# Research on the dynamics of the hydraulic drive of the boom lifting mechanism of a mobile robot crane-manipulator

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**Abstract.** In this work, a mathematical model of the lifting mechanism of the crane-manipulator boom was build, which takes into account the mass-inertial parameters of the mechanical part and the dynamic characteristics of the hydraulic drive. Based on the constructed models, numerical simulation of the movement of the hydraulic cylinder rod was carry out depending on external loads and system parameters. This allowed us to assess the influence of the main factors on the dynamic characteristics of the drive. To solve the problem, the 4th order Runge-Kutta method was used. The result of the research showed that to solve this problem it is necessary to use more accurate calculation methods. Increasing the number of calculation points in less accurate methods does not allow finding a solution, as it leads to a significant increase in memory for calculations.

Numerical simulations were perform in Mathematica and the programming language Python.

This research also showed how strongly the size of the diameter of the drive cylinder affects the dynamics of the movement process. In particular, an increase in the diameter of the drive hydraulic cylinder leads to a decrease in the speed of extension and movement and creates micro-oscillations in the hydraulic system, which is a consequence of wave processes in such a drive system. A decrease in the diameter of the hydraulic cylinder is also undesirable, since the system can quickly carry out uncontrolled movement, which will lead to an increase in acceleration, and therefore dynamic loads. Also, such movement is quite difficult to control by the control system.

**Keywords**: motion dynamics, crane-manipu-lator, Runge-Kutta method, hydraulic drive, wave processes.

## INTRODUCTION

Mobile robotics has recently become very is popular. Among the common variants of mobile robot, structures are mobile crane manipulators. Modern mobile cranes-manipulators are widely used in the construction, utility and logistics industries due to their versatility and high productivity. One of the key components of such machines is the boom lifting mechanism, which ensures the accuracy of load positioning and stability of operation under dynamic loads. As a rule, hydraulic drives based on hydraulic cylinders used to implement the boom lifting mechanism. This allows for high specific power, reliability and ease of operation. However, due to the complexity of dynamic processes that occur during the operation of a hydraulic drive, the issue of modeling and optimizing its parameters remains relevant. The dynamic characteristics of such a drive significantly affect the positioning accuracy, energy consumption and reliability of the equipment. One of the important tasks for mobile systems is optimal power supply, in particular, there are tasks where it is necessary to create systems with a battery power supply systems and such a system must operate for a long period on a single battery charge. Research into the dynamics of the hydraulic drive of the boom lifting mechanism of mobile cranes-manipulators will allow finding the optimal drive design, which will allow realizing the task of efficient energy supply of such systems.

In this paper proposed to investigate a dynamic model of the hydraulic drive of the lifting mechanism of the crane-manipulator boom, which takes into account inertial, gravitational and damping loads, as well as the kinematics of the mechanism.

## OVERVIEW OF EXISTING WORKS

Significant parts of scientific works are devoted to the issue of modeling and optimization of hydraulic drives. In works [1-3], mathematical models of hydraulic systems was consider, taking into account the compressibility of the liquid, friction and dynamics of valves. The authors of scientific works [4, 5] investigate the influence of hydraulic cylinder parameters on the dynamics of manipulators, in particular on the accuracy of positioning and the stability of their operation, and in works [6, 7], methods for optimizing hydraulic drives according to the criteria of energy efficiency and reliability are proposed.

However, most existing models do not take into account the specifics of the operation of mobile cranes, in particular the dynamics of the boom when lifting a load under conditions of variable external loads. In [7], a model of a hydraulic drive it was present, taking into account the inertial characteristics of the boom, but without a detailed analysis of the kinematics of the mechanism and the influence of damping forces. Thus, the development of a generalized dynamic model that would take into account all the main factors affecting the operation of the hydraulic drive of the boom lifting mechanism remains relevant.

## PURPOSE OF THE ARTICLE

The main goal of the work is to develop dynamic and mathematical models of the hydraulic drive of the lifting mechanism of the boom of the crane-manipulator of a mobile robot. Based on the developed mathematical model, it was propose to investigate the numerical modeling of the movement of the hydraulic cylinder rod depending on external loads and system parameters.

## THE MAIN MATERIAL

The object of research in this work is the hydraulic drive of the lifting mechanism of the boom of the mobile robot crane-manipulator (Fig. 1). The main elements of the system: a double-acting drive hydraulic cylinder *3*; a boom system *2* with distributed mass and inertia characteristics; a mechanism for attaching the hydraulic cylinder *1* to the boom; a hydraulic pressure control system. In the future, a research will be conduct on the loads that occur on the hydraulic cylinder rod of the boom lifting mechanism.

