Forest Machinery Crane Compound Scheme Synthesis: Optimization of Hydraulic Cylinder Operating Mechanisms

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ABSTRACT

Previously we published some results of the procedure for the improvement of intermediate technology and machinery performance criteria. The aim of the present paper is to describe the model of the middle level of hierarchy - the optimization of the hydraulic cylinder operating mechanisms. Drive and transmission mechanisms for forest machinery crane synthesis are discussed as optimizations for boom operating mechanisms and outboom operating mechanisms. Concentration is on the static part of this problem. Although the speed of movement of the crane elements affects productivity, it is not taken into account in this study. Future studies will take into consideration a dynamic analysis and output of machinery.

The results of this study show the necessity to separate the design of different types of forest machine cranes. For example, for the parallel crane type, a modification of the proposed algorithms is necessary.

This paper may be useful for forest machine designers as well as university students, who take courses in forest machine design..

Keywords: *design, optimization, operating mechanism, forest crane, boom, outboom.*

Figure 1. Forest machinery crane.

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INTRODUCTION

Drive and transmission mechanisms for forest machinery cranes involve a system of hydraulic cylinders and mechanical levers (Figure1). The hydraulic cylinders create working forces for turning the boom and outboom. Lever mechanisms transmit the forces to crane elements.

In an ideal case from the static point of view, the forest crane hydraulic transmission mechanisms would ensure equality of the loading moment (LM) and the driving moment (DM) at every point in the working space.

In a real case, complete equality of the LM and the DM for all of a crane's positions can not be achieved due to construction limitations and the limits of the hydraulic cylinder's standard range. Therefore, the problem of crane mechanism synthesis is formulated as an optimization: *to select transmission mechanism parameters, at which the maximum ratio of the DM to the LM is minimized, as well as more than one or equal one for the whole scope of changes for the angles:*

$$\min_{(s,d,r)} (\max[\{DM(R,n)LM(R,n)\}>1]) \quad Q_0 < Q < Q_k, \quad n_0 < n < n_k \qquad (1)$$

where

DM (Q,n)	=	the driving moment function;
LM (Q,n)	=	the loading moment function;
s	=	the hydraulic cylinder stroke;
d	=	the hydraulic cylinder diameter;
r	=	the hydraulic cylinder mounting coordinates and the sizes of lever mechanism elements;
Q_0 - Q_k	=	the range of the boom turn angles;
n_0 - n_k	=	the range of the outboom turn angles.

In addition to consideration of construction and materials on the operating parameters, it is necessary to set the corresponding area and functional limitations. This is done with a multiparameteral nonlinear programming optimization problem.

BOOM OPERATING MECHANISM

A scheme for a boom operating mechanism is presented in Figure 2.

The input data for the calculations are the key design parameters for the crane frame or "skeleton" part as we called it in our previous studies [2,3]:

- the lengths of the crane boom (l_c),
- the length of the crane outboom (lp),
- the length of the telescopic lengthener (lt),
- the length of the pillar (ho),
- the limiting angles for boom rotation (Q_0 and Q_k),
- the limiting angles for outboom rotation (n_0 and n_k).

There is also a standardized range for the hydraulic cylinders characterized by the combination of: cylinder diameter (d), stroke (s), and length (l) of the cylinders when completely retracted.

Figure 2. Scheme of the boom operating mechanism.

According to condition (1), it is required to select hydraulic cylinders and their mounting coordinates so that they ensure that the angular rotation of the boom within the set range of angles during the power stroke meet working requirements. There are no special limitations on the mounting coordinates for the hydraulic cylinder except that cylinder should be located close to the crane sections.

