Remanufacturing of Hydrostatic Systems for Heavy Duty Vertical Lathes

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Abstract: *In this paper the author presents a part of the theoretical and experimental research conducted during the remanufacturing of a heavy duty vertical lathe. The remanufacturing of this machine-tool involved the re-design of the hydrostatic system for table suspension. These very heavy machine-tools use hydrostatic systems for supporting the main spindle on bearings, which allows them to take over thrust loads over 100 to at speeds of 50 RPM. In these conditions, the hydrostatic system must be well dimensioned, stable and controllable both in conventional operation mode and especially in CNC operating mode in order to ensure the working parameters.*

Keywords: *heavy vertical lathe, hydrostatic supporting on bearings, CNC machine-tools*

1. Introductory notions

The systems where the load operates on an oil film made and maintained by means of a pressure hydraulic source that necessarily includes a pump [1, 2, 3]. are defined as hydrostatic lubrication systems. If there is also a movement between surfaces, the lubrication system is mixed (hydrostatic and hydrodynamic). In practice there is no clear differentiation of these systems: the hydrostatic lubrication refers to the systems where the lubricating film is obtained from a pump.
The specific elements of the hydrostatic lubrication systems are:

- High stiffness, which allows a steady and accurate position even if there are big variations of the load;
- The wear of elements is non-existent thanks to the permanent film of oil, even during the stop phases;
- The loads are discharged evenly because in a lubrication pocket the pressure is the same anywhere;
- The quality of surfaces is less important than in the case of hydrodynamic lubrication because the load is taken over on a continuous surface rather than in points;
- A good cooling of the moving surfaces is achieved due to the continuous flow.

Among the disadvantages of these systems we can mention:

- The units are expensive, requiring specific devices such as: pumps, pressure valves, filters, throttle valves, flow control valves etc. [4, 5, 6, 7];
- Because of the relatively large number of elements, the reliability of these systems can raise issues.

Solving them involves constructive complications. First, the existence of permanent pressure must be confirmed because its lack could lead to the destruction of the surfaces in contact.
As for the mathematical apparatus used below, it has to be said that it is based on flow equations that are valid only if the thickness of lubricant film is smaller than its length and width. Also, inertial phenomena and temperature variations are neglected.

2. Hydrostatic pockets for heavy duty vertical lathes [1, 8, 9]

In the case of these machine-tools, the hydrostatic pockets are located at the level of the bed and are distributed on one or several rows, in the area of taking over the efforts from the faceplate, as in Figure 1, where it was noted: 1 - zone of pockets, 2 - pockets, 3 - zone for external discharge, 4 - zone for internal discharge.
Hereinafter we take into consideration an unfolded pocket to which the radiuses are neglected as in figure 2, where the following notations were made: L and l - total sizes of pocket, l-2a and L-2b - actual sizes of the pocket of depth e.

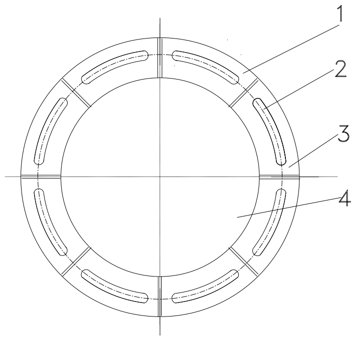


Fig. 1. Location of hydrostatic pockets on heavy duty vertical lathes

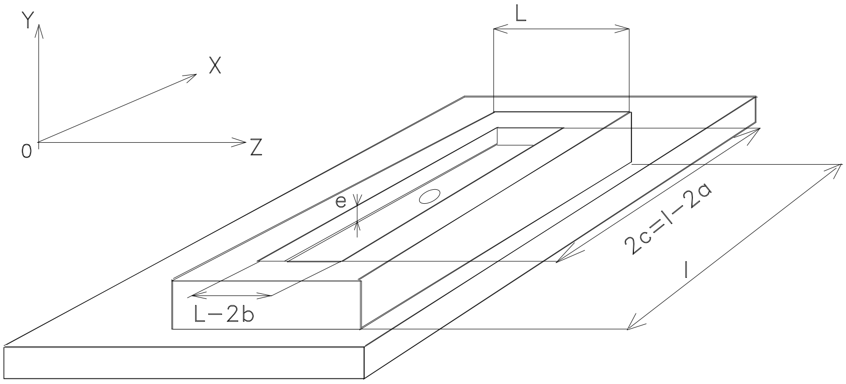


Fig. 2. Geometrical elements of a hydrostatic pocket

The table of the vertical lathe, on which the workpiece is located, is supported on an oil film that has the thickness h, as in Figure 3.

