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Selection of optimal parameters boom lifting mechanism

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Abstract. This document describes the calculation procedure for selecting the optimal parameters of the boom lifting mechanism in agricultural machinery. A boom lifting scheme was developed. Studies have been carried out to find the optimal values of linear dimensions and angle, at which the required value of the force developed by the drive when lifting the boom with a hydraulic cylinder of the specified dimensions will be minimal. Based on the results of the study, an algorithm was developed for solving the optimization of boom lifting parameters. An expression is obtained for the objective function of optimizing the boom lifting mechanism. Algorithm for selection of optimal parameters of lifting mechanism considering weight of boom, hydraulic cylinder and rod has been developed.

1 Introduction

In lifting machines, a hydraulic drive for lifting a boom is widely used. The efficiency of the lifting machine largely depends on the optimal selection of drive parameters at the design stage. Reducing the maximum force required to lift the boom with the payload reduces the power consumption of the machine and the drive cost [1]. The monograph [2] discusses the problem of optimizing the parameters of the load lifting mechanism, without taking into account the masses of the boom and hydraulic cylinder.

The present work deals with the problem of optimal selection of coordinates of the points of attachment of the hydraulic cylinder relative to the axis of rotation of the boom based on the condition of the minimum maximum required value of the force Fmax developed by the drive at different positions of the output link taking into account the masses of the boom, hydraulic cylinder and rod. A diagram of the boom lifting mechanism is shown in Figure 1.

Hydraulic drive is used in mechanization of agricultural machinery for boom lifting, for example, on cranes or other lifting devices.

Hydraulic drive consists of several components, including pump, hydraulic cylinder, hydraulic tubes and valves. The operation of the hydraulic drive uses hydraulic pressure, which is created by the pump, and then transmitted through hydraulic tubes to the hydraulic cylinder, causing its movement.

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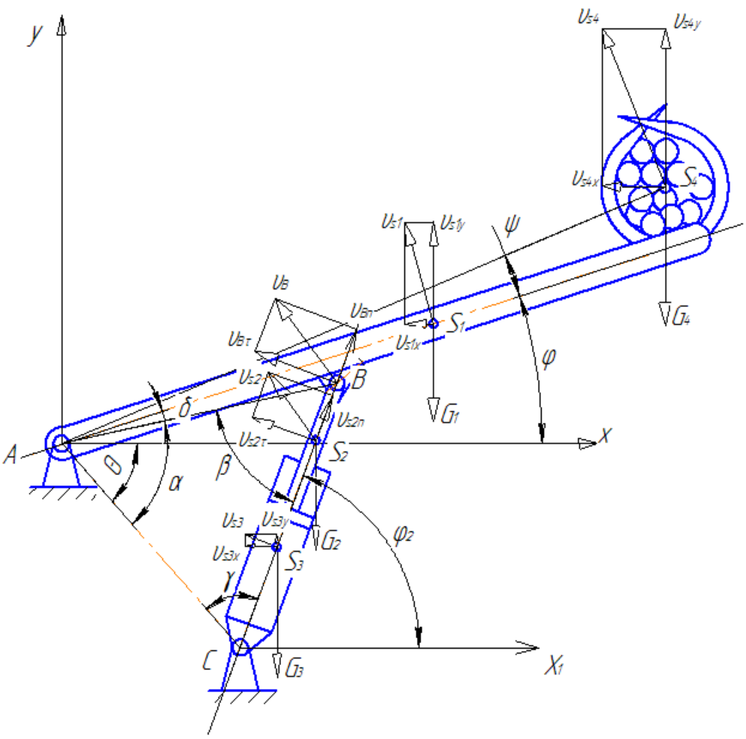

Fig. 1. Boom Lifting Mechanism Diagram.

When the boom is to be lifted, the hydraulic pump transfers hydraulic pressure through the pipes to the hydraulic cylinder, which causes it to expand and lift the boom. When the boom is to be lowered, the valves in the hydraulic drive shut off, blocking the hydraulic pressure and allowing the boom to descend under its weight or by gravit.

Hydraulic drives provide high force and control accuracy, which is especially important when lifting and moving heavy loads. They also have high reliability and durability, which is especially valuable in agricultural conditions, where machines are often subjected to intensive operation and difficult conditions.

Hydraulic drives play an important role in the mechanization of agricultural machinery during boom lifting and other lifting operations. They provide high strength, accuracy and reliability, which helps to improve efficiency and safety in agriculture.

