DOI: 10.34220/issn.2222-7962/2021.4/8

УДК 169.113

МОДЕЛИРОВАНИЕ РАБОЧИХ ПРОЦЕССОВ ЭНЕРГОСБЕРЕГАЮЩЕГО ГИДРОПРИВОДА МЕХАНИЗМА ПОДЪЕМА ЛЕСНОГО МАНИПУЛЯТОРА

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В настоящее время в России принят курс на создание конкурентоспособных машин и оборудования. Это относится и к мобильным лесозаготовительным машинам манипуляторного типа. Поэтому проектирование и создание лесных манипуляторов с энергосберегающим гидроприводом является актуальной задачей. Приведен анализ исследований технологических, динамических и кинематических характеристик машин манипуляторного типа. Представлена новая гидрокинематическая схема механизма подъема стрелы с системой рекуперации энергии в пуско-тормозных режимах. Разработана математическая модель процесса подъема стрелы, описываемая системой нелинейных дифференциальных уравнений второго порядка. Вследствие нелинейности решить задачу в явном виде невозможно, поэтому применяется метод конечных разностей, в котором все производные заменяются соответствующими разностными аналогами. Тем самым, исходная система сводится к системе рекуррентных соотношений второго порядка. Для решения соотношений применяются методы теории операторов, функционального анализа. Искомые функции найдены в виде совокупности точек в узлах разбиения отрезка по времени. Предварительные расчеты с использованием программы MathCad показали, что подключение энергосберегающего демпфирующего устройства в гидропривод механизма подъема стрелы позволяет снизить пиковое давление в гидролинии поршневой полости гидроцилиндра при переходных режимах в 1,5-1,6 раза. Экспериментальные исследования, проведенные на лабораторном стендовом гидроманипуляторе, позволили получить статистические данные анализа давления системы без аккумулятора и с аккумулятором, а также величину запасаемой энергия за один цикл, которая составила около 30%. Математическая модель представленного энергосберегающего гидропривода лесного гидроманипулятора показывает принципиальную возможность реализации принципа энергосбережения при пуско-тормозных режимах погрузочно-разгрузочных работ. Снижение динамической нагруженности и энергоемкости рабочих процессов механизма подъема стрелы лесного манипулятора путем обоснования параметров энергосберегающего демпфирующего устройства гидропривода позволяет повысить надежность манипулятора, уменьшить энергозатраты, а также время простоев в ремонте из-за выхода из строя гидр оборудования. Полученные результаты могут использоваться при проектировании рекуперативных систем другого грузоподъемного оборудования в строительстве и сельском хозяйстве.

Ключевые слова: гидроманипулятор, механизм подъема, рекуперация, динамические нагрузки, энергозатраты.

Благодарности: Авторы выражают признательность заведующему кафедрой механизации лесного хозяйства и проектирования машин доценту Дручинину Денису Юрьевичу за оказанную помощь в подготовке материалов для статьи, а также главному редактору доценту Новикову Артуру Игоревичу за оказание помощи с оформлением данной статьи

Конфликт интересов: Авторы данной статьи заявили об отсутствии личных, коммерческих, идеологических, интеллектуальных конфликтов интересов при подготовке, написании и публикации данной статьи.

Для цитирования: Моделирование рабочих процессов энергосберегающего гидропривода механизма подъема лесного манипулятора / С. П. Глушков, А. Т. Рыбак, П. И. Попиков, Д. С. Богданов, А. В. Конюхов, В. И. Усков // Лесотехнический журнал. – 2021. – Т. 11. – № 4 (44). – С. 88–99. – Библиогр.: с. 96–98 (13 назв.). – DOI: 10.34220/issn.2222-7962/2021.4/8.

Поступила: 15.11.2021 ***Принята к публикации***: 22.12.2021 ***Опубликована онлайн***: 30.12.2021

MODELING OF THE WORKING ENERGY-SAVING PROCESSES OF THE HYDRAULIC DRIVE OF THE LIFTING MECHANISM OF A FORESTRY MANIPULATOR

