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STABILIZATION OF THE SPEED OF THE HYDRAULIC MOTOR THROUGH THROTTLE COMPENSATION FOR VOLUME LOSSES

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Abstract: *A schematic diagram of a hydraulic drive system is provided to stabilize the speed of the working body by compensating for volumetric losses in the hydraulic motor. The diagram shows the inclusion of an originally developed self-adjusting choke whose flow rate in the inlet pressure change range tends to reverse - with increasing pressure the flow through it decreases. Dependent on the hydraulic characteristics of the hydraulic motor and the specific operating conditions.*

Keywords: hydraulic motor, through throttle, hydraulic system

1. Setting the task

It is known that the stability of the relative speed between the workpiece and the workpiece is seriously influenced by the quality of the technological process in the machine tools. For example, when the mass velocity (feed rate) of the flat grinder is uneven, the quality of the surface to be treated is greatly impaired [1,2]. This is also the case with other surface finishing machines, ferromagnetic presses, extrusion machining, etc.

When the machine is hydraulically driven, the issue of speed stabilization is about reducing the impact of volumetric losses in the hydraulic system and in particular in the hydraulic motor or hydraulic cylinder.

The criterion for evaluating the effect of the change in feeder speed on the process can be used:

$$\varepsilon = \frac{n_{max} - n_{min}}{n_{max}} \qquad (1)$$

where:

- n_{max} is the maximum feed rate during the stroke;
- n_{min} - the minimum feed rate during the run.

The difference usually occurs as a result of the change in the resistance force during the stroke. This causes the hydraulic system pressure to change.

As is known [3,4,5] (Tower), the flow of Q_1 flow to the working fluid is divided into two parts:

$Q_{working} = q.n$ - effectively used to drive the flow rate motor;

where:
q-working volume;
n-speed of the motor.

$Q_{loss=k\Delta p}$- volume losses (leaks) passing through the slots in the hydraulic motor directly to the outlet;
$- k$ is the hydraulic conductivity of the slots in the hydraulic motor. The flow in them is assumed to be laminar $[l\ /\ (s.Pa)]$;
$- \Delta$p - pressure drop in the hydraulic motor.
It depends on the load on the driven feeder or cutting tool;

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According to the above, the debit balance can be expressed by the equality [6,7]:

$$Q_1 = q.n + k.\Delta p \quad (2)$$

Assuming that during the working stroke $Q_1 = const.$, It becomes clear that as the load increases (respectively, Δp), the rotational speed (n) decreases due to the increase in volume losses.
From (1) and (2) it is easy to obtain:

$$\varepsilon = \frac{k.(\Delta p_{max} - \Delta p_{min})}{q.n} \quad (3)$$

According to (3), the degree of irregularity increases with the increase of (k), respectively the wear of the hydraulic motor, and at low operating speeds (n). When the feed speed of the feeder is low $(eg\ 3 - 4cm\ /\ min)$, as in finishing (grinding, polishing, etc.), the usable flow rate $(q.n)$ is in the order of volume losses. In this case, the degree of uneven speed is very high and there is a severe deterioration in the surface quality [8,9].

2. Schematic diagram of the hydraulic system

Figure 1 shows a part of the schematic diagram of the hydraulic system, which shows a solution to minimize the impact of volume losses (leaks) on the quality of the technological process [10].