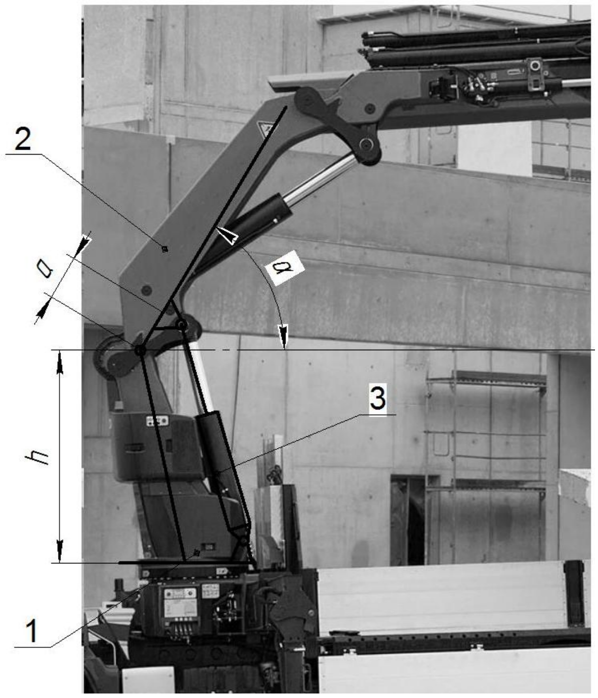



**Fig. 1**. Palfinger crane-manipulator unit: *1* – slewing column; *2* – boom system; *3* – drive hydraulic cylinder of the boom lifting mechanism

*Dynamic model of the boom system.*

To build a dynamic model of the lifting mechanism, we will assume that only this hydraulic cylinder is working, and the boom system is a structure with a constant geometry and moment of inertia $J_c$. The next assumption is that the boom is a beam of rectangular cross-section with length $L_1$ and one of the ends of which is hinge to a support. The boom drive cylinder is mounted at *a* distance a from the boom pivot point (see Fig. 1). The boom jib has a mass $m_1$, which is concentrated in its center. We will model the external load in the form of additional mass $m_b$. This mass is place at the end of the boom at a distance $L_1$ and during the movement of the boom, it have same moves. The force *F* on the rod of the drive hydraulic cylinder acts along its axis.

Fig. 2 shows a dynamic model of the manipulator boom system under study. The following model parameters were subsequently adopted for the research: *a* = 0,4 m; *b* = 1,4 m; $L_1$ = 2 m; h=1,307 m; $m_1$ = 100 kg; $m_b$ = 50 kg.

*Geometrical parameters of the model*.

The rotation angle of the manipulator boom, measured between its upright and lifting beam according to the law of cosines with ΔAOK (see Fig. 2) is equal to:

$$\alpha_0 = \arccos\left(\frac{a^2 + b^2 - q_1^2}{2ab}\right), \tag{1}$$

where: *a* and *b* – hydraulic cylinder installation dimensions, m; $q_1$ – the length of the hydraulic cylinder, which takes into account the movement of its rod.

Boom elevation angle relative to the horizontal plane:

$$\alpha = \alpha_0 - \alpha_1, \tag{2}$$

where: $\alpha_1 = \frac{\pi}{2} - \alpha_2$.

The angle $\alpha_2$ is constant and depends on the configuration of the manipulator boom system:

$$\alpha_2 = \arccos\left(\frac{h}{b}\right). \tag{3}$$

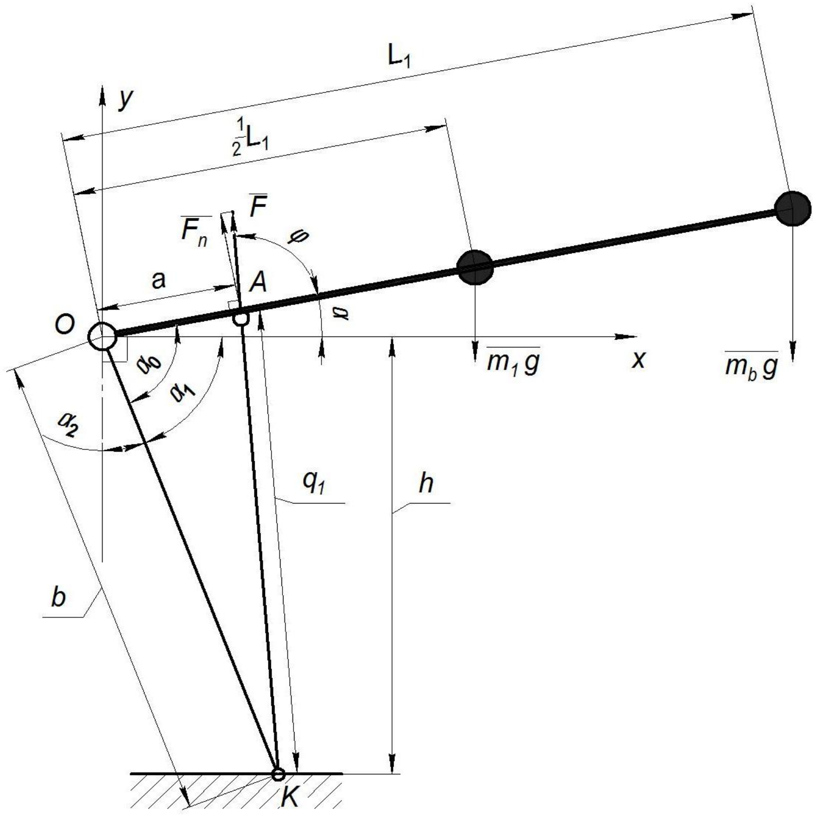



**Fig. 2**. Dynamic model of the boom system under study

where: $h = 1{,}3$ m – boom support height, m.

