THE INFLUENCE OF THE HYDRAULIC FLUID COMPRESSIBILITY ON THE KINEMATICS OF MECHANISMS ACTUATED WITH HYDRAULIC CYLINDERS

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ABSTRACT

In the description of the kinematics of mechanisms driven by hydraulic cylinders, hydraulic agent is usually considered incompressible. This computing hypothesis introduces errors which, in certain situations, can not be accepted. The volume of hydraulic agent sent by hydraulic pump it changes due to its compressibility. As a result, differences appear on the positioning of mobile elements. This paper presents a method of evaluation by calculation of this kind of influence on the positioning accuracy of the moving parts of the mechanism analyzed.

KEYWORDS: Hydraulic cylinders, compressibility, bulk modulus

1. INTRODUCTION

In many practical applications is no longer possible to consider that the hydraulic fluids used are incompressible medium.

The compressibility is the property of the hydraulic fluid to modify his volume when the pressure varies. The phenomenon occurs in the case of liquids and it is influenced by the nature of the fluid and by the speed of compression / expansion. In general, it is accepted that the process is isothermal and therefore it operates with isothermal compressibility coefficient β. In some cases, it is preferable to use the inverse of the compressibility coefficient

$$\varepsilon = \frac{1}{\beta}. \tag{1}$$

This is called the modulus of elasticity or the bulk modulus of the liquid.

The relationship between the variations of the fluid volume, $V_0 - V$ with the pressure variation, $p - p_0$ is

$$V = V_0 \cdot e^{-\beta \cdot (p - p_0)}. \tag{2}$$

Expanding the exponential expression from 2 in Taylor series and retaining only the first two terms of approximation, the following formula is obtained

$$e^{-\beta \cdot (p-p_0)} = 1 - \frac{\beta(p-p_0)}{1!} + \frac{\beta^2 (p-p_0)^2}{2!} + \dots \cong 1 - \beta(p - p_0) \tag{3}$$

hence

$$\Delta p = \frac{1}{\beta} \cdot \frac{\Delta V}{V_0} \tag{4}$$

with V_0 initial total volume.

2. THE ASSUMPTIONS AND THE MATHEMATICAL MODEL

In the analysis of the compressibility influence on the hydraulic fluid on the kinematics of the mechanism it is considered the case of a hydraulic cylinder hinged at both ends, acting a rigid with fixed axis.

A case in point is the crane arm, positioned by means of a differential hydraulic cylinder.

The Figure 1 presents an example. It highlights some of the dimensions considered for achieving the mathematical model.

Weight G acts on the center of mass of the moving parts. At the end of the crane arm acts the technological force, F_t.

The cylinder and the hydraulic elements are considered rigid. In this way, the entire volume variation will be of the hydraulic fluid. The hydraulic fluid is considered free of dissolved gases, so the compressibility modulus is constant and precisely defined.