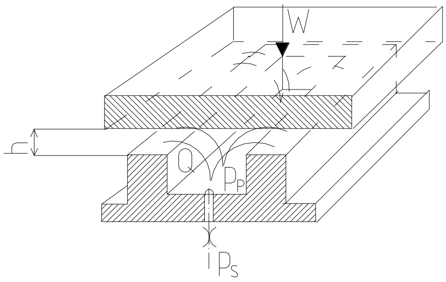


Fig. 3. Defining film thickness

The source of hydrostatic energy is a unit [1, 4, 5, 10] with the pressure p_S; the flow control valve allows the pressure p_P to be adjusted for each pocket. The flow Q, at the level of a pocket, is discharged through the gap h on the surface described on its perimeter [1]. The flow rate of the oil through the gap h depends on a number of factors: speed of table, pressure p_P, flow area characterized by the coordinates (x, y, z), according to Figure 2.
In order to establish the mathematical model it is considered that the principle of superposition can be applied.
For the calculation diagram in Figure 3 one obtains the values of the load W and the required flow Q [1]:

$$W = p_P \cdot S \cdot K_W \tag{1}$$

$$Q = \frac{p_P \cdot h^3}{\mu} \cdot K_Q \tag{2}$$

$$S = L \cdot l \tag{3}$$

In the relations above there are also noted: p_P - pressure in pocket, μ - dynamic viscosity of oil, K_W - load constant of the pocket, K_Q – flow constant of the pocket. K_W and K_Q constants depend on the geometrical sizes that define the pocket [1, 10].
In order for the system to be stable it is necessary that a tendency to return occurs after a movement out of the balance position which was caused by an external force.
This desideratum is achievable if the pressure p_P is properly chosen, depending on film thickness. In the presence of a hydraulic resistor placed between the pressure source and the pocket as in Figure 3, it can be written for the flow rate:

$$Q_f = f(p_S - p_P) \tag{4}$$

The following notations have been used in the relation above: Q_f – flow rate through resistor, considered equal to Q flow of the relation (2); $f(p_s\text{-}p_P)$ - function of definition of pressure drop flow; p_s – source pressure; p_P – pressure in pocket.
If the two expressions of the flow rate are equal, the rigidity λ can be defined as follows:

$$\lambda = -\frac{\partial W}{\partial h} = -\frac{\partial W}{\partial p_P}\frac{\partial p_P}{\partial h} = -\frac{3W}{h}\frac{1}{1-\frac{p_P}{Q}\frac{\partial Q}{\partial p_P}} \tag{5}$$

Rigidity λ depends on the type of resistor chosen. The intended values of rigidity can be obtained by a suitable choice of the resistor. There are used fixed or adjustable resistors.

3. How to adjust the hydrostatic pockets

Capillary tubes, hydraulic resistors (throttle valves) or flow control valves [1]. are commonly used for flow and pressure regulation.
For these systems, in this order, the relation (5) becomes:

$$Q_{f1} = K_1 S_1 (p_S - p_P) \tag{6}$$

$$Q_{f2} = K_2 S_2 \sqrt{p_S - p_P} \tag{7}$$

$$Q_{f3} = K_3 S_3 \tag{8}$$

In the relations (6) - (8) it has been also noted: K_1, K_2, K_3 - specific constants, S_1, S_2, S_3 - flowing surfaces of the respective element. If the subunitary ratio of the pocket pressure is noted by β and the source ratio ($\beta = p_P/p_S$), for the rigidity λ it is obtained:

$$\lambda_1 = -\frac{3S_1K_W}{h} p_S \beta (1-\beta) \tag{9}$$

$$\lambda_2 = -\frac{6S_2K_W}{h} p_S \beta \frac{1-\beta}{2-\beta} \tag{10}$$

$$\lambda_3 = -\frac{3S_3K_W}{h} p_S \beta \tag{11}$$

$$0 \le \beta \le 1 \tag{12}$$

Figure 4 shows, for the three cases, the rigidity dependence on the thickness of oil film. Usually, this film does not exceed 0.5 mm in the case of these machines.

