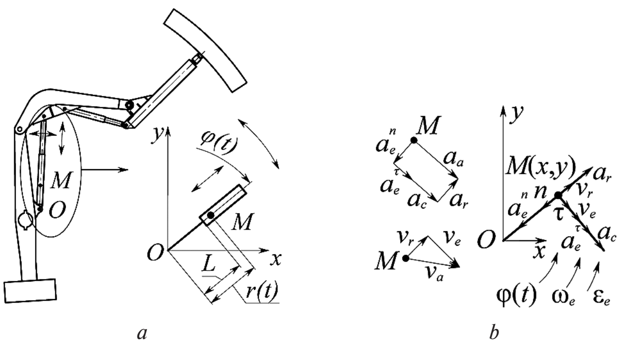


Fig. 2. Calculation scheme of the model problem & velocity and acceleration plans during the motion of a mechanism fragment

- the angular velocity of the rocker during translational motion

$$\omega_e = \pi\omega_1 \sin\omega_1 t;$$

- the linear velocity of the slider during relative motion

$$v_r = h\omega_2 \sin\omega_2 t;$$

- the angular acceleration of the rocker during translational motion

$$\varepsilon_e = \pi\omega_1^2 \cos\omega_1 t;$$

- the linear acceleration of the slider during relative motion

$$a_r = h\omega_2^2 \cos\omega_2 t;$$

- the linear velocity of the slider during translational motion

$$v_e = r(t)\omega_e.$$

The absolute motion of the slider in the projection onto the Cartesian coordinate axis can be described in the form of the following equations

$$x(t) = -r(t)\sin\varphi(t); \quad y(t) = r(t)\cos\varphi(t).$$

Then the absolute values of the slider velocity in the projection on the coordinate axis are

$$\vec{v}_a = \vec{v}_x + \vec{v}_y,$$

here $v_x = -v_r \sin\varphi - v_e \cos\varphi$; $v_x = -v_r \cos\varphi - v_e \sin\varphi$ (Fig. 2, *b*).

Absolute acceleration values are

$$\vec{a}_a = \vec{a}_x + \vec{a}_y,$$

here $a_x(t) = -a_n(t)\sin\varphi(t) + a_t(t)\cos\varphi(t)$; $a_y(t) = a_n(t)\cos\varphi(t) + a_t(t)\sin\varphi(t)$; $a_n(t) = a_r + a_e^n = h\omega_2^2\cos\omega_2 t - r(t)\omega_1^2$ (projection of the slider absolute acceleration on the radial axis of the rocker); $a_\tau(t) = 2v_r\omega_1 + r(t)\varepsilon_e$ (the tangential component of this acceleration); $a_c(t) = 2v_r\omega_1$ (the Coriolis acceleration).

After determining the kinematic characteristics of the movement of the mechanism elements, other parameters can be found, such as:

- motor torque $M_M(t) = -ma_\tau(t)r(t) + J_o\varepsilon(t)$;
- forces in the power hydraulic cylinder $F_M(t) = ma_n(t)$;
- motor power $P_{M_M}(t) = M_M(t)\omega(t)$;
- power of the power hydraulic cylinder $P_{F_M}(t) = F_M(t)v(t)$.

The kinematic and dynamic characteristics of the movement of the mechanism elements are calculated using the following initial data: slider mass $M = 1.25$ kg; the rocker inertia moment relative to the axis of rotation $J_o = 25$ kg · m^2; the distance from the rotation axis to the slider mass center in its lowermost position $L = 220$ mm; rotational motion frequency $\omega_1 = 0.4\pi$ s^{-1}; translational motion frequency $\omega_2 = 2\pi$ s^{-1}; the time of analysis $T_F = 5$ s.

The dependency graphs of the kinematic and dynamic parameters of the mechanism movement on time according to the above formulas are given below in the research.