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Abstract

At present Russia has adopted a course towards the creation of competitive machinery and equipment. This also applies to mobile manipulator-type forestry machines. Therefore, the design and creation of forestry manipulators with an energy-saving hydraulic drive is an urgent task. The analysis of research of technological, dynamic and kinematic characteristics of manipulator-type machines is given. A new hydrokinematic diagram of the boom lifting mechanism with an energy recuperation system in starting and braking modes is presented. A mathematical model of the boom lifting process has been developed, described by a system of nonlinear differential equations of the second order. Due to nonlinearity, it is impossible to solve the problem in an explicit form; therefore, the finite difference method is used, in which all derivatives are replaced by the corresponding difference analogs. Thus, the original system is reduced to a system of second-order recurrence relations. To solve the relations, the methods of operator theory and functional analysis are used. The sought-for functions are found in the form of a set of points at the nodes of the division of the segment in time. Preliminary calculations using the MathCad program showed that the connection of an energy-saving damping device to the hydraulic drive of the boom lifting mechanism makes it possible to reduce the peak pressure in the hydraulic line of the piston cavity of the hydraulic cylinder during transient modes by 1.5-1.6 times. Bench hydraulic manipulator, made it possible to obtain statistical data on the analysis of the pressure of the system without and with an accumulator, as well as the amount of stored energy per cycle, which amounted to about 30%. The mathematical model of the presented energy-saving hydraulic drive of the forest hydraulic manipulator shows the fundamental possibility of implementing the principle of energy saving during starting and braking modes of loading and unloading operations. Reducing the dynamic loading and energy intensity of the working processes of the boom lifting mechanism of the forestry manipulator by justifying the parameters of the energy-saving damping device of the hydraulic drive allows increasing the reliability of the manipulator, reducing energy costs, as well as the downtime for repairs due to the failure of hydraulic equipment. The results obtained can be used in the design of recuperative systems for other lifting equipment in construction and agriculture.

Keywords: hydraulic manipulator, lifting mechanism, recuperation, dynamic loads, energy costs.

Acknowledgments: the authors express their gratitude to the head of the department of mechanical forestry and machine design Denis Yuryevich Druchinin for his help in preparing materials for articles, as well as to the chief editor Novikov Artur Igorevich for helping with the design of this article.

Conflict of interest: The authors of this article stated that there were no personal, commercial, ideological, intellectual conflicts of interests in the preparation, writing and publication of this article.

For citation: Glushkov S. P., Rybak A. T., Popikov P. I., Uskov V. I., Bogdanov D. S., Konyukhov A. V. Modeling of the working energy-saving processes of the hydraulic drive of the lifting mechanism of a forestry manipulator. Lesotekhnicheskii zhurnal [Forestry engineering journal]. 2021. Vol. 11. № 4 (44). – P. 88–99. – DOI: 10.34220/issn.2222-7962/2021.4/8.

Received: 15.11.2021 ***Revised:*** 13.12.2021 ***Accepted***: 22.12.2021 ***Published online***: 30.12.2021

Introduction

At present, Russia has adopted a course towards the creation of competitive machinery and equipment. This also applies to mobile timber harvesting machines of the manipulator type; at the present moment, loading and transport vehicles with manipulators are widely used in thinning forest management. In the Russian Federation, hydraulic manipulators of the brands LV-184A-10, LV-185-14A, LV-190-05, MM-100 are often used, which are installed as part of a road train behind the cab of a car or on the rear overhang of a subframe Hydraulic manipulator LV-184A-10 is the lightest in the class of carrying capacity (52 kN) and is well adapted to our conditions. The manipulator MAYMAN-100S (MM-100) is equipped with a higher quality and efficient hydraulic pump SUNFAB SC-064 (084) with increased productivity and Swedish or Italian hydraulic distributors. It is possible to complete it with a two-loop control system, which allows to reduce dynamic loads and to save fuel consumption by up to 7%. Basic models of Epsilon manipulators: C70L, M100L and Q150L are equipped with strong metal structures, cast column bases.

The experience of operating forest hydraulic manipulators in the regions of the Komi Republic showed that due to high dynamic loads at low air temperatures, failures of high-pressure hoses are 29.7–56%, and hydraulic cylinders 14.0–24.1% [1]. With the wear of the seals, the initial amplitude of the pressure fluctuations of the working fluid decreases, and the period of the oscillations increases. The logarithmic damping decrement decreases to the limit value δ = 0,533 which requires repair of the hydraulic cylinder. Using the developed mathematical model of dynamic processes inside the hydraulic cylinder, a new method for diagnosing hydraulic cylinders has been substantiated.