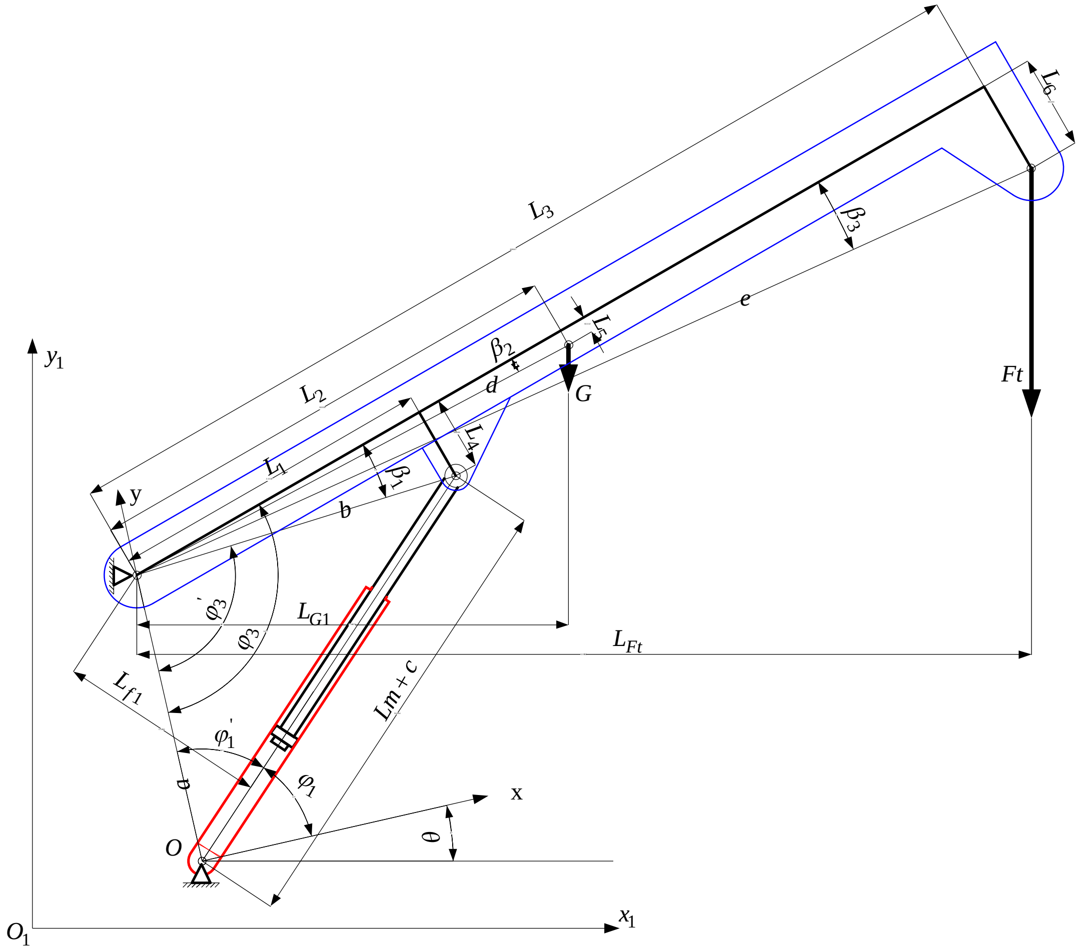

Fig. 1 Kinematic scheme of the hydraulic driven mechanism

The hydraulic installation, except the hydraulic cylinder, has a volume V_{0e}.

The initial volume of the hydraulic cylinder depends on the current stroke c.

The equilibrium of the crane arm is evaluated under static conditions.

The axial force produced by the hydraulic cylinder is due to the compression of the hydraulic fluid.

The distance from the fixed joint of the arm of the crane to the operating forces considered is

$$
\begin{aligned}
&b_{F1} = a \cdot \sin\left(\varphi_1^{'}\right) \\
&b_G = e \cdot \cos\left(\varphi_3 + \theta - \beta_2 - \pi/2\right). \qquad (5) \\
&b_{Ft} = e \cdot \cos\left(\varphi_3 + \theta - \beta_3 - \pi/2\right)
\end{aligned}
$$

The frictional forces from the hydraulic cylinder are constant and are estimated based on the starting pressure p_d. The torque produced by the force of the hydraulic cylinder on the crane arm articulation is

$$M_a = \left(\Delta p \cdot S_{cil} - p_d \cdot S_{cil}\right) b_{F1} \qquad (6)$$

The torque produced by the technological force and the weight of the moving parts, compared to the same joint, is

$$M_r = b_G \cdot G + F_t \cdot b_{Ft} \qquad (7)$$

Δp pressure occurs due to compression of the total volume V_0 due to the piston displacement with dx distance, as shown in figure 2. The relationship between the displacement, Δx, and the pressure, Δp, is

$$\Delta x = \frac{\beta \cdot \Delta p \cdot V_0}{S_{cil}} \qquad (8)$$

If the hydraulic agent is considered incompressible, the crane boom would be located at the point A_0. In reality, due to the hydraulic agent compressibility, the arm is rotated with $\Delta\varepsilon$, reaching in A point. We want to determine the position of equilibrium.