For calculation of the boom loading moment LM, the moment forces for the crane sections and the load weights were determined for the column-boom junction hinge (Figure 3). Note that the greatest moment (if horizontal components are not in consideration) will correspond to when the boom is in the horizontal position with the outboom at maximum extension:

For $Q_0 < Q < B/2$:

$$LM(Q) = \{Gc \bullet Lc + Gg \bullet Lg + (Gm+Gp+Q) \bullet lc\} \; x \; x \; \sin(Q) + Gp \bullet Lp + Q\,(lp+lt); \qquad (2)$$

For $B/2 < Q < Q_k$, outboom in line with boom:

$$LM(Q) = \{Gc \bullet Lc + Gg \bullet Lg + (Gm+Gp+Q) \bullet lc + {} + Gp \bullet Lp + Q \bullet (lp+lt)\} \bullet \sin(Q), \qquad (3)$$

Figure 3. Scheme for calculating the loading moment, LM, for the boom operating mechanism.

where

Q = the force of gravity acting on the load and the working device;

Gc, Gp, Gg, and Gm = the forces of gravity acting on the boom, the outboom with extension, the outboom hydraulic cylinder, and the lever mechanism, respectively;
Lc, Lg, and Lp = the distances between the sections, junction hinges and centers of gravity for: the boom; the outboom hydraulic cylinder; and the outboom, respectively.

The DM is determined in the following way:

$$Mg(Q) = Fg \bullet pg \bullet h(Q) \bullet ng, \quad (4)$$

where

Fg = the area of the boom hydraulic cylinder piston;
pg = the pressure on the hydraulic system;
ng = efficiency (ng » 0.95);
h(Q) = the arm of the force on the boom hydraulic cylinder.

The arm, h(Q), in the immovable coordinates system with its center at point O (Figure 2) is determined by the formula:

(5)

where

$x_1 = r_1$;
$y_1 = -r_2$;
$x_2(Q) = r_3 \bullet \sin(Q) + r_4 \bullet \cos(Q)$;
$y_2(Q) = r_3 \bullet \cos \bullet (Q) - r_4 \bullet \sin(Q)$.

The boom hydraulic cylinder mounting axis coordinates meet the following conditions:

(6)

(7)

where

l = the length of the hydraulic cylinder when the stroke fully retracted;
s = the hydraulic cylinder stroke.

The optimum solution is as follows:

$$\min_{(s,d,r)} (\max[Fg \bullet pg \bullet ng \bullet h(Q)/ML(Q)>1]) \quad Q_0 < Q < Q_k, \quad (8)$$

where

$r = \{r_1, r_2, r_3, r_4\}$ is the hydraulic cylinder mounting coordinates (see Figure 2) accounting for the conditions (6) and (7).

Two independent equations (6) and (7) correspond to four operating parameters for the multiple r. A solution can be found with an optimization method, for example, the random search method [4].

From an engineering point of view it is worth while to simplify the problem into a particular form. In such a form the search for a solution may be presented in the following algorithm:

1. Set the pressure for the hydraulic system pg, and the sizes for standard hydraulic cylinders (diameters d and strokes s).

2. Set the construction limits r_1 and r_4: to ensure the hydraulic cylinder is mounted a safe distance from the construction surfaces and the kinematics axis.

3. Then set the boundary constraints for r_2 and r_3.

4. Solve equations (6) and (7) for each hydraulic cylinder within its standard range. Get the corresponding optimum values of r_2 and r_3.

5. Change the angle Q with in the range from Q_0 to Q_k (for example, by 5^0 increments). Graph the DM and LM dependence on the boom turn angle for each hydraulic cylinder (Figure 4) using formulas (2) through (5). Choose the maximum value for the ratio of DM over LM.

Figure 4. Graph of the driving moment, DM, and loading moment, LM, dependence on the boom turn angle.

6. Choose the hydraulic cylinder, characterized by the minimum value for the maximum ratio of DM over LM. Fix that hydraulic cylinder and its mounting coordinates.

7. If necessary, set new values for pg, d, and s, then return to step 1.

This algorithm can be easily solved using a personal computer with any mathematical software, for example, MathCAD.

OUTBOOM OPERATING MECHANISM

The most commonly used outboom operating mechanism schemes as shown in Figure 5. The scheme shown in Figure 5a is used in the most powerful models of forest cranes. The algorithm for arranging the operating mechanism for this scheme is almost the same as for the boom operating mechanism previously described. The hydraulic lever mechanism shown in Figure 5b is more popular in modern forest cranes. For further detail of this second mechanism, the lever transmission mechanism is shown in Figure 6.