In work [2], technological operations of a forwarder manipulator were investigated: loaded movements; empty movements and movements of the manipulator when performing operations inside the cargo compartment. It was found that the technological cycle of the manipulator includes 63% of the time of the busy movement, 30% of the time is spent on idle transfers and 7% for the operations performed inside the cargo compartment. As a result of the obtained experimental data, linear regressions of the loading time (sec / m3) were constructed depending on the loading volume and the number of loaded assortments (on average, 47 pieces). The movement time of the manipulator links was recorded by a stopwatch from the start of movement in the cargo space until the jaws of the grabber touched the logs lying on the ground, and until the opening of the jaws, when the assortments were unloaded inside the cargo area of the forwarder. Cases of empty manipulator movements to delete branches were taken into account. sorting and moving of individual log truck.

In addition to studying the influence of various factors on individual elements of the technological cycle, there are a number of works in foreign journals, in which the authors published the results of research on various aspects of automation of manipulators. In the works [3,4,7,8], the concept of an electro hydrostatic drive, applicable to hydraulic manipulators of large carrying capacity, is presented. The drive is designed and analyzed for requirements such as load holding, overload handling and differential flow compensation. Numerical analysis is carried out with the system connected to a constant load mass. Load weight 30,000 kg, accumulator precharge gauge pressure 5 bar. The cylinder moves the mass vertically. It is accepted that the force acting on the cylinder consists of three parts: hydraulic force, gravity force and friction force. The friction force is modeled by the viscous friction coefficient and the Coulomb friction force. Comparison with traditional valve operated actuators showing efficiency gains and energy recovery capabilities.

The work [5, 6] is devoted to the study of a promising scheme of a recuperative hydraulic drive of a timber truck-manipulator. On the basis of mathematical and simulation modeling of the operation of the pneumatic recuperation system, the dependences of the influence of time on the amount of compressed air, pressure and temperature in the pneumatic accumulator were obtained. The pressure after three strokes of one and the other pneumatic cylinders is 2.3 MPa. With continuous consumption of compressed air, pressure pulsations do not exceed 0.4 MPa. It has been established that with intensive periodic compression of air in pneumatic cylinders, high temperatures arise above 1000 K, and the average air temperature is about 200-300 °C. High air temperatures in pneumatic cylinders for energy recovery are a disadvantage of the proposed pneumatic systems.

Investigations of the kinematic and dynamic characteristics of hydraulic manipulators with the combination of the movements of the boom and stick are devoted to works [9, 10]. The graphs of the changes in the pressure values of the working fluid in the hydraulic system with the course of time were obtained. When the movement of the two links is combined, increased fluctuations in the working pressure are observed, and when the movement of the arm stops, the pressure is set at a nominal pressure of 12.5 MPa. The graphs of angular velocities were obtained, which at the beginning of the movement have a smooth character; the boom is overloaded.

However, in our opinion, studies of the dynamic and kinematic characteristics of energy-saving devices as applied to the lifting mechanisms of forest manipulators have not been carried out enough, therefore, additional research in this direction is urgent.

Purpose of the study is to reduce the dynamic loading and energy intensity of the working processes of the boom lifting mechanism of a forest manipulator by substantiating the parameters of an energy-saving hydraulic drive based on solving a mathematical model.

Materials and methods

A serial hydraulic manipulator LV-184 A -10, which is mounted on the chassis of short log trucks, was chosen as the object of research. We propose a new energy-saving hydraulic drive for the boom lifting mechanism of a forestry manipulator [11], which includes an additional recuperation hydraulic cylinder and a hydropneumatic accumulator, which accumulates braking energy when lowering the load and returns it during subsequent lifting of the load (Fig. 1).

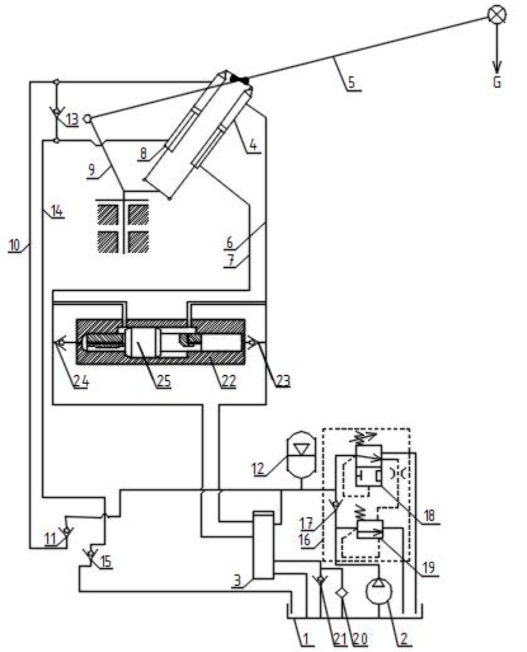

Рисунок 1. Энергосберегающий гидропривод механизма подъема стрелы лесного манипулятора 1-бак; 2- насос; 5; 6; 7; 10; 14- гидролинии; 4-гидроцилиндр привода стрелы 5; 8-дополнительный гидроцилиндр рекуперации; 11; 13; 15; 17; 21; 23- обратные клапаны 12-гидроаккумулятор; 19-предохранительный клапан; 18-разгрузочное устройство; 22-демпфер; 25-плунжер