Computer modeling. Further, the mechanism, whose scheme is shown in Fig. 2, *a*, is modeled in the SOLIDWORKS program using the following components: a fixed axis Shaft *1*, a Rocker *2* and a Slider *3* (Fig. 3).

When developing a computer model, we use the following connections of elements:

1. Concentric Hinge between the two cylindrical faces of the Shaft and Rocker parts, as well as Coincident between the frontal planes of these parts.
2. Coincident between the frontal planes of the Rocker and Slider parts.
3. Coincident between the side butt faces of the Rocker and Slider.

To analyze the movement of mechanism elements, it is necessary to start working in the SOLIDWORKS MOTION program. To do this, the following concepts should be added:

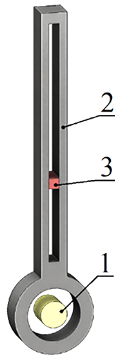


Fig. 3. Computer model of a mechanism fragment

1. Rotary Motor, applied to the cylindrical face of the Rocker part (Fig. 4, *a*).
2. Linear Motor (Actuator), applied to the Slider part (Fig. 4, *b*).

Let us determine the number of degrees of the mechanism fragment movement freedom, given that there are two moving parts with twelve degrees of freedom; one rotational coupling, which prohibits motion with five degrees of freedom; two couplings on planes that prohibit movement with six degrees of freedom; two motors (Rotary and Linear), prohibiting movement with two degrees of freedom. Then, in total, $12 - 5 - 6 - 1 - 1 = -1$, that is, one redundant coupling has been revealed. The SOLIDWORKS MOTION program only models fully defined systems. When creating a computer model in the SOLIDWORKS program, there is no possibility to correct elementary couplings, in particular, the prohibition of the element's relative movement along a specified direction or its rotation around a specified axis. Therefore, the rotation around the *OY* axis entered the Coincident connection twice. In this regard, the program analyzer has removed this redundant coupling from the first specified connection.

Further, the following characteristics of the element movement should be provided: the number of frames per second; integrator type, for example, SI2_GSTIFF. This makes it possible to calculate the displacement, its velocity and acceleration with the necessary accuracy, as well as to determine the maximum number of iterations and the research duration. Other settings can be taken by default.

In the process of computer modeling of the mechanism movement, errors in calculations are possible due to incorrect settings of acting "interpolator" and "integrator" modules.

The SOLIDWORKS MOTION program enables setting at the input of the motor motion law as a function. The results of research performed by this program can be saved as files in the form of two-column matrices using the Export CSV command: the first indicates the frame time; the second – the value displayed on the graph at this time.

Let the translational motion law of the rocker be defined in the form of expression (1), and the relative motion law of the slider is represented by dependence (2).

Figs. 5 and 6 show graphs of time dependence of the following parameters: the rocker rotation angle, the slider relative motion, the projection of the absolute slider displacement on the *x* and *y* axes, the moment and power of the Rotary Motor, the force and power of the Linear Motor (Actuator). In this case, 1 shows curves constructed from the analytical modeling results (solid lines); 2 – curves constructed based on a computer model (dotted lines).

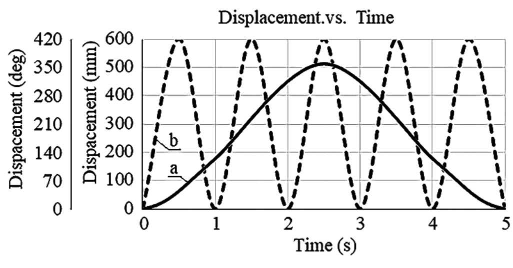


Fig. 4. Curves of the motion law specified to the Rocker (a) and Slider (b) parts of the mechanism fragment

