Simulation of loading dynamics and hydrodynamics of drives of forest machine working bodies

N Yu Yudina¹*, R V Yudin² and A I Maksimenkov³

¹Department of Information Technology, Faculty of Computer Science and Technology, Voronezh State University of Forestry and Technologies named after G.F. Morozov, 8 Timiryazev Street, Voronezh, 394087, Russian Federation
²Department Forestry Mechanization and Machine Design, Faculty of Mechanical Engineering, Voronezh State University of Forestry and Technologies named after G.F. Morozov, 8 Timiryazev Street, Voronezh, 394087, Russian Federation
³Department of Forestry, Metrology, Standardization and Certification, Faculty of Forestry Industry, Voronezh State University of Forestry and Technologies named after G.F. Morozov, 8 Timiryazev Street, Voronezh, 394087, Russian Federation

*E-mail: YudinaNY@vgltu.ru

Abstract. The need to take into account the dynamic processes that occur when performing technological operations using the working bodies of forest machines with a hydraulic drive is due to the fact that its absence leads to a distortion of the obtained results, since the kinematic schemes do not take into account the influence of hydraulic fluid in the elements of the hydraulic system. In this regard, it became necessary to develop models that take into account the dynamics of working body loading and the hydrodynamics of its drives. It is recommended to consider the influence of the process dynamics by introducing dynamic factors. The resulting mathematical model allows you to determine the actual values of the hydraulic cylinder, taking into account the errors associated with fluid leakage. As a result of the study, it was possible to connect the pressure change in the hydraulic cylinder, the movement of the piston seals in the hydraulic cylinder and the rotor of the hydraulic motor, as well as the appearance of inertial forces. The connecting component is the balance of fluid and hydraulic system volumes.

1. Introduction

In the Russian timber industry, timber is harvested by logging machines equipped with hydraulic manipulators [1-3]. It is possible to increase the indicators of the technical level of hydraulic manipulators due to deeper studies of the kinematics, dynamics and layout of the hydraulic cylinders for lifting the boom group. One of the most important elements of the working bodies of forest machines is the hydraulic drive. Hydraulic drive is a set of devices designed to transfer mechanical energy and transform motion by a fluid.

Most often, when designing and describing a hydraulic drive system for forest machines, a static load on hydraulic motors and the response of the system to dynamic loads is used [4, 5]. The number of hydraulic motors in the system reaches 14, including hydraulic motors - up to 4, the number of pumps - up to 3, distributors - up to 9 [6-9]. The operation analysis of the hydraulic system can be represented as a sequential analysis of its fragments, each of which contains no more than three hydraulic motors and one pump. According to the operation principle, the hydraulic units in different schemes are
completely identical, therefore the mathematical description of its work will be the same [9-10]. Based on these considerations, we will consider a fragment of the drive hydraulic system, which includes 2 hydraulic cylinders and three distributors.

2. Materials and methods
When designing and describing the operation of a hydraulic drive system, a static view of the load on hydraulic motors and the response of the hydraulic system to these loads is usually used. It is believed that the pressure in the hydraulic system is instantly set such that the hydraulic cylinders and motors will begin to overcome the applied loads and start moving. The movement speed of the output components of the hydraulic motors is assigned in advance and it is provided by the flow rate that the pump delivers to the hydraulic system. So, in the absence of loads in the system, a low pressure is established. At the design load and the selected working dimensions of the hydraulic cylinders, the design pressure must act in the system. The designer increases the initial static load: according to the recommendations of [11] - by 1.15–1.3 times and other sources, this load safety factor should be from 1.5 to 2.

The described approach allows you to be calm about a sufficiently correct choice of the dimension of hydraulic motors and other units and gives some averaged values of the pressure in the system and the speeds of the executive movements of the hydraulic motors. Within the framework of the ongoing research, this is completely insufficient; the problem must be solved more accurately.

It is understood that this refined solution should give not the average expected, but actual, depending on time, the pressure in the system and the displacements and speeds of the output components of the hydraulic motors. In this case, not only the possible change in the external static load, which was analyzed in the previous sections, should be taken into account, but also the additional inertial load caused by the uneven movement of the masses connected to the output components of the hydraulic motors. Mathematically, this means that it is necessary to link three simultaneously occurring processes:
- change in pressure in the hydraulic system;
- movement of the pistons in the hydraulic cylinders and the rotor of the hydraulic motor, associated with the current change in pressure;
- the emergence of inertial forces, depending on the accelerations, which, in turn, are the second derivative of the displacement.