2 Materials and methods

At specified values of boom lifting angle minimum (φ_{min}) and maximum (φ_{max}), dimensions of hydraulic cylinder minimum ($l_{BC.\min}$) and maximum ($l_{BC.\max}$), angles of δ and ψ determining position of hinge A and load relative to boom axis, the task of selecting optimal parameters of boom lifting mechanism can be formulated in the following form. Find the optimal values of linear dimensions l_{AB} and l_{AC} and θ angle, at which the required value of the force $F_{\max}$ developed by the drive when lifting the boom with a hydraulic cylinder of the specified dimensions will be minimal [3]:

$$F_{\max}(\varphi,\varphi_{\min},\varphi_{\max},l_{AB},l_{AC},\theta,\delta,\psi,l_{BC.\min},l_{BC.\max}) \underset{l_{AB}\to l_{AB.opt},\, l_{AC}\to l_{AC.opt},\, \theta\to\theta_{opt}}{\longrightarrow} F_{\min} \quad (1)$$

The minimum function $F_{\max}(l_{AC}, l_{AB}, \theta)$ in the draft design can be determined from the minimum load reduced to the piston $F_{\text{n.pr}}$:

$$l_{AB.opt}, l_{AC.opt}, \theta_{opt}(F_{\max} \to F_{\min}) = l_{AB.opt}, l_{AC.opt}, \theta_{opt}(F_{\text{n.pr}} \to F_{\min}) \quad (2)$$

3 Results and discussion

Gravity forces of boom G_1, rod G_2, piston G_3 and load G_4 will be considered as loads. When reducing forces, according to the Lagrange equations of the II kind, the condition of equality of elementary works (or capacities) must be observed. In projections on the coordinate axes, you can write [4]:

$$F_{\text{n.pr}} = \sum_{i=1}^{4} \left(F_{ix} \frac{\upsilon_{ix}}{\upsilon_{pr}} + F_{iy} \frac{\upsilon_{iy}}{\upsilon_{pr}} \right) \quad (3)$$

Where F_{ix} and F_{iy} are projections, respectively, on the x axis and on the y axis, the external force applied to the i-th link; υ_{ix} and υ_{iy} - projections, respectively on the x axis and on the y axis, of the velocity of the center of mass of the i-th link; υ_{pr} - relative speed of the piston; i = 1, … , 4; 1 - boom; 2 - rod; 3 - cylinder; 4 - cargo.

Since the projections of gravity on the x-axis are zero, equation (3) can be written as follows:

$$F_{\text{n.pr}} = \sum_{i=1}^{4} G_i \frac{\upsilon_{iy}}{\upsilon_{pr}} \quad (4)$$

Let's bring the gravity of the boom to the piston. The boom rotates around point A with an angular velocity of ω_1, and its center of mass (point S_1 in Figure 1) moves at a circumferential velocity of υ_{S1}. The υ_{S1} speed can be decomposed into vertical υ_{S1y} and horizontal υ_{S1x} components. The angle between the absolute speed of the υ_{S1} and its vertical component is υ_{S1y} equal to the angle of φ inclination of the boom. [5] The relative speed of the piston is equal to the normal component of the $\upsilon_{B.n}$ speed of point B, which is common to the rod and boom. Point B, as the point belonging to the boom, moves circumferentially with respect to point A at the circumferential velocity of the υ_B. The velocity of point B can be found through the angular velocity of the ω_1. As a result of simple transformations, we get:

$$F_1 = G_1 \frac{l_{AS1} \cos\varphi}{l_{AB} \sin\beta} \quad (5)$$

Let's bring the gravity of the load to the piston. On the forest loader, the cargo is pressed by the jaw to the boom and, together with the boom, rotates relative to point A. The center of mass of the cargo (in Figure 1, point S_4) moves at the circumferential speed of the υ_{S4}. The υ_{S4} speed can be decomposed into vertical υ_{S4y} and horizontal υ_{S4x} components. The angle between the absolute velocity of the υ_{S4} and its vertical component is υ_{S4y} equal to the sum of the angles of φ and ψ. Conducting similar reasoning as for the center of mass of the boom, we get the expression for the reduced gravity of the cargo:

$$F_4 = G_4 \frac{l_{AS4} \cos(\varphi + \psi)}{l_{AB} \sin\beta} \quad (6)$$

Let's bring the gravity of the rod to the piston. The rod together with the hydraulic cylinder rotates around the point B with the angular speed of the ω_2 and simultaneously moves translationally relative to the cylinder. The center of mass of the rod (point S_2 in Figure 1) moves at an absolute υ_{S2} speed, which can be decomposed into a portable speed $\upsilon_{S2\tau}$ rotational movement and a relative one - υ_{S2n}. The projection of the absolute υ_{S2} velocity onto the y-axis is found as the algebraic sum of the projection of the portable $\upsilon_{S2\tau}$ and the relative υ_{S2n} of the velocities onto the y-axis. The angle between the portable speed of the $\upsilon_{S2\tau}$ and the y axis is equal to the angle of the φ2. The angle between the relative speed of the υ_{S2n} and the y axis is equal to the angle (90- φ2). [6] The portable speed of the $\upsilon_{S2\tau}$ can be expressed through the angular speed of the ω_2, and the relative speed of the υ_{S2n} is equal to the normal component of the υ_{S2n} speed of point B. As a result of simple transformations, we get:

$$F_2 = G_2 \frac{\cos\beta(l_{BC} - l_{BS2})\cos\varphi_2}{l_{BC}\sin\beta} + \sin\varphi_2 \quad (7)$$

Let's bring the gravity of the cylinder to the piston. The cylinder rotates around point B with an angular velocity of ω_2. The center of mass of the cylinder (point S_3 in Figure 1) moves at absolute υ_{S3} speed. The υ_{S3} speed can be decomposed into vertical υ_{S3y} and horizontal υ_{S3x} components. Angle between absolute speed of υ_{S3} and its vertical component is υ_{S3y} equal to angle of φ_2 inclination of hydraulic cylinder. As a result of simple transformations, we get:

$$F_3 = G_3 \frac{l_{CS3}\cos\beta\cos\varphi_2}{l_{BC}\sin\beta} \quad (8)$$

Let's take the independent variable angle φ lifting the boom. We exclude the angles of β and φ_2 from the equations (5-8) using the sine theorem, and the distance l_{BC} using the cosine theorem:

$$\sin\beta = \frac{l_{AC}}{l_{BC}}\sin\alpha, \quad (9)$$

$$l_{BC}^2 = l_{AB}^2 + l_{AC}^2 - 2l_{AB}l_{AC}\cos\alpha, \quad (10)$$

And the relationship between angles:

$$\cos\beta = \sqrt{1-(\sin\beta)^2}, \quad (11)$$

$$\alpha = \varphi + \theta - \delta, \quad (12)$$

$$\varphi_2 = \pi - (\gamma + \theta), \quad (13)$$

$$\sin\gamma = \frac{l_{AB}}{l_{BC}}\sin\alpha, \quad (14)$$

$$\cos\gamma = \sqrt{1-(\sin\gamma)^2}, \quad (15)$$

As a result, we get:

$$F_1 = G_1 \frac{l_{AS1}l_{BC}\cos\varphi}{l_{AB}l_{AC}\sin(\varphi+\theta-\delta)}, \quad (16)$$

$$F_4 = G_4 \frac{l_{AS4}l_{BC}\cos(\varphi+\psi)}{l_{AB}l_{AC}\sin(\varphi+\theta-\delta)}, \quad (17)$$

$$\begin{aligned} F_2 = {} & G_2 \frac{(l_{BC} - l_{BS2})}{l_{AC}\sin(\varphi+\theta-\delta)}\sqrt{1-\frac{l_{AC}^2}{l_{BC}^2}\left[\sin(\varphi+\theta-\delta)\right]^2} \times \\ & \times\left[\frac{l_{AB}}{l_{BC}}\sin\theta\sin(\varphi+\theta-\delta) - \cos\theta\sqrt{1-\frac{l_{AB}^2}{l_{BC}^2}\left[\sin(\varphi+\theta-\delta)\right]^2}\right] + \\ & + G_2\left[\frac{l_{AB}}{l_{BC}}\cos\theta\sin(\varphi+\theta-\delta) + \sin\theta\sqrt{1-\frac{l_{AB}^2}{l_{BC}^2}\left[\sin(\varphi+\theta-\delta)\right]^2}\right] \end{aligned} \quad (18)$$

$$F_3 = G_3 \frac{l_{CS3}}{l_{AC} \sin(\varphi + \theta - \delta)} \sqrt{1 - \frac{l_{AC}^2}{l_{BC}^2} [\sin(\varphi + \theta - \delta)]^2} \times \\ \times \left[\frac{l_{AB}}{l_{BC}} \sin\theta \sin(\varphi + \theta - \delta) - \cos\theta \sqrt{1 - \frac{l_{AB}^2}{l_{BC}^2} [\sin(\varphi + \theta - \delta)]^2} \right], \quad (19)$$