Figure 1. Energy-saving hydraulic boom lift for forestry manipulator hydraulic tank 1; pump 2; hydrolines 5; 6; 7; ten; fourteen; hydraulic cylinder 4 of the boom drive 5; additional hydraulic cylinder for recuperation 8; check valves 11; 13; 15; 17; 21; 23; hydroaccumulator 12; safety valve 19; unloading device 18; damper 22; plunger 25

Источник: собственная композиция автора
Source: author’s composition

To study the process of loading operations, taking into account the connection of a hydropneumatic device to the hydraulic drive of the boom lifting mechanism, we considered the calculated case of lifting a bundle of assortments load with a maximum reach of the manipulator on a slope, for example, when the soil subsides under the right outrigger at point A. To substantiate the parameters of the energy-saving hydraulic drive of the boom lifting mechanism a computational scheme was drawn up, which is shown in Fig. 2

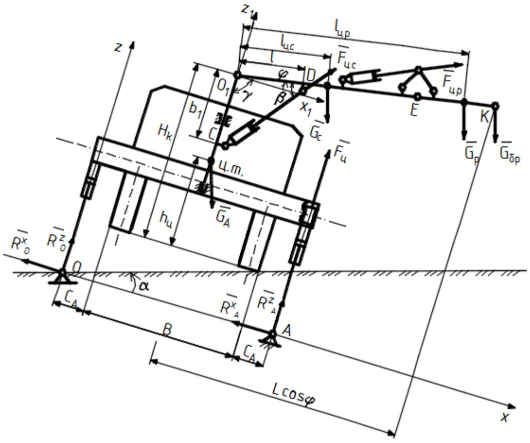

Рисунок 2. Расчетная схема манипулятора на склоне (действующие силы: $G_{бр}$ - сила тяжести бревен в захвате, Н; G_c - сила тяжести стрелы, Н; G_p - сила тяжести рукояти, Н; $F_{цс}$ - усилие в гидроцилиндре для подъема стрелы, Н)

Figure 2. Calculation diagram of an auto-log truck manipulator on a slope (acting forces: Gbr is the gravity of the logs in the grip, N; Gs is the gravity of the boom, N; Gs is the gravity of the handle, N; Fcs is the force in the hydraulic cylinder for lifting the boom, N)

Источник: собственная композиция автор(ов)

Source: author’s composition

We have developed a mathematical model for lifting the boom of a manipulator with an energy-saving hydraulic drive when working on a slope, when the accumulator is fully charged and gives up the accumulated energy when lowering the load to raise the boom:

$$\begin{cases} \left(J_{бр}+J_p+J_c\right)\dfrac{d^2\varphi}{dt^2}=\dfrac{\pi d_c^2 l\ \sin\beta}{4}\cdot p- \\ -\left(G_{бр}L+G_p l_{ц.р}+G_c l_{ц.с}\right)\cos(\varphi-\alpha), \\ q_н n_н=\dfrac{\pi d_c^2 l\ \sin\beta}{4}\cdot\dfrac{d\varphi}{dt}-k_{ак}\sqrt{P_A-p}\ + \\ +a_y p-\dfrac{V_{сум}}{E_{пр}}\cdot\dfrac{dp}{dt}, \\ \dfrac{dP_A}{dt}=\dfrac{-E_ж\, k_{ак}}{V_0\left[1-\left(\frac{P_o}{P_A}\right)^{\frac{1}{K}}+\frac{E_Ж}{KP_A}\left(\frac{P_0}{P_A}\right)^{\frac{1}{K}}\right]}\cdot\sqrt{P_A-p}\,, \end{cases} \tag{1}$$

where $J_{бр}$, $J_р$, $J_с$ – moments of inertia of a bundle of logs, a stick, an arrow relative to a point O_1, kg·m^2;

φ – boom angle, radians;

α – slope angle, radians;

t – time, s;

$G_{бр}$ – gravity of the bundle of logs in the gripper, H;

$d_с$ – inner diameter of the hydraulic cylinder, m;

$q_н$ – pump displacement, м^3/ turnover;