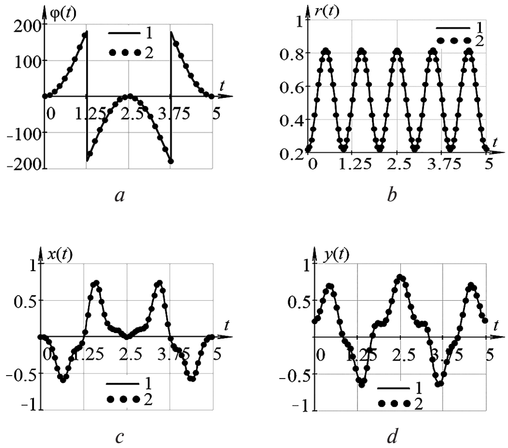


Fig. 5. Graphs of time dependence of the following parameters: the rocker rotation angle (a), the slider relative translational motion (b), the projection of the absolute slider displacement on the x (c) and y (d) axes

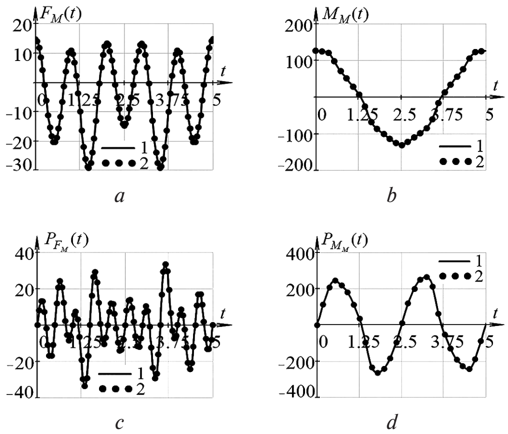


Fig. 6. Graphs of time dependence of the moment (a) and power (c) in the Rotary Motor; the force (b) and power (d) in the Linear Motor (Actuator)

Conclusion: it follows from the above results that if the motion law is determined as smooth analytical functions, then there is a complete coincidence of the results of calculations performed in the SOLIDWORKS MOTION program and those provided by mathematical modeling tools.

When the input element displacement is specified as a vector formed from a discontinuous function, then the program uses the "interpolator" module, which can implement three types of interpolation: linear spline, cubic spline and Akima spline. Then, as a consequence, a function is formed that has infinite values of velocity, acceleration or jerk parameters (Fig. 7), which, naturally, gives erroneous results of the object's motion analysis.

In the following, we will consider only smooth functions of harmonic type. In this case, everything depends on the number of points N in the designation of the discrete time of motion. Thus, consider the case when $N = 10$. From Fig. 8, the continuous curve shows the initial specified motion functions of the Rotary Motor and Linear Motor (Actuator) elements, and the dotted curve shows the result of processing the specified vectors using the interpolator. As can be seen, the resulting rotational motion function differs from the original one insignificantly, while the translational motion function has a significant difference.

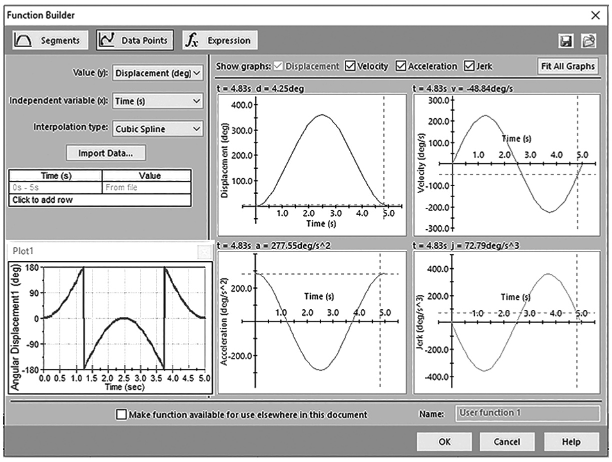
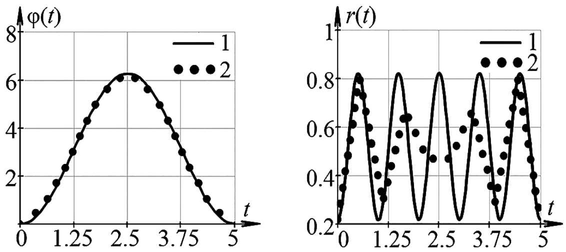