Where

$$l_{BC} = \sqrt{l_{AB}^2 + l_{AC}^2 - 2 l_{AB} l_{AC} \cos(\varphi + \theta - \delta)}. \quad (20)$$

At specified parameters of boom lifting mechanism: minimum (φ_{min}) and maximum (φ_{max}) values of boom lifting angle, minimum ($l_{BC.min}$) and maximum ($l_{BC.max}$) dimensions of hydraulic cylinder, angles of δ and ψ determining position of hinge A and load relative to boom axis, independent parameter will be one of three: linear dimensions l_{AB} or l_{AC}, or θ angle (Figure 1). The relationship between l_{AB}, l_{AC}, and θ parameters is determined by the cosine theorem for boom boundary positions [7]:

$$l_{BC.min}^2 = l_{AB}^2 + l_{AC}^2 - 2 l_{AB} l_{AC} \cos(\varphi_{min} + \theta - \delta), \quad (21)$$

$$l_{BC.max}^2 = l_{AB}^2 + l_{AC}^2 - 2 l_{AB} l_{AC} \cos(\varphi_{max} + \theta - \delta). \quad (22)$$

Let's take the θ angle as an independent parameter, then the linear dimensions l_{AB} and l_{AC} will be functions of the θ angle. Solving the system of equations (21) and (22) with respect to l_{AB} and l_{AC}, we get two solutions [8]:

$$l_{AB.1} = \sqrt{\frac{K_2 + \sqrt{K_2^2 - 4K_1^2}}{2}}, \quad (23)$$

$$l_{AC.1} = \frac{K_1}{l_{AB.1}}, \quad (24)$$

$$l_{AB.2} = \sqrt{\frac{K_2 - \sqrt{K_2^2 - 4K_1^2}}{2}}, \quad (25)$$

$$l_{AC.2} = \frac{K_1}{l_{AB.2}}, \quad (26)$$

Where

$$K_1 = \frac{l_{BC.max}^2 - l_{BC.min}^2}{2[\cos(\varphi_{min} + \theta - \delta) - \cos(\varphi_{max} + \theta - \delta]}, \quad (27)$$

$$K_2 = l_{BC.min}^2 + K_1 \cos(\varphi_{min} + \theta - \delta). \quad (28)$$

With formulas (23) to (26) it is easy to show that

$$l_{AB.1} = l_{AC.2} \quad (29)$$

$$l_{AB.2} = l_{AC.1} \quad (30)$$

$$l_{AB.1} > l_{AC.1} \quad (31)$$

$$l_{AB.2} < l_{AC.2} \quad (32)$$

Taking the interval of changing the α angles and φ the specified:

$$0 < \alpha < \pi, \quad (33)$$

$$\varphi_{min} \le \varphi \le \varphi_{max}, \quad (34)$$

Find the range of change in the θ angle:

$$\delta - \varphi_{min} < \theta < \pi + \delta - \varphi_{max}. \quad (35)$$

In this formulation (given φ_{min}, φ_{max}, $l_{BC.min}$, $l_{BC.max}$, δ, and ψ), the objective function is a function of the two variables φ and θ, and the optimization problem can be written as follows:

$$\begin{cases} (F_{n.pr}(\varphi,\theta))_{max} \to F_{min} \\ \quad \theta \to \theta_{opt} \\ \quad \varphi_{min} \le \varphi \le \varphi_{max} \\ \delta - \varphi_{min} < \theta < \pi + \delta - \varphi_{max} \end{cases}, \tag{36}$$

Where

$$F_{n.pr} = F_1 + F_2 + F_3 + F_4, \tag{37}$$

And the components of F_1,F_2 ,F_3 and F_4 of reduced force F_n are determined by formulas (16) to (20).

4 Conclusion

The algorithm for solving the given optimization problem consists in the following: for each value of the θ_i angle from the interval determined by the inequality (35), we find the maximum value of the reduced load $(F_{n.pr.max})_i$, then from the set of maximum values $(F_{n.pr.max})_i$ we find the minimum $(F_{n.pr.max})_{i.opt}$ = min, which corresponds to the optimal value of the $\theta_{i.opt}$ angle. Further, formulas (27) and (28) find optimal values of $K_{1.opt}$ and $K_{2.opt}$, and formulas (23) and (24) - $l_{AB.1}$ and $l_{AC.1}$ or formulas (25) and (26) - $l_{AB.2}$ and $l_{AC.2}$

Received expression for objective function. An algorithm for solving the given optimization problem is proposed. On the basis of the described algorithm, a program for calculating the optimal parameters of the boom lifting mechanism on a computer has been developed.

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