$n_н$ – pump speed, s^{-1};

p – current pressure in the discharge line, Pa;

P_A – current value of pressure in the accumulator , Pa;

Po – accumulator pre-charge pressure, Pa;

Vo – working volume of the accumulator, m^3;

$E_{пр}$ – reduced modulus of elasticity of the working fluid and elastic elements of the hydraulic drive, Pa;

$E_{ж}$ – modulus of elasticity of the liquid, Pa

K – gas adiabatic index in the accumulator equal to 1,41;

μ – the flow coefficient is 0,7...0,8;

$d_{ак}$ – inner diameter of the hydraulic accumulator nozzle, m

$d_р$ – internal diameter of the energy recovery hydraulic cylinder, m;

g – acceleration of gravity, m/s^2;

ρ – working fluid density, kg/m^3;

V_{sum} – total volume of the supply pipeline, m^3;

$k_{ак}$ – battery choke throttling ratio, m^3·c·Pa$^{-1/2}$

$$k_{ак} = \frac{\mu \pi d_{ак}^2}{4}\sqrt{\frac{2}{\rho}}. \qquad (2)$$

The designations of the remaining geometric parameters of the boom lifting mechanism included in equations (1) are clear from Fig. 2. Note that in the triangle O1DC, the cosine theorem implies the relation between the angles β, γ, φ

$$\sin\beta = \frac{b_1 \sin(\gamma+\varphi)}{\sqrt{l^2+b^2-2lb_1\cos(\gamma+\varphi)}}. \qquad (3)$$

In this work, for the possibility of solving the problem, it is assumed that the angle β does not depend on t on a separate site, this means that the ratio on the right side of the formula is constant for each t.

On the segment $t \in [0; t_k]$ we consider the Cauchy problem:

$$\begin{cases} \varphi(0) = \varphi_0, \quad p(0) = p_0\,, \quad P_A(0) = P_{A_0}, \\ \varphi'(0) = \varphi_1, \quad p'(0) = p_1\,, \quad P_A'(0) = P_{A_1}. \end{cases} \qquad (4)$$

Research results. A mathematical model of the boom lifting process is presented, described by a system of second-order nonlinear differential equations. The highest derivative has an irreversible operator - such systems are unresolved with respect to this derivative. Due to the nonlinearity of the system, it is impossible to find a solution in the explicit form of the dependence on t, which entails the need to use approximate methods. The sought functions are calculated at the nodal points t_i. We denote

$$\varphi_i = \varphi(t_i), \; p_i = p(t_i), \; P_{A_i} = P_A(t_i)\,, \qquad (5)$$

where number i varies from 0 to n inclusive. One of the approximate methods is the finite difference method, in which all derivatives are replaced by the corresponding difference analogs:

$$\begin{cases} \frac{d\varphi}{dt}(t_i) \approx \frac{\varphi_{i+1}-\varphi_i}{h}, \frac{d^2\varphi}{dt^2}(t_i) \approx \frac{\varphi_{i+2}-2\varphi_{i+1}+\varphi_i}{h^2}; \\ \frac{dp}{dt}(t_i) \approx \frac{p_{i+1}-p_i}{h}, \frac{d^2p}{dt^2}(t_i) \approx \frac{p_{i+2}-2p_{i+1}+p_i}{h^2}; \\ \frac{dP_A}{dt}(t_i) \approx \frac{P_{A i+1}-P_{A i}}{h}, \frac{d^2P_A}{dt^2}(t_i) \approx \frac{P_{A i+2}-2P_{A i+1}+P_{A i}}{h^2}. \end{cases} \qquad (6)$$

which leads to a system of second-order recurrence relations.

Since system (1) is not resolved with respect to the derivative, the Cauchy problem has a solution not for all initial values.

To solve it with respect to the highest derivative, the result obtained in [12, Theorem 2.1] is used, due to which the system splits into equalities in subspaces of decreasing dimensions. This solution is called the cascade decomposition method. This method was successfully applied, for example, in the study of perturbations of a linear algebraic-differential equation caused by the presence of a small parameter in [13]. For systems of recurrence relations, such an approach is applied, in our opinion, for the first time.

We have carried out the necessary calculations.

