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SOME ASPECTS REGARDING THE WORKING PERIODS OF MECHANICAL SYSTEMS DRIVEN WITH HYDRAULIC FEEDBACK

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ABSTRACT

The mechanisms of mechanical systems, driven by hydraulic feedback systems, have two degrees of mobility. Between the power system and the feedback system there is a hydraulic connection whose behavior is affected by a number of factors. Because of this connection the real mobility of mechanism is reduced by one degree. However, appear functional aspects affecting addiction. Therefore, obtaining a model of behavior of the system in time is difficult. This paper presents a series of aspects resulting from the design and testing of such systems.

KEYWORDS: hydraulic cylinders, mechanism

1. INTRODUCTION

In various applications of mechanical engineering, it is required obtaining the high torque in limited space. An example of this is the machinery that provides the rotation of ship rudder.

The mechanisms used are various. Usually, it is preferred the hydraulics providing high power density and safety in operation.

In this case the elements of the force are various:

- Linear hydraulic motors:
  - Articulated at both ends;
  - Fixed and equipped with cogwheel.
- Oscillating rotary hydraulic motors for high torque.

Figure 1 presents in 3D a system with four hydraulic cylinders hinged to ends, acting on the rudder shaft.

The hydraulic oil supply is achieved by variable flow hydraulic pump. It works in closed loop. The losses and the changes in volume are compensated by an additional hydraulic circuit. Also, from the hydraulic system of compensation is powered the hydraulic tilting system of main pumps. A simplified hydraulic scheme is presented in Figure 2.

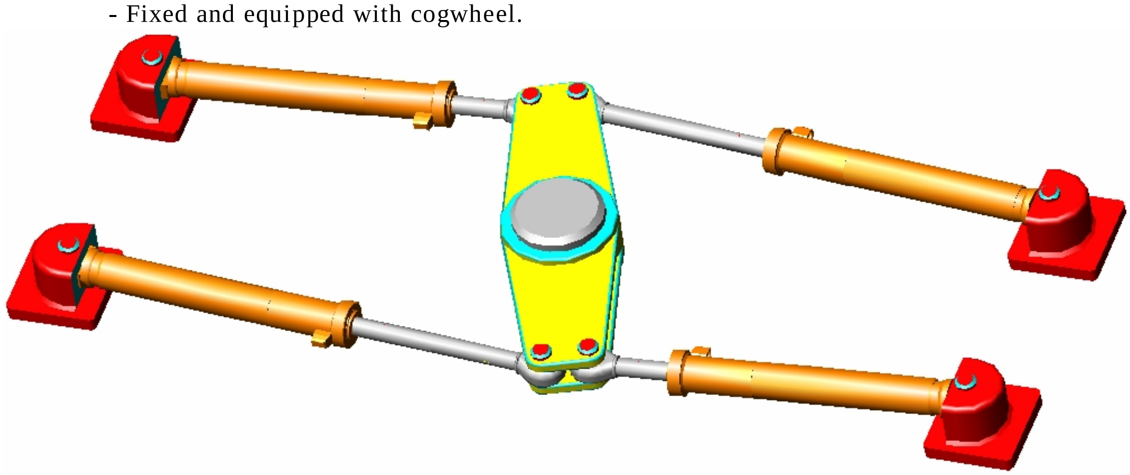

Fig. 1. The main mechanism of the rudder machine

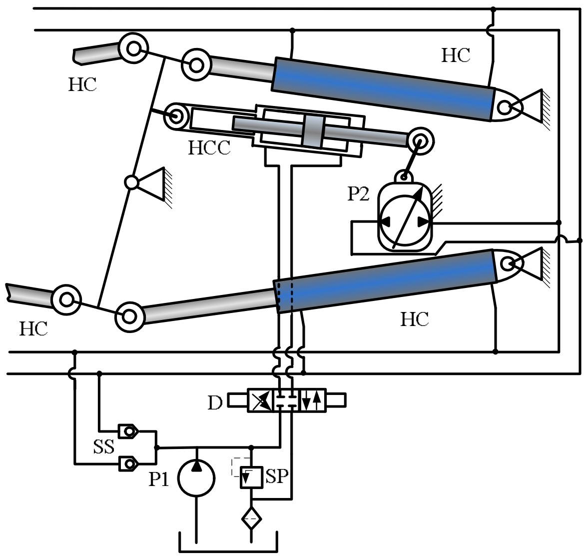

Fig. 2. Simplified hydraulic circuit.

