Braking Energy Recovery Hydraulic System for Heavy-Duty Machine Tools. Mathematical Modeling and Simulation

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Abstract: *This paper examines the opportunity to recover the kinetic energy accumulated by the main kinematic chains of the heavy-duty machine-tools in downtime phase. The proposed unit can shorten the stopping time of the main spindle, with or without a blank, and would allow the recovery and storage of the accumulated kinetic energy in order to use it in a new start. The paper presents a basic diagram, and also the results of the first simulations.*

Keywords: *Heavy-duty machine-tools, kinetic energy recovery, increase of efficiency*

1. Introduction

In the case of some heavy-duty machine-tools, the main spindle, with or without a blank, is a large inertial load during the acceleration and braking phases [1, 2]. Large masses, distributed on radii that can reach values of 4, 6 or 8 - 10 m, are accelerated and braked within the specific main kinematic chains. The mechanical energy accumulated by these ones can be transformed into hydraulic energy [1].
Figure 1 shows the table of a vertical lathe with the diameter of 4300 mm. Working pieces up to 60 t can be machined on this table, with a speed range of 1 ÷ 100 rpm [2, 3].

Fig. 1. Main spindle of a heavy-duty vertical lathe

The main spindle of a heavy-duty normal lathe, able to machine parts of more than 100 t, with maximum length of 20 m, is presented in Figure 2.
In both cases, the power of the main kinematic chain drive motor [2, 3] is of the order of 100 KW. In the phases of clamping and centering the blank, repeated starts and stops are required, which entails peak charges of the electric motor caused by the high value of the acceleration or braking torque.
The braking energy recovery system presented below is especially recommended for such machines.

Fig. 2. Main spindle of a normal heavy-duty lathe

2. Hydraulic system for energy recovery and storage

In the case of normal braking, the kinetic energy, accumulated by the distributed mass during the rotational motion, is transformed into heat due to the friction from bearings, gears or even from mechanical braking systems [2]. Once transformed into heat, this energy is lost, which results in the diminution of the machine efficiency.
The energy recovery hydraulic system would allow the recovery and storage of the energy in order to be used again in the subsequent acceleration phases. The basic hydraulic diagram is shown in Figure 3.

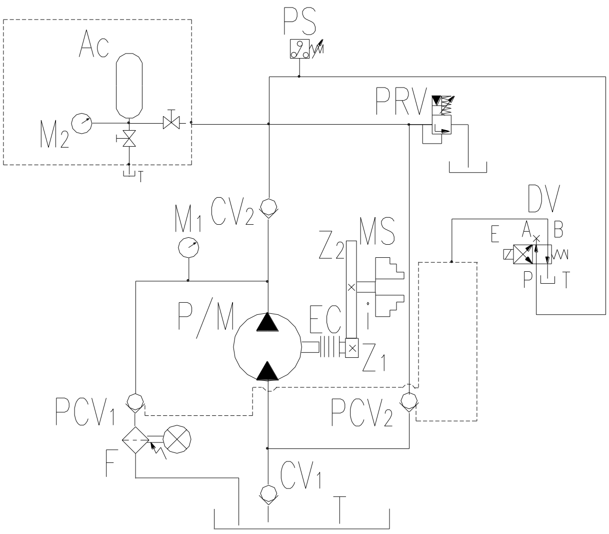

Fig. 3. Hydraulic diagram for energy recovery and storage

The following denotations are used in Figure 3: CV_1, CV_2 - check valves; PCV_1, PCV_2 - pilot operated check valves; F - filter; MS - main spindle; EC - electrical coupling; M_1, M2 - manometers; DV - directional valve; E - electromagnet; PRV - pressure relief valve; Ac - accumulator; PS - pressure switch; P/M – pump/motor reversible hydraulic machine; T - tank; Z_1, Z_2 - gear number of teeth; i - transfer ratio.
During the operation of the main kinematic chain, the electrical coupling EC is not actuated. This coupling will be actuated if the main spindle MS must be braked. The P/M reversible machine (a unit with axial pistons is recommended) will operate in PUMP mode, sucking oil through the check valve CV_1 from the tank T. The oil is sent to the accumulator (or the battery of accumulators) Ac through the check valve CV_2. The pressures are displayed on the manometers M_1 and M_2 and

electrically confirmed by the pressure switch PS. In this phase, the directional valve DV is not actuated, as the electromagnet E is not under power. When the main spindle MS is completely stopped, the coupling EC must be disengaged. The entire unit is protected against overpressure by the pressure relief valve PRV. The accumulator Ac is bladder type and is charged with nitrogen at the appropriate pressure [4, 5, 6, 7]. Depending on the machine size, batteries of accumulators can be used [3, 4, 7].
For restarting the machine, the coupling EC and the electromagnet E of the directional valve DV will be actuated. The P/M reversible machine will operate in hydraulic MOTOR. The pressure existing in the accumulator will drive, on the P - B path of the directional valve DV, the two pilot operated check valves PCV_1 and PCV_2, allowing the supply of the motor through PCV_2 and its discharge through PCV_1. In this phase, the hydraulic motor together with the motor of the main kinematic chain will start the main spindle.
Figure 3 shows a basic diagram; in reality there are also other mechanical and electronic systems for synchronization and protection. After reaching the intended rotational speed, the EC couplingis deactivated.