Optimization calculations for this mechanism require the same input data as for the previous mechanism.

It is necessary to select a hydraulic cylinders, its mounting coordinates, and the sizes of the lever mechanism r_1, r_2, r_3, r_4, r_5, r_6 (Figure 6) so that the angle shifts of the boom within the set range of angles on the power stroke are complete and according to formula (1). There are no additional restrictions to the mounting coordinates, other than that the hydraulic cylinder is located close to the crane surface.

Determination of the LAM for the outcome operating mechanism is more complicated than for the boom operating mechanism. The LAM is connected by both the boom turn angle, Q, and the outcome turn angle, n. To illustrate this, the four crane positions are shown in Figure 7.

Figures 7a and 7b show the stages of movement when the hydraulic level mechanism develops maximum negative moments (the hydraulic cylinder head stop is reached). In Figure 7a, the boom turn angle changes from B/2 to Q_k while the outcome maintains the same horizontal orientation, i.e. the LM is a constant and can be determined as follows:

Figure 5. Outboom operating mechanism.

Figure 6. Lever transmission mechanism.

Figure 7. Crane positions for the calculation of the loading moment, LM, for the outboom operating mechanism.

$$Ml(Q, n) = Q \bullet lp + Gg \bullet Lp$$
$$B/2 < Q < Q_k \quad (9)$$
$$n = Q - B/2$$

In Figure 7b, the value of $Q = Q_k$ and the outboom moves from the horizontal to the vertical, i.e. the LM changes according to the sine law from its maximum value to zero:

$$LM(Q, n) = Q \bullet lp + Gg \bullet Lp) \bullet \sin(n - Q_k)$$
$$n_k = B/2 < n_k, \quad (10)$$
$$Q = Q_k = \text{const.}$$

Figures 7c and 7d show the stages of movement when the hydraulic lever mechanism develops maximum positive moments (the hydraulic cylinder rod stop is reached).

In Figure 7c, the boom turn angle changes from B/2 to Q_0, the outboom maintains the same horizontal orientation, i.e. the LM is a constant and can be determined as follows:

$$LM(Q, n) = Q \bullet lp + Gg \bullet Lp, \quad (11)$$
$$Q_0 < Q < B/2$$

In Figure 7d, the value of $Q = Q_0$ and the outboom moves from the horizontal to the vertical, i.e. the LM changes from its maximum value to zero according to the sine law:

$$LM(Q, n) = Q \bullet lp + Gp \bullet Lp) \bullet \sin(n - Q_k)$$
$$Q_0 < n < Q_0 + B/2 \quad (12)$$
$$Q = Q_0 = \text{const.}$$

The maximum moment of the outboom operating mechanism (MM) is determined as follows:

$$MM(Q, n) = MC(Q, n) \bullet i(r) \quad (13)$$

where

MC(Q, n) = the moment developed by the hydraulic cylinder;
i(r) = the hydraulic lever mechanism transmission function.

The transmission function, i(r), is expressed with the hydraulic lever mechanism parameters (Figure 6):

$$i(r) = h_2/h_1 = (x_0 - r_4) / x_0. \quad (14)$$

To determine x_0, draw a line through points 1 and 2 (Figure 6), then assume their coordinates can be determined by the equation of a line on a plane:

$$(y - y_1) / (y_2 - y_1) = (x - x_1) / (x_2 - x_1). \quad (15)$$

The intersection of that line and the axis X gives the point x_0, therefore from equation (15) get:

$$x_0 = x_1 - y_1 (x_2 - x_1) / (y_2 - y_1). \quad (16)$$

Because

$$x_1 = r_1 \bullet \cos(Q_1); \quad x_2 = r_3 \bullet \cos(Q_2) + r_4,$$
$$y_1 = r_1 \bullet \sin(Q_1); \quad y_2 = r_3 \bullet \sin(Q_2), \quad (17)$$

expression (14) transforms into the following:

(18)

To determine the relationship between the angles Q_1 and Q_2 use the following condition:

$$(x_1 - x_2)^2 + (y_1 - y_2)^2 = (r_2)^2, \quad (19)$$

which after some transformations result in the following:

Finally, the desired relationship is reached:

(20)

The moment of the hydraulic cylinder is determined in the following way:

$$MC(Q, n) = Fg \bullet pg \bullet h(2) \bullet ng, \quad (21)$$

where

Fg = the area of the piston for the outboom hydraulic cylinder;
pg = the pressure on the hydraulic system;
ng = the efficiency (ng » 0.95);
h(2) = the arm of the force on the outboom hydraulic cylinder.

The arm, h(2), in an immovable coordinates system with its center at point O (Figure 6) determined by the formula:

(22)

where

2 = the angle dependent on the hydraulic cylinder stroke.

(23)

where

l = the hydraulic cylinder length when the piston rod is completely withdrawn;
s = the maximum stroke for the outboom hydraulic cylinder;
k = a coefficient, showing the degree that the piston rod is extended (0<k<1).

There is a functional relationship between angles Q_1 and 2 :

Q_1 = B/2 - 2 - arccos(r_5 / R). (24)

It is necessary to set the following functional restrictions on the multitude r :

$r_1 \bullet \sin(Q_1) > r_6$,
$r_1 + r_3 > r_4$.

Finally, the optimum solution for the operating parameters is written in the following form:

min (max[{i(r) • MC(Q, n)/LM(Q, n)}>1])
(s,d,r) $Q_0 < Q < Q_k$, (25)
$n_0 < n < n_k$,

where

r = { r_1, r_2, r_3, r_4, r_5, r_6} is the multitude of the joining sizes accounting for the restriction:
$r_1 \bullet \sin(Q_1) > r_6$.

Once again, the number of independent equations is fewer than the number of design variables. The solution can therefore be found using an optimization method, for example, the random search method.

The optimization algorithm is:

1. Set the pressure within the hydraulic system, pg, and the standard range of hydraulic cylinder sizes (diameter d and stroke s).
2. Set the construction limits (r) systematically or using a random number generator, with the following restrictions. The hydraulic cylinder should be mounted a safe distance from the boom, outboom, and the lever mechanism elements. For first attempt at solutions it is possible to use an example from existing cranes.
3. Angle Q_2 corresponds to the turn of section r_3 when the rod of the hydraulic cylinder is completely extended. The calculation is made using equations (23), (24), (20), (22), and (18) for each hydraulic cylinder within the standard range.
4. Return to step 2 if the restriction $Q_2 > n_k - n_0$ is not met.
5. Change angle n in the range from n_0 to n_k (for example, by 5^0 increments). Draw the graphs of the DM and the LM dependence on the outboom turn angle (Figure 8) using formulas (9) through (13). Calculate the value of the maximum ratio of DM over LM.
6. Return to step 2 if the transmission function, i(r), is less than 1 when n_1 = B (the boom and the outboom are in one line) or $n = n_k < B$.

Figure 8. Graphs of the driving moment, DM, and loading moment, LM, dependence on the outboom turn angle.

7. Choose the hydraulic cylinder from the minimum value of the maximum ratio of DM over LM (see step 5). Fix this hydraulic cylinder's parameters and its mounting coordinates.

8. Return to step 3 until the assigned number of repetitions have been exceeded or the designer is satisfied.

9. If necessary, set new values for pg, d, and s then return to step 2.

This algorithm can be easily utilized with a personal computer using any code language or mathematical software, for example, MathCAD.

CONCLUSIONS

The method and algorithms proposed above support forest engineers in providing them with the knowledge to design new optimal crane constructions. It allows to work out optimum drive and transmission mechanisms for forest machinery cranes.

This method also allows designers to synthesize the hydraulic boom operating and outboom operating mechanisms for traditional types of forest cranes located on machines. The algorithms can be easily utilized with a personal computers.

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