When the distributor D, sends a supply flow rate from pump P1 to the control cylinder, HCC, its length it is modified. As a result, the lever of the hydraulic pump, P2, start to rotate. This produces a flow rate that gets into the main hydraulic cylinders.

At the beginning the pivoting speed is high. After the tipping has reached the maximum, the pump P1 ensures only the maintenance of tipping for hydraulic pump P2. When the distributor D closes the hydraulic circuit or when HCC has reached the end of the race it starts the reverse rotation of command lever of the hydraulic pump P2. Product flow decreases until the movement stops.

2.THE FLOW IN HYDRAULIC PUMPS

External flows, realized with the hydraulic pumps with variable flow, depend on:

- The position of lever control of the hydraulic pump;
- Internal flow losses;
- The compressibility of the working fluid.

$$Q_P = Q_{tP} - Q_{lP} \quad (1)$$

The theoretical flow rate of the hydraulic pump depends on:

-Its geometric characteristics (the maximum displacement);

-Rotation speed ;

-Relative position of the flow control lever

$$\varepsilon = \frac{\beta}{\beta_{\max}} \quad (2)$$

The external flow of hydraulic pumps can be described by the equation:

$$Q_P = \varepsilon \cdot D \cdot n \cdot \eta_V \quad (3)$$

where η_V is the volumetric efficiency.

For the loss of flow rate of the hydraulic pumps there are several theories, depending on the desired calculation accuracy.

Wilson describes the flow produced by hydraulic pumps with variable flow with relation (4)

$$Q_P = \varepsilon \cdot D \cdot n - c_S \cdot \frac{D \cdot \Delta p}{2 \cdot \pi \cdot \mu} - Q_R . \quad (4)$$

This relation considers that the spillage is produced only in laminar regime. c_S is the internal loss coefficient and μ the viscosity of hydraulic oil. Q_R is the constant component of the leakage flow.

Thoma's model neglects this component of flow losses.

Schlösser inserts the component of flow losses due to flows in turbulent regime.

$$Q_P = \varepsilon \cdot D \cdot n - c_S \cdot \frac{D\Delta p}{2\pi\mu} - c_{ST} D^{\frac{2}{3}} \sqrt{\frac{2 \cdot \Delta p}{\rho}} . \quad (5)$$

Olaf Olsson introduces a more general relation (6) that takes into account the effects of compressibility of the hydraulic fluid.

$$Q_P = \varepsilon \cdot D \cdot n - c_S \cdot \frac{D \cdot \Delta p}{2 \cdot \pi \cdot \mu} - c_{ST} \cdot D^{\frac{2}{3}} \cdot \sqrt{\frac{2 \cdot \Delta p}{\rho}} - \left(c_{C1} + c_{C2} \cdot \varepsilon\right) \cdot D \cdot n \cdot \frac{\Delta p}{\beta_e} . \quad (6)$$

The relations of flow (7) and volumetric efficiency (8) are made by K. E. Rydberg in a more convenient form.

$$Q_P = \varepsilon \cdot D \cdot n - a_0 \cdot \varepsilon \cdot \Delta p \cdot n - \left(a_1 + a_2 \varepsilon\right) D \cdot n \cdot \frac{\Delta p}{\beta_e} - a_3 \frac{D \cdot \Delta p}{2 \cdot \pi \cdot \mu} - a_4 D \cdot \Delta p^2 . \quad (7)$$

$$\eta_V = 1 - a_0 - \left(\frac{a_1}{\varepsilon} + a_2\right) \cdot \frac{\Delta p}{\beta_e} - a_3 \cdot \frac{D \cdot \Delta p}{2 \cdot \pi \cdot \mu \cdot \varepsilon \cdot n} - a_4 \cdot \frac{\Delta p^2}{\varepsilon \cdot n} . \quad (8)$$

In these relation the coefficients a_i depend on the characteristics of pumps and they are specific for each manufacturer.