Finally, the following expression was obtain for the rotation angle $\alpha$:

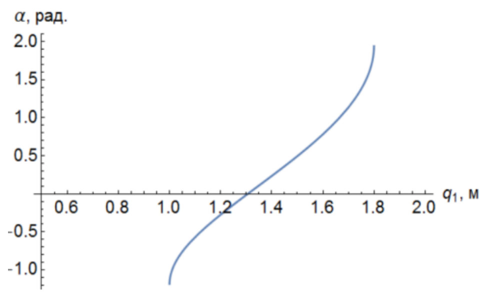
$$\alpha = \arccos\left(\frac{a^2 + b^2 - q_1^2}{2ab}\right) - \frac{\pi}{2} + \arccos\left(\frac{h}{b}\right). \quad (4)$$

The angle between the force vector on the hydraulic cylinder rod and the boom beam according to the cosine theorem also from the ΔAOK see on Fig. 2:

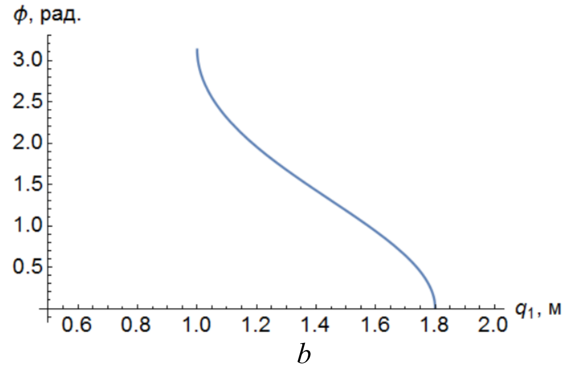
$$\varphi = \arccos\left(\frac{a^2 + q_1^2 - b^2}{2aq_1}\right). \quad (5)$$

From this ratio, it is clear that $q_1 \neq 0$.

Since the physical model of the manipulator crane has limitations on movement, the range of rotation angles was investigate $\alpha$ and $\varphi$ (see Fig. 3). These graphical dependencies show that changing the generalized coordinate $q_1$, which corresponds to the movement of the hydraulic cylinder rod, occurs in the range from 1.0 to 1.8 m. All other values will lead to collisions in the solution of the mathematical model for the given system dimensions. In the following, we will consider the case of motion where the initial position of the rod is $q_1 = 1{,}4$ m.




*a*




*b*

**Fig. 3**. Angle change graphs $\alpha$ (*a*) and $\varphi$ (*b*)

*Static research*.

The change in static force on the hydraulic cylinder rod in the static load mode was investigate. For this purpose, the equation of moments relative to the boom pivot point was determined, from which the following follows:

$$F = \frac{m_1 g \frac{L_1}{2} \cos\alpha + m_b g L_1 \cos\alpha}{a \sin\varphi}, \quad (6)$$

where: $F$ – force on the hydraulic cylinder rod, $m_1$, $m_b$ – weights of the boom and load respectively.

From equation (6) it is obvious that a decrease in the arm a will lead to an increase in the force on the hydraulic cylinder rod, and it is not desirable to achieve such a case when the angle $\varphi = 0$.

The graph showing the change in static force over the entire range of movement is show on Fig. 4. If the static force on the hydraulic cylinder rod will be know, it is possible to estimate the pressure in the pressure line to support the piston (see Fig. 5).

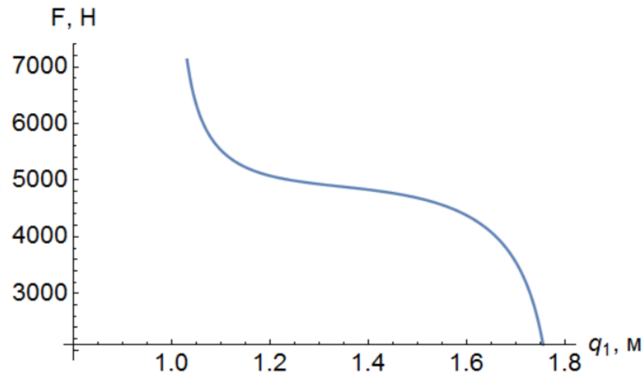



**Fig. 4**. Graph of change in static force on the rod of the drive hydraulic cylinder of the developed physical model

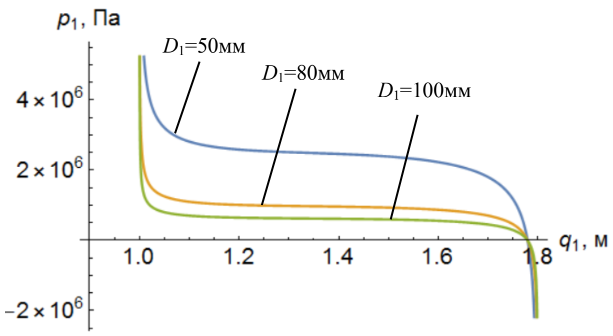



**Fig. 5**. Graph of the backpressure in the discharge cavity of the boom drive hydraulic cylinder in a state of static equilibrium for different rod extension positions and piston diameters

*Dynamic research.*

To construct a mathematical module of the mechanical system of the boom, which would describe the dynamics of work, we will use the Lagrange equations of the 2-nd kind for such systems:

$$\frac{d}{dt}\frac{\partial T}{\partial \dot{q}_1} - \frac{\partial T}{\partial q_1} = F - \frac{\partial \Pi}{\partial q_1}. \qquad (7)$$

The kinetic energy of the mechanical system of the boom has the form:

$$T = \frac{1}{2} J_{c1} \dot{\alpha}^2, \qquad (8)$$

where: $J_{c1} = J_{01} + m_b L_1^2$ - combined moment of inertia of the boom and load; $J_{01} = \frac{m_1 L_1^2}{3}$ - moment of inertia of the boom; $m_1$ — weight of the boom.