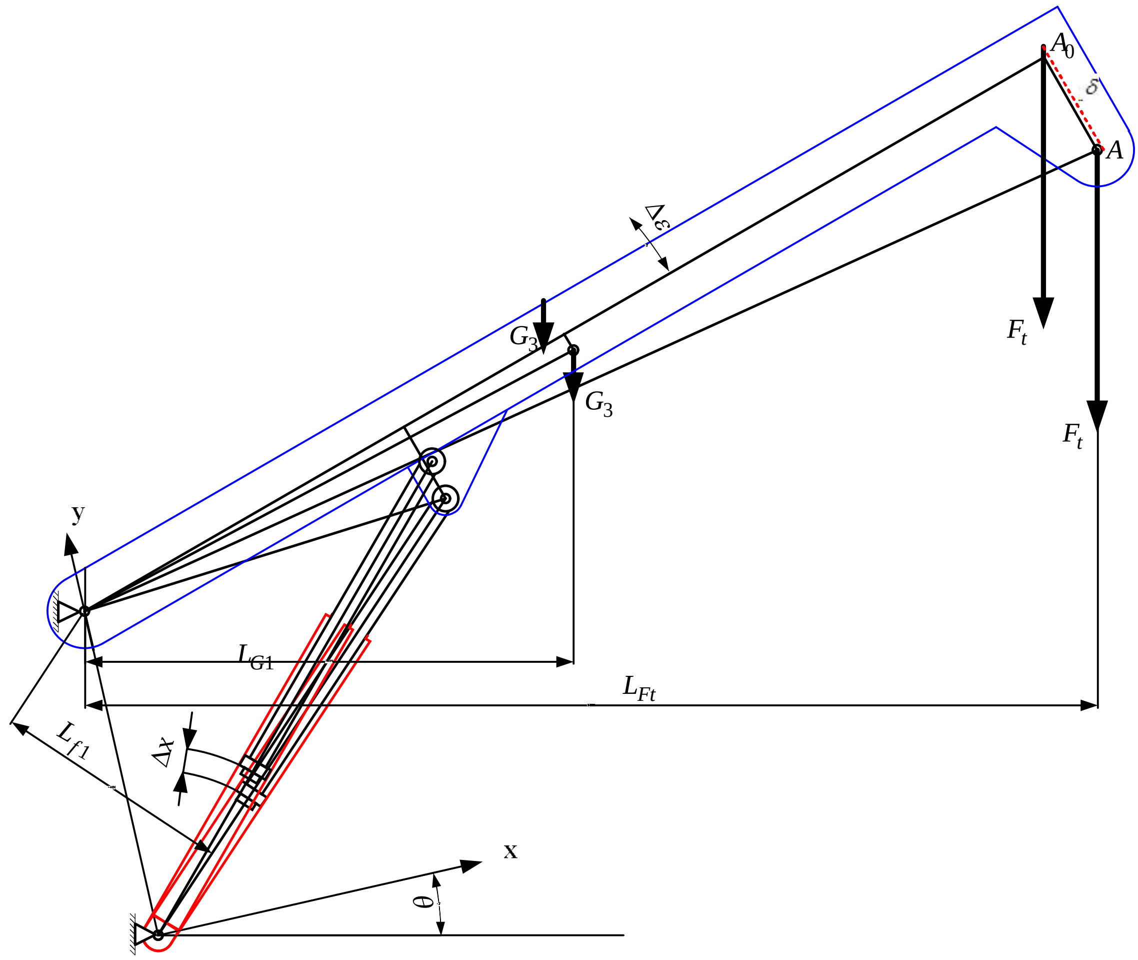

Fig.2 Graphical representation of functional parameters

The equilibrium is achieved

$$M_a = M_r . \quad (9)$$

The equation is a transcendental one and must be resolved while the Δx and Δp are not known.

3. NUMERICAL SIMULATION

To solve equation 9 a numerical method is used. Figure 3 is a schematic diagram of the bisection method. The first step is to identify the interval in which M_a and M_r functions intersect. Then the interval is narrow until the difference between the two functions is less than acceptable error. The value of Δx_m, at which the calculated error is smaller than the accepted error, is the root of equation 9.

To make a virtual simulation, a computing numerical program was done.