Relation (4) or (5) are used where detailed information about the pump used is not available. According to these relationships, the flow losses depend only on pressure. 4 shows a proportional relationship even between flow and pressure losses, without depending on β.

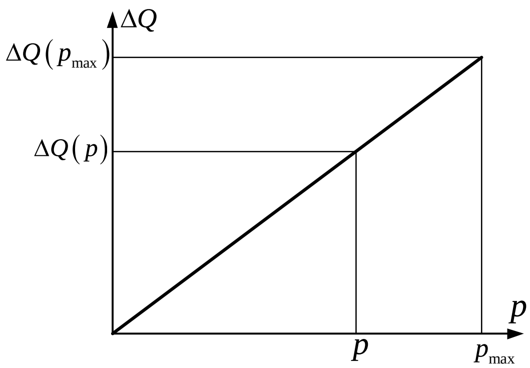

Fig. 3. Flow rate losses.

From Figure 3 we deduce that

$$\Delta Q(p) = \frac{p}{p_{\max}} \cdot \Delta Q(p_{\max}) \tag{9}$$

and how as

$$\Delta Q(p_{\max}) = (1-\eta_{HP}) \cdot Q_{tP} \tag{10}$$

it results

$$\eta_{HP}(p) = 1 - \frac{p}{p_{\max}}(1-\eta_{HP}) \tag{11}$$

It is also necessary to know the torque variation to the rudder stock.

3. PHASES AND TIMES OF OPERATION

The total rotation of the driven element is regulated by specific rules depending on the application. To analyze, in terms of kinematic behavior, a machine, it is necessary to know all phases of its operation. In this case these are:

- A - start - which consists in putting into motion the rudder stock;
- B - pump Ph generates maximum flow;
- C - stops the rotation of rudder.

In Figure 4 is shown the system geometry. Points A and B represent the joints of the main hydraulic cylinder. Points C and D represent the joints of the control cylinder.

To calculate the movement of the driven element each stage is consider separately.

In phase a, the starting, the control hydraulic cylinder is supplied by pump P1 through distributor D.

The length of the control cylinder begins to change and is changing the position of the flow control lever of the hydraulic pump P2. In this phase, the driven element and pump lever rotate in opposite directions, due to the cinematic and hydraulic schemes. Projections of points C and D on Ox axis are moving in different directions simultaneously. At one point the pump P2 lever reaches $\beta_{\max}$ and point D becomes fixed.

From this moment begins phase b. The lever of the pump P2 is tilted to the limit and the maximum flow rate is produced.