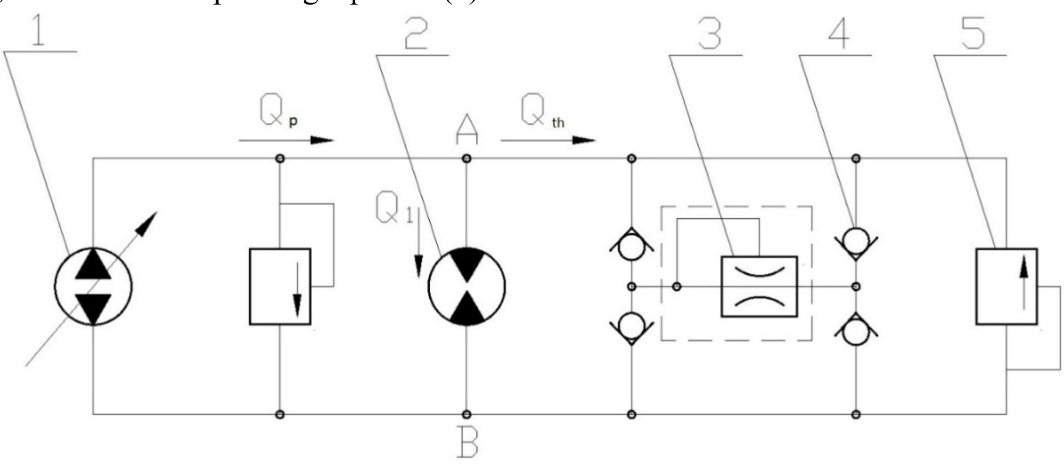

Figure 1

1.Adjustable reversible pump.
2. Reversible hydraulic motor.
3. Self-adjusting throttle.
4.Related valve.
5.Safety valve.

The drive motor (2) is powered by an adjustable volume pump (1). In parallel to this connection, a self-adjusting throttle (3), described in detail in [2], is included by means of a bridge combination of check valves.

The specific reason for the creation of this throttle is to limit the dynamic shocks in the structure of the truck when lowering the load at maximum speed in accordance with its mass. This is achieved by limiting the flow of the displacement fluid from the lift cylinder to the drive hydraulic system.

In the proposed scheme, this device performs another reliability function. According to the presented description [11,12,13], when the pressure in the operating range changes $(\Delta p_{min} \div \Delta p_{max})$, the throttle changes its hydraulic conductivity within the limits, respectively $(Q_{max} \div Q_{min})$. In particular, for $\Delta p = \Delta p_{max}\, Q_{min} = 0$. In other words, as the inlet pressure rises, the flow (Q) of the flowing through the throttle fluid decreases.
The idea is this: if in the upper branch of the pipe network, in Fig. 1, the pressure is high (p_1), the flow of the pump in it (Q_p) is equal to the sum of Q_1 the motor and Q_{th} the adjustable throttle.

$$Q_p = Q_1 + Q_{th} = const \quad (4)$$

The maximum flow through the throttle corresponds to the minimum inlet pressure, respectively, to the minimum volume losses in the hydraulic motor. The value of $Q_{th,max}$ is used for hydraulic sizing.
When the load of the motor increases, the inlet pressure increases and, as a result of the
increase in volume losses, its speed decreases. At this point, the self-adjusting throttle closes partially and reduces its flow. This leads to a redistribution of the debits in item A. (under the scheme). The Q_1 flow is now increasing and offsetting much of the increase in volume losses optimization process [14,15]. By correctly determining the coefficient k and constructing the throttle design, full compliance with the flow changes and keeping the speed within acceptable limits can be obtained.

3. Determination of the output data for hydraulic dimensioning of the throttles
The output dependence is between the maximum volume loss flow rate flow rate and the flow rate through the fully open self-tuning choke, identical with the dialectical flow of information [2,16]:

$$k.\Delta p_{max} = f_{max}.\eta.\sqrt{\Delta p_{min}} \qquad (5)$$

Here f_{max} is the area of the light section of the throttle in the fully open position;

$-\eta$-flow rate (preliminary design assumes $\eta = 0.62$);
$-p_{min}$-minimum pressure drop at the ends of the throttle. Accepts equal to that at which the motor rotates idle. This value is taken from the hydraulic motor catalog depending on the type (plate, tooth, piston, etc.);
$-$ p_{max} -maximum pressure drop during the stroke. It is determined by the parameters of the technological process and the kinematic connection of the motor with the driven mechanism at a fixed speed;
$-k$ is the hydraulic conductivity of the slots in the hydraulic motor $[1/(s.Pa)]$;
-the value may be given in the passport of the hydraulic motor. It can also be determined from the data of the debit head.
Figure 2 shows the shape of the flow dependence and the rotation speed of a tooth hydraulic motor. On the abscissa is the change in speed in relative units $\left(n = \frac{n_d}{n_{max}}\right)$, where:
- n_d are the actual rpm for the respective load. The ordinance changes the pressure at the inlet of the motor and the throttle, also in dimensionless units.
The figure shows as an example how the flow through the throttle is determined at pressure drop $(p = 0{,}8p_{max})$. The markings make it clear that the throttle flow rate is numerically equal to the increase in volume losses in the hydraulic motor!