The link between these three processes is the balance of fluid and hydraulic volumes. This means that the volume of fluid that has entered the hydraulic system from the pump during the time interval, where by that moment there was already some initial pressure $P_{nah}$, must find a place in the hydraulic system by moving the pistons and turning the rotor of the hydraulic motor. It should be taken into account that part of the volume received will be simultaneously spent on:
- compensation for inevitable leaks in the seals of the included hydraulic units;
- compensation for the compression of the fluid that was already in the hydraulic system at the beginning of the interval $dt$;
- filling of the additional volume of the hydraulic system due to deformation of the walls of rigid and flexible lines, if an increase in pressure $dt$ occurs in the interval $dp$.

The average pressure in the system in the $dt$ interval will be $(P_{nah} + dp/2)$, this pressure will overcome the external static and inertial load. If the pressure in the system is not enough to overcome the load, there will be no movement on the hydraulic motors. In this case, the $dp$ values will be significant (the pressure in the system rises rapidly). If the pressure in the system rises above the setting pressure of the safety valve, it will open slightly, and the volume balance will need to take into account the amount of liquid that will leave the system through the valve.

As part of the work being performed, technological objectives are pursued, quantifying the impact of dynamic processes on the duration of the operation cycle. Therefore, the initial hypothesis is taken that

$$ K_{dn} = \frac{t_s + t_{st} + t_d}{t_{st}} $$

(1)
where \( t_a \), \( t_d \) is the duration of transient processes in the considered mechanism, during acceleration and deceleration under the given conditions of its loading, \( s \); \( t_{st} \) is the duration of the process at steady state when the angular or linear speed of the drive component is constant, \( c \).

When building dynamic models of the processes of interaction between the working body and the subject of labor, it is proposed to use the following approaches:

- the working body is considered as a dynamic system with several discrete masses;
- bringing the masses to the desired point is carried out on the basis of the condition of equality of kinematic energies: the reduction points of \( T_{rp} \) and the total kinematic energy of all components of the dynamic system \( \sum_{i=1}^{y} T_i \);
- the forces are brought to a given point subject to the condition: the power at the considered possible displacement from the reduced forces is equal to the sum of the powers developed by the actual forces applied to the components of the dynamic system;
- the law of motion of the point of reduction is described by the Lagrange equations of the \( P \) order, representing, in the general case, a system of second-order nonlinear differential equations;
- if necessary, the linearization of these equations can be performed by decomposition into a Taylor series or in some other way;
- for a quantitative assessment of the hydrodynamics of the drives of the working bodies, a system of differential equations is used, taking into account the phenomena occurring in three simultaneously occurring processes:
  a) the pressure changes in the hydraulic system ensuring the movement of the working body, taking into account, among other things, leakage through seals, compression of the working fluid, deformation of the walls of rigid and flexible sections of hydraulic lines;
  b) the movement of pistons in hydraulic cylinders associated with the current pressure change;
  c) the change of inertial forces, depending on the moments of inertia and acceleration, which, in turn, depend on possible movements;
- for all movements of hydraulic components, the balance of the volume of fluid in the hydraulic system elements must be observed;
- to solve the system of differential equations describing the elements of the hydraulic drive, numerical integration is used using the method of successive approximations, with preliminary linearization of the equations; the result of a numerical solution of the system of equations will be the results of a quantitative assessment of pressure, displacements and speeds of movement of the working body from time to time.

The implementation of these approaches will provide quantitative information from which, in connection with the goal, we will be interested only in the duration of the transition processes during acceleration and deceleration. However, the rest of the information may be useful for solving other applied problems encountered in practice [3-5].

As a generalized design scheme, we take a dynamic two-component manipulator system, from which, if necessary, most special cases can be obtained.

The calculated circuit diagram of the two-component manipulator is shown in figure 1. As it can be seen from figure 1 the dynamic manipulator circuit is a dynamic system with five discrete lumped masses (kg):
- reduced to cargo weight (\( m_{gr} \));
- the mass of the handle with the center of mass in the point \( S_p - m_6 \);
- the mass of the hydraulic actuator handle, concentrated in the point \( S_{a2} - m_{a2} \);
- the mass of the boom drive cylinder (point \( S_{a1} - m_{a1} \).

As the points of reduction of the masses we choose: 1) the center of the hinged attachment of the rod of the hydraulic cylinder of the handle drive to its end (point F) and 2) the center of the hinged attachment of the rod of the hydraulic cylinder of the boom drive cylinder to the boom (point E). As will be shown below, such a choice of reduction points is of great practical importance. We also note that in the dynamic system in figure 1 the moments of inertia also act: \( J_6 \) – the moment of inertia of the boom.
relative to the axis passing through the center of mass (point $S_c$), kg/m$^2$; $J_p$ – is the moment of inertia of the handle relative to the axis passing through the tons. $S_p$ kg/m$^2$.