3. Calculation of the braking system

For the beginning, the following elements are considered as known: E_C – kinetic energy accumulated by the mobile assembly that is to be recovered; n_T – number of revolutions accepted at the level of the spindle of the P/M reversible hydraulic machine. The following notations will be also used: V_0 - total volume of the accumulator; p_0 – nitrogen pre-charge pressure [4, 8, 9]; γ - nitrogen adiabatic coefficient; ΔV – volume of oil circulated between the P/M hydraulic machine and the accumulator in the braking and acceleration phases; p_1 - nitrogen maximum pressure in accumulator; p_2 - nitrogen minimum pressure in accumulator; p_{PRV} – pressure set at the pressure valve. It is assumed that between all these pressures there is a relation as follows:

$$p_0 < p_2 < p_1 < p_{PRV} \tag{1}$$

If the mechanical losses by friction are neglected, it can be considered that the energy transfer is made according to the relation:

$$E_C = \frac{p_1+p_2}{2} \times \frac{p_0 \times V_0}{p_1} \times \left[\left(\frac{p_1}{p_2}\right)^{\frac{1}{\gamma}} - 1\right] \tag{2}$$

In relation (2) it was considered that the discharge of the accumulator is made adiabatically [4, 5, 8] and that the pressure throughout this entire phase is the average of the minimum and maximum pressures.
The volume circulated, in the absence of flow losses, will have the value:

$$\Delta V = q_{P/M} \times n_1 \tag{3}$$

$$\Delta V = \frac{p_0 \times V_0}{p_1} \times \left[\left(\frac{p_1}{p_2}\right)^{\frac{1}{\gamma}} - 1\right] \tag{4}$$

In the above relation, the number of revolutions performed till the stop of the main spindle is noted by n_1. The main spindle in turn will perform a number of revolutions n_2, according to the relations:

$$n_2 = i \times n_1 \tag{5}$$

$$i = \frac{Z_1}{Z_2} = \frac{n_2}{n_1} < 1 \tag{6}$$

The analysis of the relation (2) highlights a limitation of the energy that can be stored, according to the relation:

$$E_C < p_0 \times V_0 \tag{7}$$

At the present moment, the most commonly used accumulators operate at pressures of 320 bar at the most. According to relation (2), it is observed that under these conditions, depending on the field of application, one or several accumulators will be chosen. It should be noted once again that the very large machines may require a big number of accumulators, which can lead to excessive price increases.
For example, consider the following situation: p_0 = 180 bar, p_1 = 280 bar, p_2 = 200 bar, p_{PRV} = 300 bar. It is assumed that two accumulators with the total volume of V_0 = 50 l will be selected. In these conditions, the maximum accumulated energy, in conformity with the relation (2), is E_C = 100285 J.
If one considers that a machine with axial pistons is used, having $q_{P/M}$ = 150 cm^3, according to the relations (3) it will make a number of ~60 revolutions until it stops. If one considers that i = 1/20, the main spindle will perform ~20 revolutions until stopping.
If it is intended to use smaller hydro-pneumatic accumulators, they could be associated with accumulators exclusively used for gas.
Given these conditions, the volume V_0, distributed as in Figure 4, could be considered:

$$V_0 = V_{O/G} + V_G \tag{8}$$

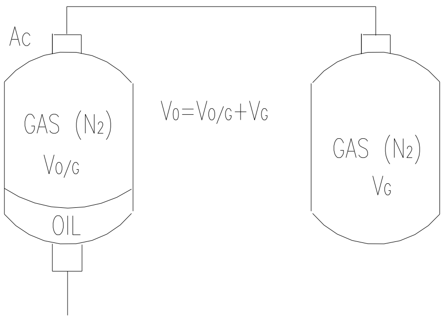

Fig. 4. Gas-oil distribution when using supplementary accumulators

The minimum recommended volume for the hydro-pneumatic accumulator (gas and oil) is:

$$V_{O/GMin} = \frac{V_0}{0.9} \times \left(1 - \frac{0.9 \times p_2}{p_1}\right) \tag{9}$$

The system will be loaded with gas taking into account the following recommendations:
If:

$$\frac{V_0}{3} \ll V_{O/G} \ll \frac{V_0}{2} \tag{10}$$

It is recommended to make the loading at the value:

$$p_0' = 0.95 \times p_0 \tag{11}$$

If the value of $V_{O/G}$ verifies the condition:

$$\frac{V_0}{4} \ll V_{O/G} \ll \frac{V_0}{3} \tag{11}$$

It is considered that the pre-load pressure must be:

$$p_0' = 0.97 \times p_0 \tag{12}$$

Subsequent research will examine the extent to which the use of additional accumulators, only with gas, can have positive influences.