Theorem. A solution to problem (1), (3) exists if and only if the following equalities hold:

$$p_1 - p_0 = \sigma(\varphi_1 - \varphi_0) +$$
$$+h \cdot \frac{E_{\text{пр}}}{V_{\text{сум}}} \cdot \left(-k_{\text{ак}}\sqrt{P_A - p} + a_y\, p_0 - q_{\text{н}}\, n_{\text{н}}\right),$$
$$P_{A_1} - P_{A_0} = \frac{-h\, E_{\text{ж}}\, k_{\text{ак}}\sqrt{P_{A_0} - p_0}}{V_0\left[1 - \left(\frac{P_{\text{о}}}{P_{A_0}}\right)^{\frac{1}{\text{К}}} + \frac{E_{\text{Ж}}}{KP_{A_0}}\left(\frac{P_0}{P_{A_0}}\right)^{\frac{1}{\text{К}}}\right]}. \quad (7)$$

The following algorithm has been developed for solving problem (1), (3) on a computer.

1. Enter the values of the coefficients$J_{\text{бр}}$, $J_{\text{р}}$, $J_{\text{с}}$, d_c^2, l, $G_{\text{бр}}$, L, $G_{\text{р}}$, $l_{\text{ц.р}}$, $G_{\text{с}}$, $l_{\text{ц.с}}$, α, $q_{\text{н}}$, $n_{\text{н}}$, μ, a_y, $V_{\text{сум}}$, $E_{\text{пр}}$, $E_{\text{ж}}$, V_0, $P_{\text{о}}$, К, ρ. Определить $k_{\text{ак}}$ формулой (2) и σ - формулой

$$\sigma = \frac{\pi\, d_c^2\, l\, E_{\text{пр}} \sin\beta i}{4V_{\text{сум}}}. \quad (8)$$

2. Enter initial values (at time t=0) the required quantities: $\varphi_0,\ p_0, P_{A_0}$.

It follows from the theorem above that the values of the rates of change φ_1, p_1, P_{A_1} of the required quantities at the initial moment of time must be entered so that equalities (7) are satisfied.

3. Enter value t_k the right end of the segment of change of the variable t. Split this segment with equally spaced anchor points t_i with step h:

$$t_i = i \cdot h, \quad (9)$$
$$where\ \ h = \frac{t_k}{n},\ \ n - number\ of\ split\ points$$

Comment. The larger n, the smaller the error in approximating real functions by their values at the nodal points.

4. Enter the following values

$$\Phi_i^{(1)} = \frac{\pi\, d_c^2\, h^2\, l \sin\beta}{4\left(J_{\text{бр}} + J_{\text{р}} + J_{\text{с}}\right)}\, p_i -$$
$$-\frac{\left(G_{\text{бр}}L + G_{\text{р}}l_{\text{ц.р}} + G_{\text{с}}l_{\text{ц.с}}\right)h^2}{J_{\text{бр}} + J_{\text{р}} + J_{\text{с}}} cos(\varphi_i - \alpha), (10)$$

$$\Phi_i^{(2)} = \frac{\pi^2\, d_c^4\, h^2\, l^2 E_{\text{пр}}\ sin^2\beta}{16\, V_{\text{сум}}\left(J_{\text{бр}} + J_{\text{р}} + J_{\text{с}}\right)}\, p_i -$$
$$-\frac{\pi\, d_c^2\, h^2\, E_{\text{пр}}\ l\ \sin\beta\ \left(G_{\text{бр}}L + G_{\text{р}}l_{\text{ц.р}} + G_{\text{с}}l_{\text{ц.с}}\right)}{J_{\text{бр}} + J_{\text{р}} + J_{\text{с}}}\cos(\varphi_i - \alpha) -$$
$$-\frac{E_{\text{пр}}\, h\, k_{\text{ак}}\, \mu}{V_{\text{сум}}}\sqrt{P_{A_{i+1}} - p_{i+1}} +$$
$$+\frac{a_y\, E_{\text{пр}}\, h}{V_{\text{сум}}}\, p_{i+1} - \frac{E_{\text{пр}}\, h\, n_{\text{н}}\, q_{\text{н}}}{V_{\text{сум}}}, (11)$$

$$\Phi_i^{(3)} = -\frac{-h\, k_{\text{ак}}E_{\text{ж}}\, K\, P_{A_{i+1}}\sqrt{P_{A_{i+1}} - p_{i+1}}}{K\, P_{A_{i+1}}\left(\frac{P_0}{P_{A_{i+1}}}\right)^{\frac{1}{K}} V_0 - E_{\text{ж}}\, V_0\left(\frac{P_0}{P_{A_{i+1}}}\right)^{\frac{1}{K}} - K\, V_0\, P_{A_{i+1}}}. (12)$$

5. The sought values for each i=2,3...,n are determined from iterative processes, when each value is obtained by calculating the previous ones.