**Figure 1.** The calculated circuit diagram of the two-component manipulator.

Before we establish the analytical connections of the linear velocities of the centers from the acting factors, we establish the cases of motion of the components of the manipulator that are possible in the practice of operation.

When using a manipulator to load trees into a delimbing machine, the following cases will be:

1) the arm of the manipulator is stationary ($\beta = \text{const}$), and the handle under the action of the hydraulic cylinder of the drive works to raise or lower ($\alpha = V_{az}$);

2) the handle of the manipulator is not movable ($\alpha = \text{const}$), and together with the boom under the action of the hydraulic cylinder of the drive works to raise or lower ($\beta = V_{az}$);

3) the boom under the action of the hydraulic cylinder of the drive works to raise or lower ($\beta = V_{az}$) and the handle under the action of its hydraulic cylinder works to lower or raise ($\alpha = V_{az}$).

Consider each case in succession and try to create a generalized mathematical model of mass reduction, and then force. When the hydraulic cylinder is raised to the handle (V piece), the handle rotates around the counterclockwise direction.

The angular velocity is given by the formula:

$$\omega_p = \beta_n \frac{V_{sh}}{x_p}, \quad \frac{V_{sp}}{x_p} = \omega_p l_2, \quad \frac{V_{D}}{x_p} = \omega_p l_2$$

The kinematic energy of a component that is can be obtained by the formula:

$$\frac{V_{sh}}{x_p} = \frac{I_2}{x_p^2} [\arcsin \left( \frac{x_p^2 + x_p^2 + x_p^2}{2x_p} \right) - \arcsin \left( \frac{x_p^2 + x_p^2}{2x_p} \right)]$$

$x_p$ – piston stroke, m; $x_p = V_{nc}t$. After setting (3) in (2), we obtain that all linear velocities of mass center points will be functions of the parameters of the piston movement of the hydraulic cylinder, the drive and the time.

The kinematic energy of a component that is can be obtained by the formula:
\[
T = \frac{m_p V_F^2}{2} + \frac{m_p V_{sp}^2}{2} + \frac{J_{sp} \omega_p^2}{2}
\]  
(4)

The moment of inertia of the handle relative to the axis passing through the center of mass (point \( S_p \)) will be
\[
J_{sp} = m_p \frac{l_p}{12}
\]  
(5)

Then, taking into account (2) and (3), we will have (6)
\[
T = \frac{1}{2} \left( m_p, l_p^2 + m_p, l_{sp}^2 + m_p, l_{2p}^2 / 12 \right) \times \int_{\text{sp}} \arcsin \left( \frac{a_1^2 + c_p^2 S_{m1}}{2 a_1 c_p} \right) - \arcsin \left( \frac{a_1^2 + c_p^2 (S_{m1} + X_p)}{2 a_1 c_p} \right) \right) V_{sh}
\]  
(6)

expression for the kinetic energy of the mass reduction point (point \( F \))
\[
T_{sp} = \pm \frac{m_{sp} V_{E}^2}{2}
\]  
and taking into account (2) and (6)
\[
T_{sp} = \frac{m_{sp} l_p}{2} \times \int_{\text{sp}} \left( \arcsin \left( \frac{a_1^2 + c_p^2 S_{m1}}{2 a_1 c_p} \right) - \arcsin \left( \frac{a_1^2 + c_p^2 (S_{m1} + X_p)}{2 a_1 c_p} \right) \right) \right) v_{sp}^2
\]  
(7)

To obtain the expression given to the mass, let's equate (6) and (7)
\[
\left( m_{gp} l_p^2 + m_p l_{2c}^2 + m_p, l_{p}^2 / 12 \right) = \pm m_{sp} l_p^2
\]  
whence finally
\[
m_{sp} = \frac{1}{l_p} \left( m_{gp} l_p^2 + m_p l_{2c}^2 + m_p l_{p}^2 / 12 \right)
\]  
(8)

As follows from (8), the handle weight, given to the volume, does not depend on the movement parameters.

When the lowering cylinder is working, only the direction of the velocity vectors \( \vec{V}_E, \vec{V}_{sp}, \vec{V}_D \) changes, and the other arguments remain the same.