4. Simulation of the operation of the hydraulic system for braking energy recovery [9, 10]

In addition to the above data, it will be considered that the value of the starting resistance moment is T = 500 Nm, at the level of the rotary spindle.
Following the simulation, the dynamic characteristics for the system pressure and the rotational speed of the hydraulic machine with axial pistons were obtained.
In the accumulator charging stage, which corresponds to the main spindle braking, the pressure in accumulator evolves as in Figure 5.

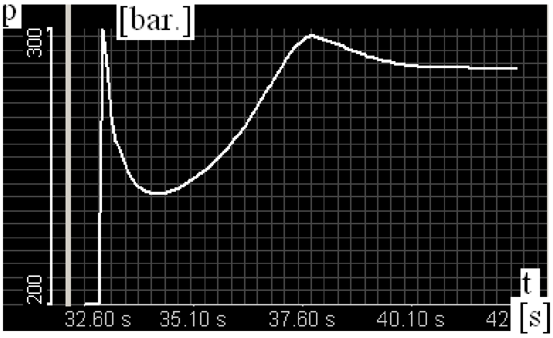

Fig. 5. Evolution of the pressure in accumulator when braking the main spindle

In a first phase, when the P/M motor is coupled, the pressure decreases due to its filling; afterwards the pressure increases up to the maximum value of 300 bar, value when the pressure valve opens and the pressure stabilizes at the value p_1.
When the main spindle is restarted, the accumulator will discharge and the pressure inside will evolve as shown in Figure 6.

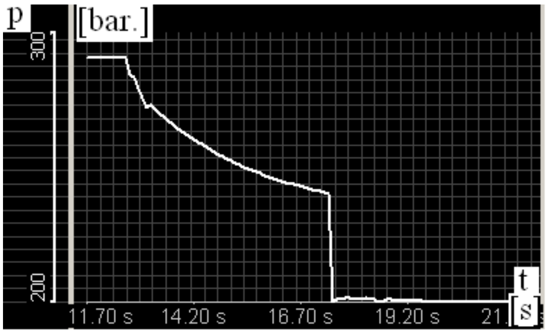

Fig. 6. Evolution of the pressure in accumulator when starting the main spindle

The pressure decreases from value p_1 = 280 bar to value p_2 = 200 bar. The system will be charged at the next braking. In this phase, depending also on the torque value, the speed of the rotary hydraulic machine, working as a motor, evolves as in Figure 7.

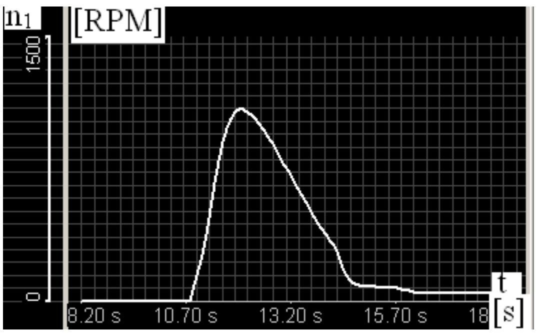

Fig. 7. Evolution of the rotary hydraulic motor in the braking phase of the main spindle

The P/M rotary hydraulic motor takes over part of the drive motor load of the main kinematic chain. The electromagnetic clutch will be deactivated after starting.

5. Conclusions

A thorough analysis of both technical and financial capacities shows that the recovery of the kinetic energy of the main kinematic chains by hydraulic way can represent an acceptable solution. Thus, the acceptable working pressures for the usual equipment do not exceed 320 bar, and the prices of the accumulators increase considerably when the accumulator volume increases.
As for the heavy or very heavy machines, the accumulators volume can exceed values of the order of hundreds of liters. This fact can also lead to considerable increases of the unit overall size, in addition to a very high price.
In further research, the aim is to develop calculation programs and methods which will allow to take also into consideration other specific parameters such as the frictions in gears and bearings and their thermal effects. It will be also studied the opportunity to use accumulators for gas only, besides the oil/gas type accumulators.

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