The angle φ (in radians) is determined by the formula

$$\varphi_i = \varphi_0 + i(\varphi_1 - \varphi_0) +$$
$$+\sum_{k=1}^{i}\sum_{j=0}^{k-1} C_i^k\, C_{k-1}^j\, (-1)^{k-1-j}\, (i-1-j)\ \Phi_j^{(1)};\ (13)$$

pressure p - by the formula

$$p_i = p_1 + i\ \sigma\ (\varphi_1 - \varphi_0) +$$
$$+\sigma\sum_{k=1}^{i}\sum_{j=0}^{k-1} C_i^k\, C_{k-1}^j\, (-1)^{k-1-j}\, (i-2-j)\ \Phi_j^{(1)} +$$
$$+\sum_{k=1}^{i}\sum_{j=0}^{k-1} C_i^k\, C_{k-1}^j\, (-1)^{k-1-j}\ \Phi_j^{(2)} + \sigma\,\Phi_{i-1}^{(1)} - \Phi_{i-1}^{(2)};\ (14)$$

P_A pressure - by the formula

$$P_{A_i} = P_{A_1} + \sum_{k=1}^{i}\sum_{j=0}^{k-1} C_i^k\, C_{k-1}^j\, (-1)^{k-1-j}\ \ \Phi_j^{(3)} - \Phi_{i-1}^{(3)},\ \ (15)$$

where indicated

$$C_n^m = \frac{n!}{m!\,(n-m)!}. \quad (16)$$

6. Knowing the values φ_i, p_i, p_{A_i}, calculated in the previous step, one can calculate their first and second derivatives using approximate formulas (4).

7. Knowing the angle φ_i, you can calculate the value of the angle β from the formula (3) and build a graph of dependence $Sin(\beta)$ from the angle of rotation of the boom φ_i (рис. 3whose data is used in the calculations.

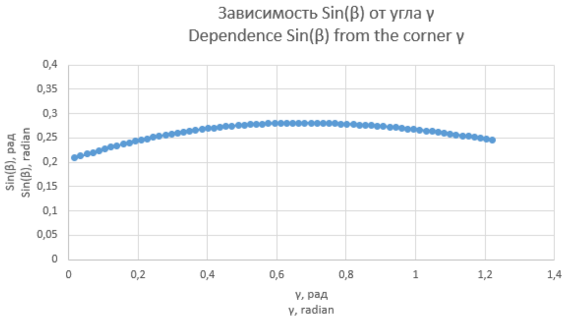

Рисунок 3 Зависимость $Sin(\beta)$ от угла поворота стрелы φ_i

Figure 3 Dependence of $Sin(\beta)$ on the angle of rotation of the boom φ_i

Источник: собственные вычисления автора

Source: own calculations

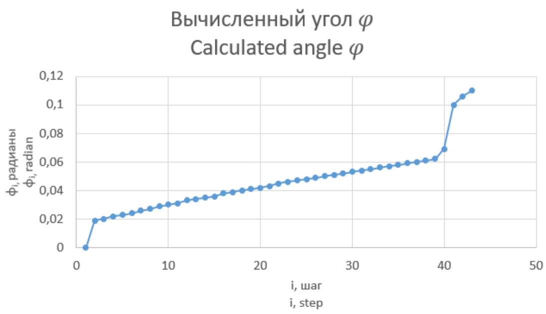

Рисунок 4. Рассчитанное значение φ_i в радианах

Figure 4. Calculated value angle φ_i in radian

Источник: собственные вычисления автора

Source: own calculations

Preliminary calculations using the MathCad program made it possible to obtain the kinematic and dynamic parameters of the energy-saving hydraulic drive of the boom lifting mechanism. Figure 4 shows the dependence of the angle of rotation of the boom on time, the angle of rotation increases smoothly at first, and then sharply increases, which is explained by the kinematic properties of the four-link mechanism The graphs of the working fluid pressure dependencies (Fig. 5) showed that the connection of an energy-saving damping device to the hydraulic drive of the boom lifting mechanism allows to reduce the peak pressure in the hydraulic line of the piston cavity of the hydraulic cylinder during transient modes by 1.5-1.6 times.