Taking for the point of casting the center of the hinged attachment rod of the hydraulic cylinder of the boom drive to it we can write:
\[
V_E = \omega_c r_{E}; \ V_{sc} = \omega_c J_{sc}; \ V_{a2} = \omega_c r_{a2}; \ V_{sp} = \omega_c r_{gp}; \ V_D = \omega_c r_{p}; \ \omega_c = \beta_n V_{sh}
\]  
(9)

where \( \beta_n \) is the boom drive parameter, rad / m, equal to
\[
\beta_n = \frac{1}{l_p} \left( \arcsin \left( \frac{a_1^2 + c_p^2 S_{m1}}{2 a_1 c_p} \right) - \arcsin \left( \frac{a_1^2 + c_p^2 (S_{m1} + X_p)}{2 a_1 c_p} \right) \right),
\]  
(10)

\( x_c \) is stroke of the boom hydraulic cylinder rod, m;
\[
x_c = V_{sh} t.
\]

After substituting (10) into (9), we obtain the linear velocities of displacement of the centers of mass, which are functions of the rod movement parameters and the boom movement time, i.e. \( \vec{V}_l = f(x_l, V_{sh}, t_l) \). In addition to speeds, it is necessary to know the moments of boom inertia (\( J_c \)) and the arm (\( J_p \)) relative to the axis passing through the centers of its masses. For this we write
\[
J_c = m_c \frac{l_c^2}{3}
\]  
(11)

\[
J_p = m_p \frac{l_p^2}{12} + m_{sp} l_p^2
\]  
(12)

Now you can write the expression for the kinetic energy of the system: boom – handle
\[
T = \frac{(J_c + J_p) \omega_c^2}{2} + m_c \frac{V_{E}^2}{2} + m_{a2} \frac{V_{a2}^2}{2} + m_{sp} \frac{V_{sp}^2}{2} + m_p \frac{V_{p}^2}{2}
\]  
(13)

Kinetic energy cast points E will
\[
T_{sp} = \frac{m_{sp} V_{E}^2}{2}
\]  
(14)

From the condition of equality of kinetic energies, taking into account (9), we can write:
\[
\frac{1}{2} \left[ (J_c + J_p) \omega_c^2 + m_c \frac{V_{E}^2}{2} + m_{a2} \frac{V_{a2}^2}{2} + m_{sp} \frac{V_{sp}^2}{2} + m_p \frac{V_{p}^2}{2} \right] \omega_c = \frac{1}{2} \left( m_{sp} l_p^2 \right) \omega_c
\]  
(15)

where, with the substitution (14) and (15), we get
When the boom actuator hydraulic cylinder is on the lift and the crank actuator hydraulic cylinder is on the rise, the boom rotates around the center of the hinged attachment to its frame (point A), and the mass point of the crank arm (point F) performs a complex movement: it moves around the radius circumference around the point C and, at the same time, around a circle of radius \( r_F \) with the center in point A.

The true velocity of the F point in the new dynamic system will be equal to \( \overrightarrow{V}\hat{t}_F=\overrightarrow{V}\hat{t}_{np}+\overrightarrow{V}\hat{t}_{pr} \) (relative to point A), and its module is found from the formula

\[
\overrightarrow{V}\hat{t}_F=\sqrt{\overrightarrow{V}\hat{t}_{np}^2+\overrightarrow{V}\hat{t}_{pr}^2}.
\]

To determine the values of the remaining linear velocities and the angular velocity of the boom rotation

\[
\begin{align*}
V_{sp} &= \omega_c l_{zc}; V_{z2} = \omega_c r_{z2}; \omega_c = \beta_p V_{st}; \\
V_F &= \beta_p V_{sp} l_1; V_F^{pr} &= \omega_c r_F^{pr}.
\end{align*}
\]

For the point A, in this case, the following expressions are recommended:

\[
J_c = m_c \frac{l_c^2}{2}.
\]

Kinetic energy of the considered dynamic system

\[
T = \frac{J_c \omega_c^2}{2} + m_c \frac{V_{zc}^2}{2} + m_{z2} \frac{V_{z2}^2}{2} + m_F \frac{V_F^{pr}^2}{2}.
\]

Kinetic energy of the cast point will be

\[
T_E = \frac{m_F V_F^{pr}^2}{2}.
\]

From the condition of equality of kinetic energies we write

\[
\frac{J_c \omega_c^2}{2} + m_c \frac{V_{zc}^2}{2} + m_{z2} \frac{V_{z2}^2}{2} + m_F \frac{V_F^{pr}^2}{2} = m_F V_F^{pr}^2.
\]

From where, taking into account (17) and (18), we get

\[
m_F^{pr} = \left( m_c \frac{l_c^2}{3} + m_c l_{zc} + m_{z2} r_{z2}^2 + m_F V_F^{pr^2} \right) \sqrt{f_p^2 V_{shtr}^2 + f_p^2 V_{shtr}^{2 np}}.
\]

Thus, we have obtained expressions (8), (15) and (21), which allow us to find the value of the reduced masses for all three cases encountered in the practice of operating a two-component manipulator of a stationary delimbing plant.