Fig. 7. General view of the interpolation function constructor window



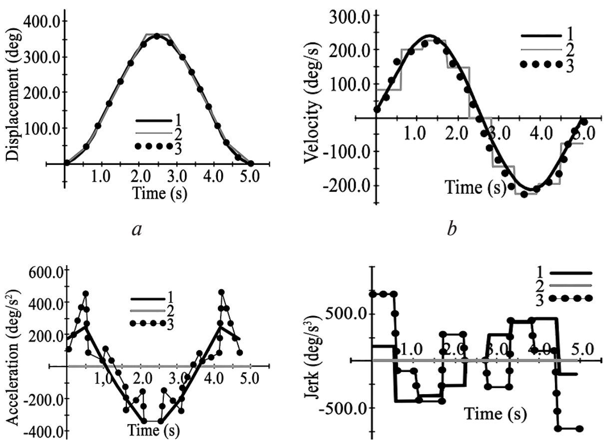
a *b*

Fig. 8. Graphs of dependence of the rocker rotation angle (a) and the slider relative translational motion (b) on time, when N = 10

Now we analyze how this vector is affected by different interpolators of the SOLIDWORKS MOTION program.

Under the action of a cubic spline (curve *1*) and Akima spline (curve *3*), the jerk is described by a piecewise constant function (Fig. 9, *d*). In addition, the use of Akima spline gives a jump in the movement acceleration (Fig. 9, *c*), which leads to shocks during the object's motion. Under the influence of linear interpolation (curve *2*), the velocity jumps (Fig. 9, *b*), which leads to infinite shock pulses.

In general, such a number of *N* discrete time intervals is obviously not enough.



a *b*

c *d*

Fig. 9. Graphs of time dependence of angular displacement (a), velocity (b), acceleration (c) and jerk (d) in a Rotary Motor under the influence of interpolators – cubic spline (curve 1), linear interpolation (curve 2) and Akima spline (curve 3), when N = 10

Consider how the number of discrete time points affects the modeling in the research. To do this, $N = 100$ is set (Fig. 10). In the graph, the continuous curve *1* denotes the initial specified motion functions of the Rotary Motor and Linear Motor (Actuator) elements, and the dotted curve denotes the result of processing the specified vectors using the interpolator.

Calculation of parameters in the SOLIDWORKS MOTION program gives the following results:

- influence of a linear spline (curve *2*) – infinite values of accelerations and jerks (Figs. 11, *c, d*);

- Akima spline (curve *3*) – acceleration jump and jerk (Figs. 11, *c, d*);

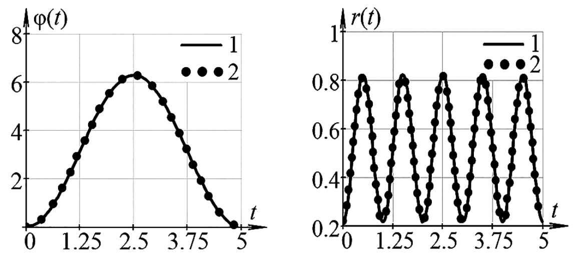
- cubic spline (curve *1*) – smooth harmonic functions that describe the movement, velocity, acceleration and jerk of the object (Figs. 11, *a–d*).

After the calculation performed by the SOLIDWORKS MOTION program using a cubic interpolator, graphs of the change in the power of the Linear and Rotary motors are plotted. Fig. 12 shows a comparison of the values of these parameters with those obtained by mathematical modeling.