Potential energy of the mechanical system of the boom:

$$\Pi = -m_1 g \frac{L_1}{2} \sin\alpha - m_b g L_1 \sin\alpha. \qquad (9)$$

From equation (7), we define a mathematical model that describes the dynamics of the load change on the drive hydraulic cylinder of the boom system:

$$F = \frac{J_{c1}}{a^2 b^2 (1-\xi^2)} \times$$

$$\times \left(\ddot{q}_1 q_1^2 + \dot{q}_1^2 q_1 - \frac{\xi}{ab(1-\xi^2)^2} \dot{q}_1^2 q_1^3\right) + \qquad (10)$$

$$+ \frac{g L_1 m_b q_1 \xi}{ab\sqrt{1-\xi^2}} + \frac{g L_1 m_1 q_1 \xi}{2ab\sqrt{1-\xi^2}}$$

де $\xi = \frac{a^2 + b^2 - q_1^2}{2ab}$

Further, to study the dynamics of the hydraulic drive, a simplified scheme was considered (see Fig. 6), which consists of an unregulated hydraulic pump *H1*, a spool valve P1, a hydraulic cylinder *C1* with rodless *1* and rod *2* cavities. The hydraulic system has an open circulation of the working fluid through the hydraulic tank *T1*.

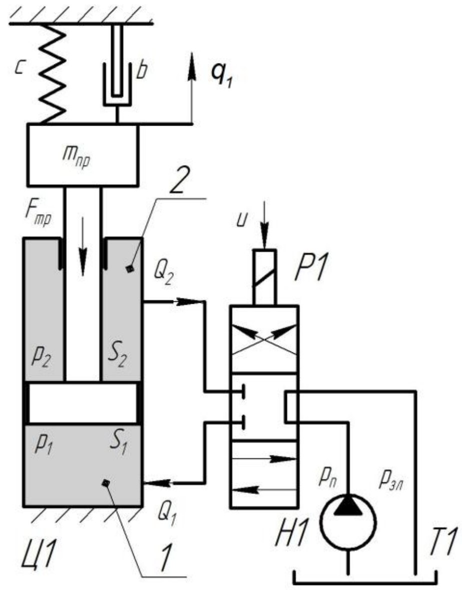



**Fig. 6.** Calculation diagram of the hydraulic drive

According to D'Alembert's principle, we write the differential equation of motion of the hydraulic cylinder rod:

$$m_{np}\ddot{q}_1 + b\dot{q}_1 + cq_1 = \\ = p_1 S_1 - p_2 S_2 - F_{mp} - F, \quad (11)$$

where:

$m_{np} = J_{c1}\dfrac{\dot{\alpha}^2}{\dot{q}_1^2} = J_{c1}\dfrac{q_1^2}{a^2 b^2 (1-\xi^2)}$ – total mass of the moving parts of the boom system, kg; $q_1$ – hydraulic cylinder rod movement, m; $p_1$ and $p_2$ – pressure in the piston and rod cavities of the hydraulic cylinder, respectively, Pa; $S_1$ and $S_2$ – effective area of the piston on the side of the rodless and rod cavities, respectively, m$^2$; $c$ – stiffness coefficient, N/m; $b$ – coefficient of viscous friction, H/(m/s); $F_{mp}$ – semi-dry friction force of the piston and rod of the hydraulic cylinder, H; $g$ – acceleration of free fall, m/s$^2$.

The pressure change in the cavities of the hydraulic cylinder was determined taking into account the equations of flow continuity and compression of the working fluid:

$$\frac{dp_1}{dt} = \frac{E_{np1}}{V_1}(Q_1 - S_1\frac{dq_1}{dt}); \quad (12)$$

$$\frac{dp_2}{dt} = \frac{E_{np2}}{V_2}(-Q_2 + S_2\frac{dq_1}{dt}), \quad (13)$$

where: $Q_1$ and $Q_2$ – fluid flow rates in rodless and rod cavities, respectively, m$^2$/s; $E_{np1}$ and $E_{np2}$ – the combined bulk modulus of the working fluid in the rodless and rod cavities, respectively, Pa; $V_1$ and $V_2$ – volumes of working fluid in the rodless and rod cavities of the hydraulic cylinder, taking into account the volumes of fluid in the adjacent pipelines, respectively, m$^3$.

A spool valve can pass a certain amount of fluid through its passage channels, in particular:

- fluid pumped into the rodless cavity of a hydraulic cylinder from a hydraulic pump,

$$Q_1 = \mu \cdot f_{\partial p1}\sqrt{\frac{2}{\rho}\left|p_n - p_1\right|} \times sign(p_n - p_1); \quad (14)$$

- fluid discharged from the hydraulic cylinder into the tank,

$$Q_2 = \mu \cdot f_{\partial p2}\sqrt{\frac{2}{\rho}\left|p_2 - p_{зл}\right|} \times sign(p_2 - p_{зл}), \quad (15)$$

where: $\mu$ - flow rate of working fluid through the throttle distributor channels; $\rho$ - working fluid density, kg/m$^3$ (860 kg/m$^3$); $p_n$ - hydraulic pump operating pressure, Pa (subsequently adopted 16 MPa); $p_{зл}$ - pressure in the drain line, Pa.