The main parameters used are:

-the crane arm length - L_5 = 6 m;

-the minimum length of the hydraulic cylinder assembly - L_m = 2.065 m;

-the technological force - F_t = 15000 N;

-the hydraulic fluid compressibility - β = 1.6 e9 Pa;

-the moving parts mass - m = 3000 kg;

-the volume of hydraulic fluid in hydraulic installation V_{0e} = 0.02 m^3.

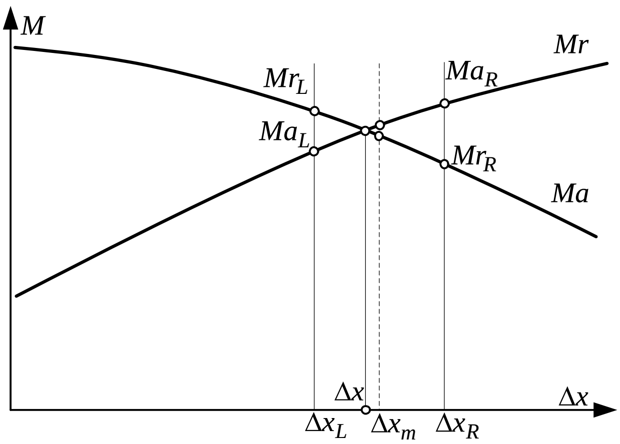

Fig. 3 Scheme calculation for bisection method

Tabular results of a numerical simulation are given in Table 1.

Table 1

c	φ_3	Δx	$\Delta\varepsilon$	δ	Δp
mm	deg	mm	deg	mm	MPa
84	79.7	12.35	0.445	46.8	25.2
168	82.8	12.59	0.462	48.6	25.6
252	85.9	12.81	0.479	50.4	26.1
336	89.0	13.01	0.496	52.2	26.5
420	92.3	13.19	0.515	54.2	26.9
504	95.6	13.56	0.535	56.2	27.2
588	99.0	13.49	0.555	58.4	27.5
672	102.5	13.62	0.577	60.7	27.7
756	106.1	13.71	0.599	63.1	27.9
840	109.8	13.77	0.624	65.7	28.0
924	113.7	13.79	0.651	68.4	28.0
1008	117.7	13.75	0.678	71.3	28
1092	122.0	13.66	0.708	74.5	27.8
1176	126.4	13.49	0.741	77.0	27.5
1260	131.0	13.21	0.776	81.7	26.9
1344	136.4	12.78	0.814	85.6	26.0
1428	142.0	12.12	0.852	89.6	24.7
1512	148.3	11.08	0.884	92.9	22.6
1596	155.7	9.29	0.884	93.0	18.9
1680	165.1	5.46	0.692	72.8	11.1

In Figure 4, the main functions analyzed are plotted.

It is found that the differences of position are significantly influenced by mechanism kinematic and its positioning from the vertical to ground.

Near to the maximum stroke of the hydraulic cylinder, the compression Δx decreases due to pressure drop. However, δ behaves differently due to the mechanism kinematic.

Differences are significant and affect the machine positioning precision.

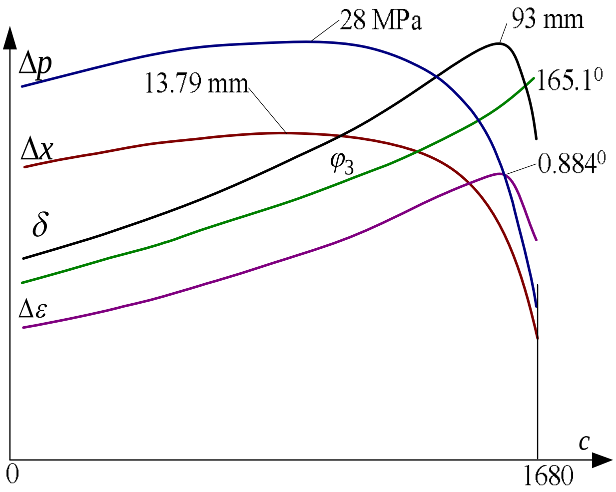

Fig 4. Graphical representation of analyzed functions

The calculation method presented is advantageous because it allows accurate calculations within a reasonable lapse of time.

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