### 3. Results and discussion

After the reduction of the masses and forces is done, a dynamic model of loading the working bodies and other components of the analyzed mechanism can be made.

In the general case, the equation of center motion of the reduced mass is written in the following form.

\[
\frac{m_F^{pr} \frac{dV_F^{pr}}{dt}}{m_F^{pr}} + \frac{V_{F^{pr}}^2 \frac{dm_F^{pr}}{ds}}{ds} = IR^{np}.
\]

In our case, \( m_F^{np} = const \). Consequently,

\[
\frac{m_F^{pr} \frac{dV_F^{pr}}{dt}}{m_F^{np}} = IR^{np}.
\]

in (22) and (23): \( m_F^{pr} \) is the mass reduced to a given point; \( V_F^{pr} \) – is the linear velocity of the point of reduction under the action of external forces, m/s; \( R^{pr} \) is the force reduced to the accepted line.
The task of dynamics consists in the study of equations (22) (or (23)) taking into account the initial conditions \( t = 0; S = S_0 \) – position of the point of reduction; \( V_{pr} = V_0 \) – speed of the point of reduction. It should be remembered that the foregoing is valid only for the period of steady motion. In the study of transients, in general, it is necessary to integrate (22). It is necessary to reckon with some features.

First of all, when choosing the initial conditions, it should be remembered that when \( \phi_{start} = \phi_0, t = 0, s_0 = 0 \), the system is at rest and \( V_0 = 0 \). At the reference point, you should take a moment of time very close to \( t_0 \), i.e. \( t_0 = t_0 + \Delta t \). Then all the other parameters will already have not zero, but significant values, although very close to zero.

The final values are determined by the geometrical parameters of the working body, the characteristics of the object and the quantitative values of organizational and technical factors, depending on the conditions of the specific task.

In the article Popikov P I, Chernykh A, Chetverikova I V, Rodionov D N and Menyailov K A [6], when developing dynamic models of logging processes, a generalized dynamic mathematical model of a unified hydraulic drive is presented, which is very simply reduced to a specific case. The articles [6-8] formulated recommendations for reducing the dynamics of the processes of performing the working operation methods. In most studies, the movement of the manipulator components is considered, and the parameters of the hydraulic cylinders that ensure the implementation of these operations are not given.

The presented mathematical model for calculating the parameters of hydraulic cylinders is universal. Its only drawback is that it was considered only for models of Russian-made manipulators. On the basis of the obtained model, an algorithm for constructing a software module was developed, which provided verification of this model. When carrying out calculations, we get the opportunity to explore various real and hypothetical situations. The results obtained were compared with experimental data, which made it possible to draw a conclusion about the correctness of the chosen approach.

4. Conclusion

The use of quantitative information for technological calculations, which does not take into account the dynamic processes of interaction between working bodies and the object of labor in working methods of technological processes in the forest industry, leads in a number of cases to a significant decrease in the reliability of the results. In this regard, the industry has a need to develop dynamic models that take into account the dynamics of the working body loading and the hydrodynamics of its drives.

The formulated fundamental approaches to the construction of dynamic models made it possible to find and build a unified algorithm that takes into account the load dynamics on the hydraulic system. The model distinctive feature is that it allows you to make a fairly correct choice of the dimension of hydraulic motors and calculate certain values of pressure in the system and speeds of the executive movements of hydraulic motors.

The scientific novelty of the author's approach lies in the fact that as a result of solving the problem, a mathematical model was obtained that combines three simultaneously occurring processes: pressure change in the hydraulic system; the movement of the pistons of the hydraulic cylinders and the rotor of the hydraulic motor, associated with a change in pressure; the emergence of inertial forces that depend on accelerations, which, in turn, are the second derivative of the displacement. On the basis of the obtained model, an algorithm for constructing a software module was developed, which provided verification of this model. The obtained results were compared with experimental ones, which made it possible to conclude that the chosen approach was correct.

The scientific novelty of the author's approach lies in the fact that as a result of solving the problem, a mathematical model was obtained that combines three simultaneously occurring processes: pressure change in the hydraulic system; the movement of the pistons of the hydraulic cylinders and the rotor of the hydraulic motor, associated with a change in pressure; the emergence of inertial forces that depend on accelerations, which, in turn, are the second derivative of the displacement.
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