From equations (10) and (11), the acceleration of the hydraulic cylinder rod was determined:

$$\ddot{q}_1 = \frac{a^2 b^2 (1-\xi^2)}{2J_{c1}q_1^2}(p_1 S_1 - p_2 S_2 - F_{mp} - \\ - b_d\dot{q}_1 - c_{np}q_1) - \frac{\dot{q}_1^2}{2q_1} + \frac{\xi\dot{q}_1^2 q_1}{2ab(1-\xi^2)} - \quad (16) \\ - (\frac{2m_b + m_1}{4J_{c1}q_1})abgL_1\xi\sqrt{1-\xi^2}.$$

Thus, the dependencies (11) – (16) allow us to describe the dynamics of the hydraulic cylinder in the mechanism for changing the reach of the crane-manipulator. These equations are the main components of the mathematical model of the hydraulic system.

## NUMERICAL MODELING

For the numerical development of the mathematical model, the Runge-Kutta method of the 4-th order was use. The application of the two-step Euler method for these equations requires a sufficiently small differentiation step for this method to work and determine the parameters of the system. This leads to significant time and memory consumption of the device where such a calculation will be perform.

Further, to simplify the calculation, pressure changes in the rodless cavity of the hydraulic cylinder and transient processes for this cavity were not take into account.

The area of the throttle gap was determine by the diameter of the hole through which the fluid passes and this size does not change during system operation:

$$f_{\partial p1} = \frac{\pi D_{\partial poc}^2}{4}, \qquad (17)$$

where: $D_{\partial poc}$ = 2 mm is throttle gap diameter.

The simulation was performed for the following initial positions of the mechanical system: $q_1(0)$ = 1,37 m; μ = 0,62; $c_{np}$ = 2000 H/m; $b_d$ = 1000 kg/s; $V_1$ = 0,00002 m$^3$; $F_{mp}$ = 100 H; Δt = 0,001 s.

From the analysis of these parameters, we see that the initial position of the boom from which the movement begins is horizontal. This means that at the beginning of the movement, the pressure in the hydraulic system must have the value of static pressure for a given boom position.

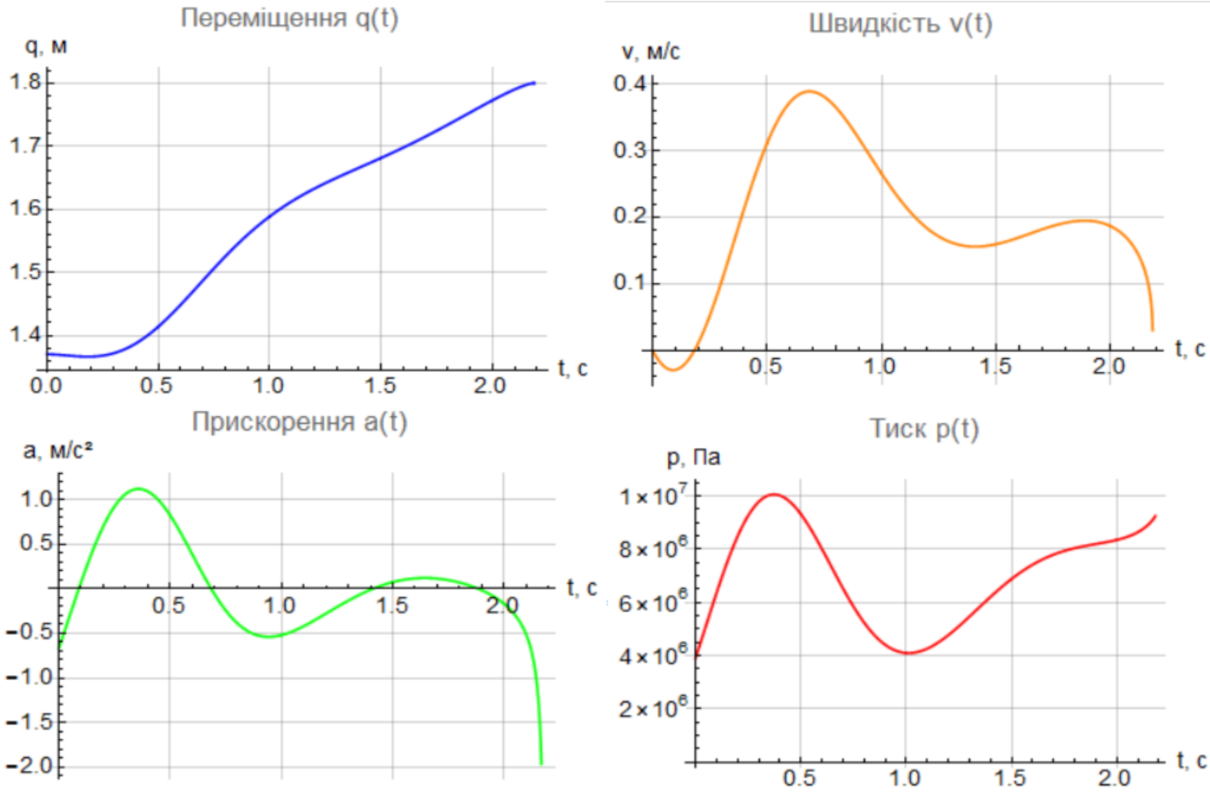



**Fig. 8.** Graphs of changes in displacement, velocity, acceleration of the piston and pressure in the piston cavity for a piston diameter of 40 mm ($E_1 = E_2$ = 1,45 MPa)