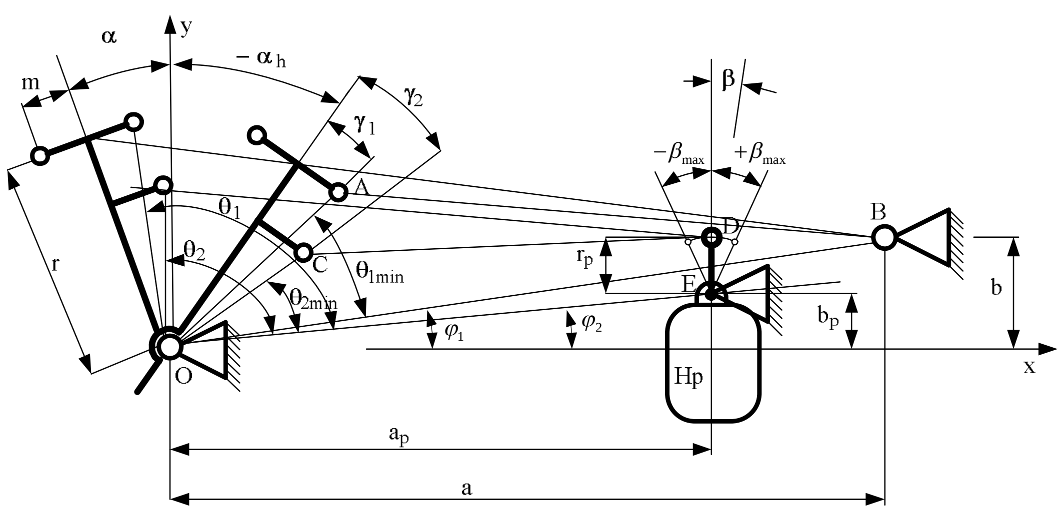

Fig. 4. Geometry of the mechanism

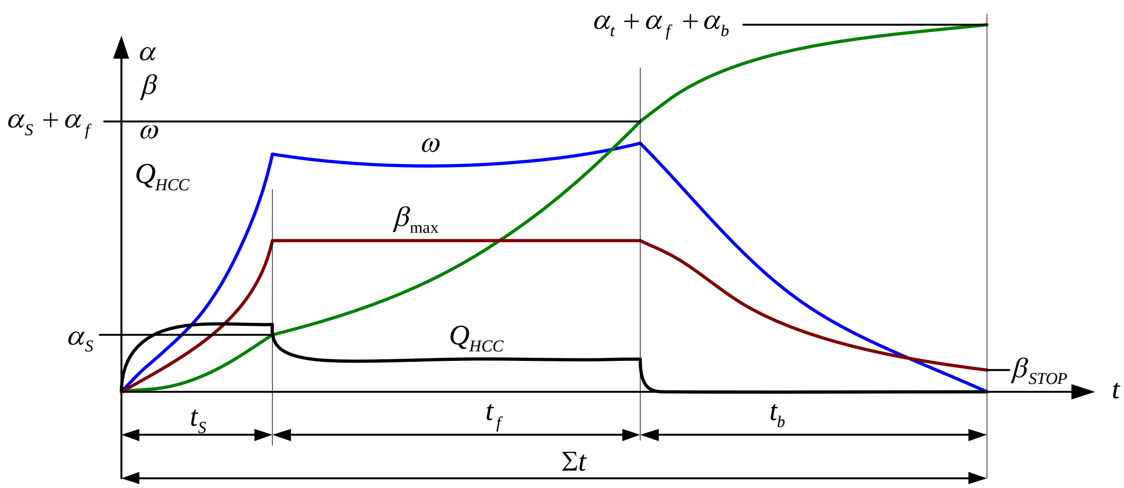

Fig. 5. The variation diagram of the main parameters.

With some advance toward the achieving of the desired angle, it stops the hydraulic oil supply for the hydraulic cylinder control (step c). Hydraulic pump continues to send hydraulic oil in the main cylinders, but with a rate flow that diminishes continuously.

The same phenomenon occurs when the hydraulic cylinder control reached maximum stroke.

If one takes into account only the theoretical flow rate of the hydraulic pump (P2), the time of realization of this last phase tends to infinity.

In fact, as previously stated, there are flow losses.

Stopping the machine is produced when all product flow rates of hydraulic pump are lost internally.

The calculation of times period for each phase of operation is based on a general model that takes into account the following:

- Mechanical equilibrium driven element;
- Continuity of flow;
- The operation of the hydraulic control system;
- How are change the forces and the moments resistant in time;
- Kinematic mechanism of force.

The resulting equations depend on the specific parameters of hydraulic pumps used.

Depending on the desired accuracy of the calculation, these can be used one of the mathematical models that describe the losses in hydraulic pumps described above.

4. CONCLUSIONS

The assessment by calculation of the behavior of the hydraulically driven mechanical systems is a complex problem.

Calculation methods depend on the complexity of the mathematical models developed.

For the type of machines analyzed, the determination of working periods (and the machine generally) requires numerical analysis method.

Figure 5 presents the results of such calculations.

The need for these calculations is dictated by the fact that the system should be automated. For this it is necessary to know when to command time-out for flow rate on the control hydraulic cylinder. Otherwise the machine will take off at different angles far from the desired ones.

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