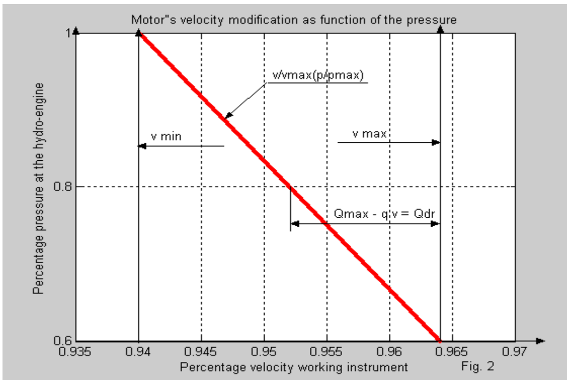

Figure 2

$$tan\,\beta = k\ или\ \frac{Q_{max} - Q_{min}}{\Delta p_{max}} = k \qquad (6)$$

In (6), the flow rate Q_{max} corresponds to Δp_{max} from the particular hydraulic system, as stated above.

4. A row for constructing the structural characteristic of the self-adjusting choke
The design characteristic of an adjustable throttle hydraulic device is the dependence of the area of the light section on the position of the regulator: $f = f(x)$, where x is the displacement of $x = x_0$, where $f = f_0$. According to the literature, such a light section should not obliterate the opening.

When the throttle is self-adjusting, it automatically responds to the change in inlet pressure, its design characteristic is a characteristic of the type:

$$f = f(\Delta p) \qquad (7)$$

i.e. the light section of the opening is a function of the inlet pressure. [3,17,18] proposes a detailed methodology for constructing the design of a throttle when it is intended for incorporation into the hydraulic system of a crank, form materials with coatings.

According to the function it performs in the proposed task, the flow through the throttle (3) of Fig. 2 can be expressed as:

$$k.\Delta p = Q_{max} - \eta . f(\Delta p).\sqrt{\Delta p} \qquad (8)$$

that is, the increase in volume losses equals the decrease in throttle flow for the same pressure change Δp.

Here, Δp takes values from Δp_{min} to Δp_{max}. Assuming $Q_{max} = k\Delta p_{max}$, from (8) for the sought function $f\ (p)$ we obtain:

$$f\big(x(p)\big) = \frac{k.(\Delta p_{max} - \Delta p)}{\eta.\sqrt{\Delta p}}. \qquad (9)$$

In [2], a conclusion was reached about the displacement of the throttle control as a function of the inlet pressure p_1:

$$x = \frac{S_{pl}}{2c}.p_1 - x \qquad \text{by [2]}$$

By expressing the pressure p_1, as an explicit function of x:

$$f_{th}(x) = \frac{k\left(\Delta p_{max} - \frac{2c}{s_{pl}}(x + x_0)\right)}{\eta\sqrt{\frac{2c}{s_{pl}}(x + x_0)}} \qquad (10)$$

The resulting expression represents the analytical type of the design characteristic of the developed throttle to compensate for the volume losses.

The last three expressions have used the quantities (10) of [2]:

$$p_1 = \frac{2c}{S_{pl}}(x + x_0) \qquad (11)$$

and replaced in (9), for the analytical type of the design characteristic of the throttle included in the drive hydraulic system according to Fig. 1, it is obtained:

$-\ s_{pl}$-area of the forehead of the self-adjusting throttle plunger;

$-\ c$-stiffness (spring constant) of the compression spring in the throttles;

$-\ x_0 -$ pre-shrink spring. It limits the minimum pressure p_min and corresponds to a fully open throttle;

$-\ x$-current contraction of the spring. The maximum shrinkage corresponds to a pressure of p_max.

The last conclusion (11) shows the unambiguous dependence of the design dimensions and parameters of the throttle on the volumetric losses in the hydraulic motor, whose speed must stabilize. Insofar as the coefficient k is variable, the throttle structure provides an opportunity to adjust the starting position of the plunger.

5. Conclusions

1. A diagram of a hydraulic system is provided which compensates for the variation of volumetric losses in the hydraulic motor in order to stabilize the processing speed during the working stroke in a machine tool;
2. The possibility of using an originally developed hydraulic element intended for incorporation into the drive hydraulic system of a car lift truck and performing other functions is presented.

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