Fig. 7 shows the computational model of the Runge-Kutta method in the Mathematica system, and the graphs of changes in the main parameters of the system calculated using this algorithm are shown in Fig. 8-10.

```
(*Initial conditions*)
  ξ0 = (a^2 + b^2 - q0^2)/(2·a·b);
  alpha0 = ArcCos[ξ0] - π/2 + ArcCos[h/b];
  ϕ0 = ArcCos[(a^2- b^2+ q0^2)/(2·a·q0)];
  p0 = ((g·L1·Cos[alpha0])/(a·Sin[ϕ0]·S1)·(0,5·m1 + mb));
(*Acceleration*)
  acceleration[q_, v_, p_] := Module[{ξ, Theta}, ξ = (a^2 + b^2 - q^2)/(2·a·b);
  Theta = ArcCos[h/b] + ArcCos[ξ];
  (a^2·b^2·(1 - ξ^2))/(2·J1·q^2)·(p·S1 - Fmp - (cpr + ((0.5·m1 + mb)·g·L1·Sin[Theta])/(a·b·Sqrt[1 - ξ^2]))·q –
  - (1 - (ξ ·q^2)/(a·b·(1 - ξ^2)))·(J1·q)/(a^2· b^2·(1 - ξ^2))·v^2 - bpr·v)];
(*Pressure change*)
  dpdt[q_, v_, p_] :=  Module[{Q}, Q = μ·fdr·Sqrt[(2/ρ·Abs[pn - p])]·Sign[pn - p]; E1/V1·(Q - S1·v)];
(*One step RK4*)
rk4Step[{q_, v_, p_}, dt_] := Module[{k1q, k1v, k1p, k2q, k2v, k2p, k3q, k3v, k3p, k4q, k4v, k4p,
   qNew, vNew, pNew},
    (*k1*)
      k1q = v;
      k1v = acceleration[q, v, p];
      k1p = dpdt[q, v, p];
    (*k2*)
      k2q = v + 0.5*dt*k1v;
      k2v = acceleration[q + 0.5*dt*k1q, v + 0.5*dt*k1v, p + 0.5*dt*k1p];
      k2p = dpdt[q + 0.5*dt*k1q, v + 0.5*dt*k1v, p + 0.5*dt*k1p];
    (*k3*)
      k3q = v + 0.5*dt*k2v;
      k3v = acceleration[q + 0.5*dt*k2q, v + 0.5*dt*k2v, p + 0.5*dt*k2p];
      k3p = dpdt[q + 0.5*dt*k2q, v + 0.5*dt*k2v, p + 0.5*dt*k2p];
    (*k4*)
      k4q = v + dt*k3v;
      k4v = acceleration[q + dt*k3q, v + dt*k3v, p + dt*k3p];
      k4p = dpdt[q + dt*k3q, v + dt*k3v, p + dt*k3p ];
(*Update results*)
   qNew = q + (dt/6)*(k1q + 2*k2q + 2*k3q + k4q);
   vNew = v + (dt/6)*(k1v + 2*k2v + 2*k3v + k4v);
   pNew = p + (dt/6)*(k1p + 2*k2p + 2*k3p + k4p);
   {qNew, vNew, pNew}];
(*Arrays for results*)
   qValues = {}; vValues = {}; aValues = {}; pValues = {}; tValues = {};
(*Initial values*)
   qCurrent = q0;
   vCurrent = v0;
   pCurrent = p0;
(*Main cycle*)
  For[i = 0, i <= ik, i++, AppendTo[qValues, qCurrent];
    AppendTo[vValues, vCurrent];
    AppendTo[aValues, acceleration[qCurrent, vCurrent, pCurrent]];
    AppendTo[pValues, pCurrent];
    AppendTo[tValues, i*\[CapitalDelta]t];
    {qCurrent, vCurrent, pCurrent} =
     rk4Step[{qCurrent, vCurrent, pCurrent}, \[CapitalDelta]t];
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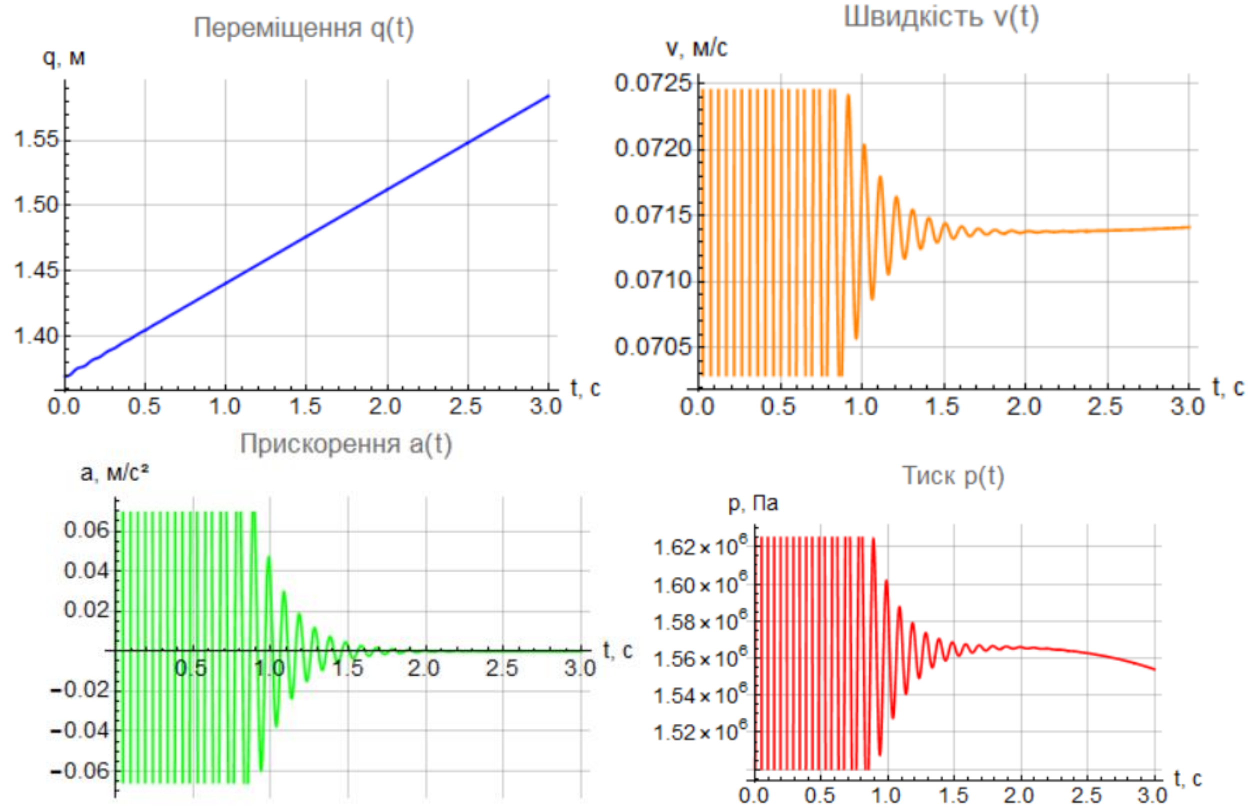


**Fig. 10.** Graphs of changes in displacement, velocity, acceleration of the piston and pressure in the piston cavity for a piston diameter of 80 mm ($E_1 = E_2 = 14,5$ MPa)

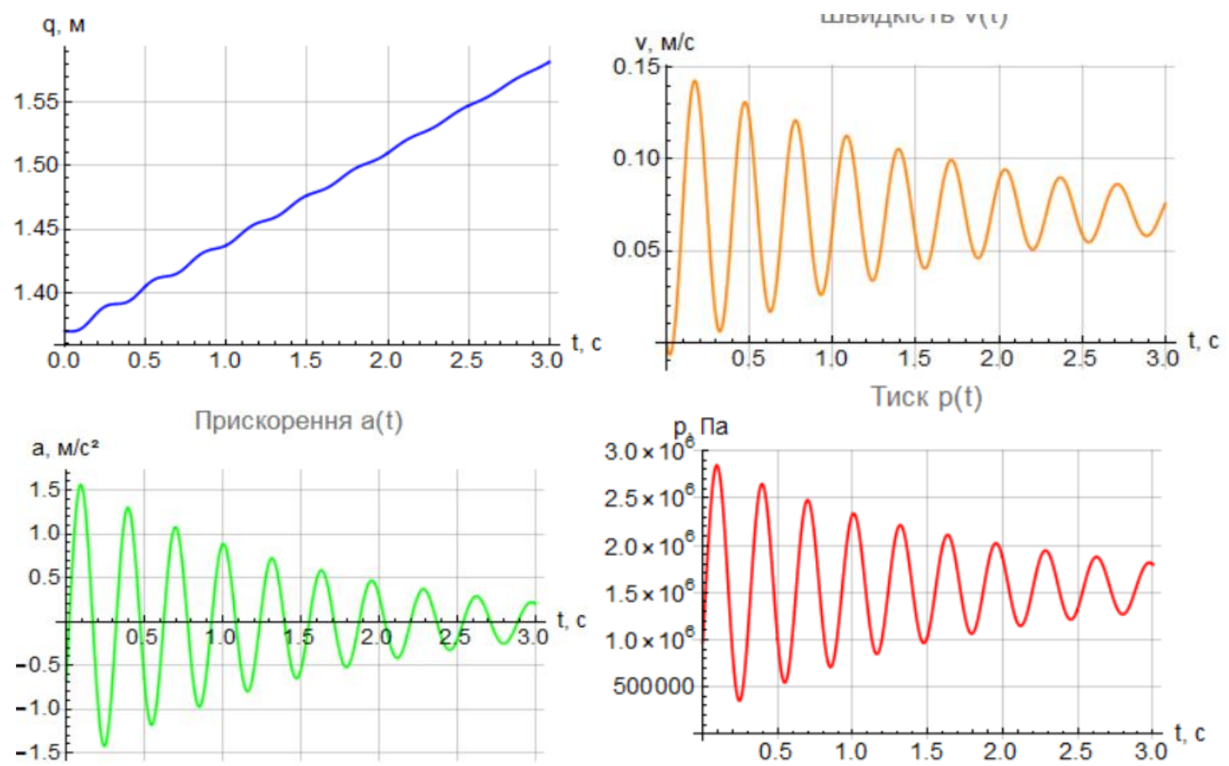


**Fig. 9.** Graphs of changes in displacement, velocity, acceleration of the piston and pressure in the piston cavity for a piston diameter of 80 mm ($E_1 = E_2 = 1,45$ MPa)

## DISCUSSION OF RESULTS

The conducted research showed that there are systems in which physical interactions significantly influence the dynamics of the economic process. In the lifting mechanism, the displacement of the rod of the drive hydraulic cylinder is carried out within predetermined limits. In the initial position, there may already be an external load on the rod, and this load will change unevenly during the movement of the rod and will depend on the position of the crane boom.