Determine the interpolation error. The result inaccuracy is estimated in the form of absolute root-mean-square e and relative b errors, then

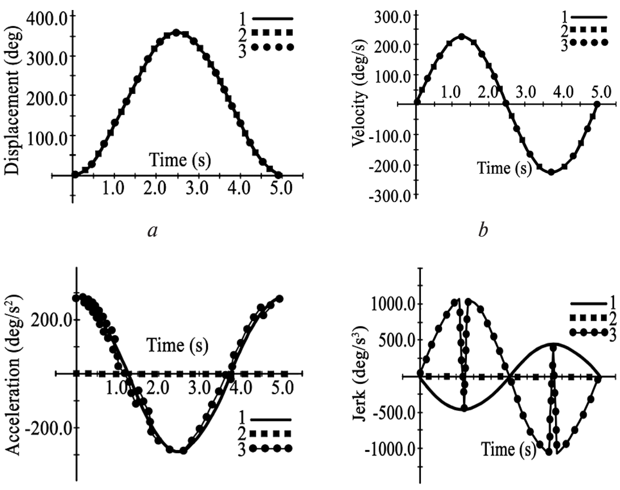
- absolute root-mean-square error for determining power in both motors

$$\varepsilon = \frac{\sqrt{\sum_{i=1}^{N} \Delta_i^2}}{N},$$



a *b*

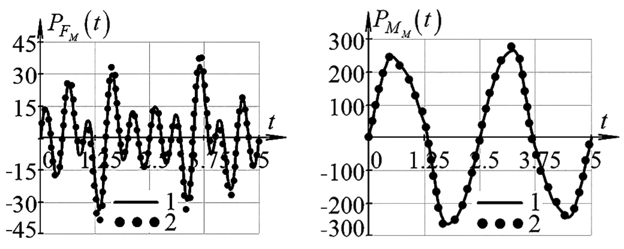
Fig. 10. Graphs of time dependence of the rocker rotation angle (a) and the slider relative translational motion (b), when N = 100



a *b*

c *d*

Fig. 11. Graphs of time dependence of displacement (a), velocity (b), acceleration (c) and jerk (d) in a Rotary Motor under the influence of interpolators – cubic spline (curve 1), linear interpolation (curve 2) and Akima spline (curve 3), when N = 100



a *b*

Fig. 12. Graphs of time dependence of the power in the Linear (a) and Rotary (b) motors, when N = 100

here Δ_i is the difference between the vectors formed under the influence of the interpolation polynomial and those found by mathematical modeling; N is the number of discrete time points of the research;

- the maximum relative error

$$\beta = \frac{\varepsilon N}{\sqrt{\sum_{i=1}^{N} P_i^2}} \cdot 100\,\%.$$

Thus, if $N = 100$, then the maximum relative error in determining the torque and power in a rotary motor is 2.66 and 4.14 %, respectively, and the force and power in a linear motor are 18.65 and 24.14 %, respectively.

In order to reduce the error, we try to increase the number of points to $N = 500$. However, this does not result in a significant change (Table).

Conclusions. Due to the fact that the tubbing erector manipulator mechanism, as a system with several degrees of freedom, must deliver the tubbing to the specified location of the processing ring assembly along the optimal trajectory of movement with minimal force and time spent on delivery, modern computer modeling methods have been used, including the SOLIDWORKS MOTION program. In this case, it becomes necessary to estimate the calculation error by comparing the results given by the mentioned program with the mathematical model of the test problem.

The error determined in the modeling performed by the SOLIDWORKS MOTION program, when solving the direct problem of the dynamics of the moving elements of the tubbing erector manipulator mechanism, is caused by the fact that, in accordance with the motion law of its drives, it is necessary to determine the static and dynamic characteristics of this device movement.

As it turned out, when modeling the manipulator mechanism movement using the SOLIDWORKS MOTION program, calculation errors have occurred caused by incorrect settings of the "interpolator" and "integrator" modules.