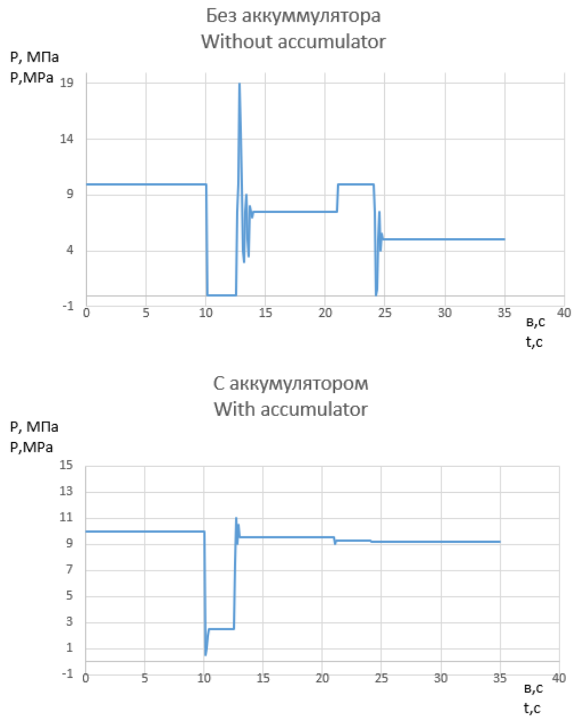

Рисунок 5. Теоретические зависимости давления в гидросистеме при использовании энергосберегающего демпфирующего устройства

Figure 5. Theoretical dependences of pressure in the hydraulic system when using an energy-saving damping device

Источник: собственные вычисления автора

Source: own calculations

Based on the statistical analysis of the experimental maximum mean values of the working fluid pressure obtained on the laboratory bench, the following coefficients were calculated (Tables 1 and 2).

All of the listed coefficients are close to 1, which indicates a fairly close relationship between the two datasets. It was also found empirically that the energy-saving hydraulic drive of the boom lifting mechanism of the manipulator stores about 30% energy when lowering the boom with a load and returns it back when the load is subsequently lifted.

Таблица 1

Статистические данные анализа давления системы без аккумулятора и с аккумулятором

Table 1

System pressure analysis statistics without accumulator and with accumulator

	Без аккумулятора | Without accumulator	*С аккумулятором | With accumulator*
Название | Name	Значение | Value	
Среднее | Average	17,933	10,460
Стандартная ошибка | Standard error	0,3092	0,3199
Медиана | Median	17,5	10,235
Стандартное отклонение | Standard deviation	1,0255	1,061
Дисперсия выборки | Sample variance	1,0518	1,1259
Эксцесс | Excess	0,09976	-0,3240
Асимметричность | Asymmetry	0,96401	0,5128

Источник: собственные вычисления автора

Source: own calculations

Таблица 2

Коэффициенты, полученные в ходе дисперсионного и корреляционного анализа

Table 2

Coefficients obtained in the course of analysis of variance and correlation

Название| *Name*	Значение | Value
Коэффициент регрессии | Regretion coefficient	0,974
R-квадрат | R-square	0,948
Коэффициент Спирмена | Spearman coefficient	0,9736
Коэффициент Пирсона | Pearson coefficient	0,974

Источник: собственные вычисления автора

Source: own calculations

The use of a new energy-saving hydraulic drive for the boom lifting mechanism of a forestry manipulator allows to significantly smooth out pressure surges in the hydraulic system, in addition, in transient modes, high-frequency pressure fluctuations are eliminated, which cause fatigue destruction of metal structures of the manipulator elements. The pressure surges of the working fluid do not exceed the setting pressure of the safety valves, which do not convert the hydraulic energy of the fluid flow into thermal energy, and the hydraulic system of the manipulator does not overheat.

Thus, reducing the dynamic loading and energy intensity of the working processes of the boom lifting mechanism of the forestry manipulator by justifying the parameters of the energy-saving damping device of the hydraulic drive makes it possible to increase the reliability of the manipulator, reduce energy costs, as well as the downtime for repairs due to the failure of hydraulic equipment.

Conclusion

1. Analysis of research on technological, dynamic, and kinematic characteristics of manipulator-type machines showed that, due to high dynamic loads, failures of high-pressure hoses are 29.7–56%, and hydraulic cylinders 14.0–24.1%.
2. To reduce the dynamic loading and energy intensity of the working processes of timber carriers during loading and unloading operations, a new energy-saving hydraulic drive of the mechanism for lifting the boom of a forest manipulator is proposed.
3. A mathematical model of the process of lifting a bundle of timber when loading by a manipulator onto a timber carrier has been developed and implemented, it has been established that the energy-saving hydraulic drive of the boom lifting mechanism reduces the peak pressure by 1.5-1.6 times, stores about 30% of energy when lowering the boom with a load and returns it back when the load is subsequently lifted.

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