The graphs show that oscillatory processes may occur in the hydraulic drive system. With a small diameter of the piston of the drive cylinder, even its reverse movement is possible at the initial stage of movement. However, with increasing pressure in the piston cavity, the system begins to advance, but at a significant velocity. This is because a significant amount of working fluid enters the piston cavity. When the diameter of the piston of the drive hydraulic cylinder increases, the extension speed decreases significantly, as the volume of the piston cavity chamber increases, and this leads to the occurrence of oscillatory processes in the hydraulic system. This phenomenon can be call the wave process of a hydraulic drive.

The nature of the motion process is significantly influence by the modulus of elasticity $E_1$. When its value increased by an order of magnitude, the picture of the oscillatory process in the hydraulic system changes significantly, in particular, initially oscillatory phenomena observed, which during the start-up process attenuated within 1 second (see Fig. 10). Changing the spring stiffness coefficient and damping coefficient slightly affect the nature of the dynamic process of movement and only with significant changes (more than 3 orders of magnitude), will there be a noticeable increase in pressure and a decrease in the speed of movement. In further studies for similar systems, the spring stiffness and damping coefficients can be neglect or the nature of their occurrence can be investigate in more detail, in particular, for systems with high load capacity, the spring stiffness can be interpreted as the elasticity of the boom beam.

## CONCLUSIONS

To study the dynamics of such systems, where the mathematical model will be represent in the form of nonlinear differential equations of the second or higher order, a rather important factor is the choice of the method for solving such equations. The main solution methods are numerical algorithms. The two-step Euler algorithm is simple to implement, but for such systems, it requires a small discretization step. This significantly increases the number of points for calculation, and therefore the amount of allocated memory of the technical system that will perform such calculations. The Runge-Kutta method is more complicated to implement, but it allows you to determine the main parameters of the system quite quickly.