From the research results it follows that when specifying the movement of elements in the form of smooth analytical functions, there is a complete coincidence of the calculation results obtained from the SOLIDWORKS MOTION program with those obtained using mathematical modeling tools.

At the same time, when the input object's motion is specified in the form of a vector formed from a discontinuous function, the "interpolator" module begins to operate, which combines three types of interpolation: linear spline, cubic spline, and Akima spline. Then, as a consequence, a function is formed that has infinite values of velocity, acceleration or jerk parameters, which, as it turned out, gives erroneous results of the object's motion analysis.

Calculation of parameters in the SOLIDWORKS MOTION program allows obtaining the following results:

- influence of a linear spline gives infinite values of accelerations and jerks of movement;
- Akima spline – acceleration jump and jerk;
- the cubic spline leads to the formation of smooth harmonic functions characterizing displacement, acceleration and jerk.

It has been determined that when modeling the mechanism parameters, it is not expedient to increase the number of discrete time points by more than 50, since the error in calculating the maximum power values of the manipulator motors does not exceed 20 % (power hydraulic cylinder) and 5 % (hydraulic motor).

Table

The results of determining the maximum of β relative error, realized by generators with different number of discrete time points N of the object's motion

Number of discrete time points N of the object's motion	Maximum relative error of the result, %			
	β_{F_M}	$\beta_{P_{F_M}}$	β_{M_M}	$\beta_{P_{M_M}}$
10	90.71	126.56	11.73	8.77
25	19.63	24.51	2.93	4.19
50	18.92	24.19	2.68	4.15
100	18.65	24.16	2.66	4.14
500	18.64	24.15	2.66	4.14

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Оцінювання точності моделювання засобами програми SOLIDWORKS MOTION механізму маніпулятора тюбінгоукладача

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Мета. Оцінити похибку обчислень у визначенні кінематичних і динамічних характеристик руху механізму маніпулятора тюбінгоукладача типу УТ62.

Методика. Використовуються сучасні методи комп’ютерного аналізу, реалізовані в низці обчислювальних комплексів. Через складність завдання, для оцінювання похибки обчислень була розроблена математична модель фрагмента механізму маніпулятора, що являє собою гідроциліндр, який має два ступеня вільності. Цю модель використовували як тестову.

Результати. На прикладі дослідження моделі показано, що при імітації руху механізму можливі помилки в розрахунках, пов’язані з некоректною постановкою задачі, а також із неточним налаштуванням модулів перетворення вихідних даних. За цих обставин був зроблений висновок про те, що коли вхідне переміщення ланки задано у вигляді вектору, то, визначаючи кінематичні й динамічні характеристики механізму, необхідно в модулі інтерполятор використовувати кубічний сплайн.

Наукова новизна. Уперше було виконане пряме завдання динаміки механізму маніпулятора, що полягає у визначенні статичних і динамічних характеристик пристрою відповідно до заданого закону руху його приводів. Було відзначено, що коли рух задано у вигляді гладких аналітичних функцій, то спостерігається повний збіг результатів розрахунків, виконаних програмою SOLIDWORKS MOTION, з тими, що отримані шляхом математичного моделювання. До того ж, коли вхідне переміщення ланки механізму змодельоване у вигляді вектору, то застосований у модулі «інтерполятор» кубічний сплайн забезпечує гладкі гармонічні функції процесів руху, прискорення й ривка. У результаті моделювання параметрів механізму маніпулятора виявилося, що використовувати в дослідженні більше 50 точок дискретного часу недоцільно. Отже, похибки обчислення максимальних значень потужності двигунів маніпулятора не перевищують 20 % щодо силового гідроциліндра та 5 % – гідромотора.

Практична значимість. Пропонований алгоритм можна використовувати для моделювання руху складних механізмів у машинах.

Ключові слова: *SOLIDWORKS, SOLIDWORKS MOTION, маніпулятор тюбінгоукладача, дискретний час, кубічний сплайн, сплайн Акіма*

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