As theoretical studies have shown, when studying the mechanisms for lifting the boom of a crane-manipulator with a lifting capacity of up to 1 ton, it is possible to ignore the elasticity of its boom and the phenomena of damping and boom hinges. At the same time, it is necessary to approach the optimal choice of the piston diameter of the drive hydraulic cylinder quite carefully, especially if such a system requires precise control.

For systems with a large lifting capacity (more than 10 tons), it is necessary to investigate the elastic properties of the boom system in more detail, since the additional elastic force will significantly increase the load on the hydraulic drive and hinged mounts due to vibration processes. An increase in the mass of the moving elements leads to a decrease in the speed of rod movement and an increase in the time of the transition process, and an increase in pressure in the hydraulic system contributes to an increase in the force on the rod. These results in faster boom lift, but causes larger oscillation amplitudes and system instability.

To dampen vibrations in such systems, it is worth installing special hydraulic accumulator systems.

The results of the study can be using to optimize the design of the hydraulic drive, in particular, the selection of optimal hydraulic cylinder parameters.

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## Дослідження динаміки гідравлічного приводу механізму підіймання стріли крана-маніпулятора мобільного робота


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**Анотація**. В даному дослідженні побудовано математичну модель механізму підіймання стріли крана-маніпулятора, яка враховує масово-інерційні параметри механічної частини та динамічні характеристики гідравлічного приводу. На основі побудованих моделей було проведено чисельне моделювання переміщення штока гідроциліндра залежно від зовнішніх навантажень та параметрів системи. Це дозволило оцінити вплив основних факторів на динамічні характеристики приводу. Для розв’язку поставленої задачі було застосовано метод Рунге-Кутта 4-го порядку. Результат дослідження показав, що для вирішення даної задачі необхідно застосовувати більш точні методики розрахунку. Збільшення кількості точок розрахунку в менш точних методах не дозволяє знайти розв’язок, так як це призводить до значного збільшення пам’яті для обчислень.

Чисельне моделювання було проведено в системі Mathematica та потворено мові Python.

Дане дослідження також показало, як сильно впливає розмір діаметру привідного циліндра на динаміку процесу переміщення. Зокрема збільшення діаметру привідного гідроциліндра призводить до зменшення швидкості висування та переміщення та створює мікроколивання в гідросистемі, що є наслідком хвильових процесів в такій системі приводу. Зменшення діаметру гідроциліндра також є не бажаним, так як система може досить швидко здійснити неконтрольоване переміщення, що призведе до збільшення прискорення, а отже і динамічних навантажень. Також таке переміщення досить складно контролювати системою управління.



**Ключові слова**: динаміка руху, кран – маніпулятор, метод Рунге-Кутта, гідропривід, хвильові процес.