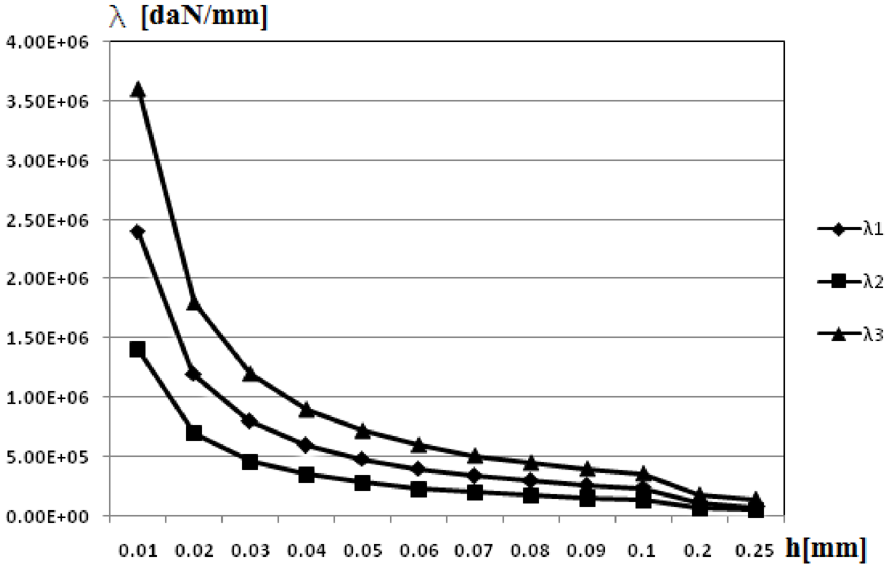


Fig. 4. Rigidity dependence on film thickness

For the lathe to be rebuilt, the intended maximum thickness was 0.15 mm. From the considerations shown above it results that the regulation systems with flow control valves are preferable [3, 4, 7]. for rigidity reasons. The use of serpentines (capillary hydraulic resistors) is a cheap and simple design solution which has however major drawbacks because of the difficulties of adjustment and the risk of clogging with impurities. In the case of clogging, the interventions for repairs last long and involve the change of serpentines and recalibration of the new ones.
The throttle valves are much cheaper than the flow control valves but, as it results from the relations above, the regulated flow rate depends on the load (pocket pressure p_P). This thing influences the accuracy of machine faceplate due to the variations of *h* film thickness, according to the relation:

$$h = \sqrt[3]{\frac{QS\mu K_W}{WK_Q}} \tag{12}$$

The variations of load *W* lead to variations of pressure; these ones change the flow rate which, in its turn, influences the thickness of the film *h*. In order to study the stability of the flow rate and, indirectly, of the film thickness, there were used specialized programs of simulation [11, 12].
Figure 5 shows the influence of pressure variations on the flow rate in the pocket when it is adjusted by a throttle valve.

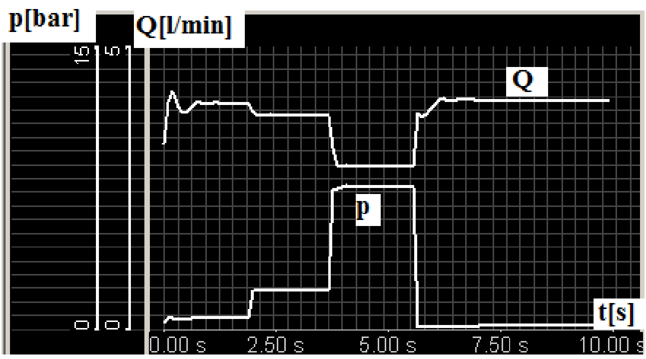


Fig. 5. Influence of pressure variations on the flow rate in the pocket when it is adjusted by a throttle valve

As shown above, the variations of pressure entail uncontrolled modifications of the flow rate and hence of the size *h*.

If a flow control valve is used instead of a throttle valve, one can notice that the pressure variations do not influence the flow rate value within certain limits, according to Figure 6. These limits depend on the adjustment of the flow control valve, on pocket geometry but also on the viscosity and temperature of the oil [1, 2].

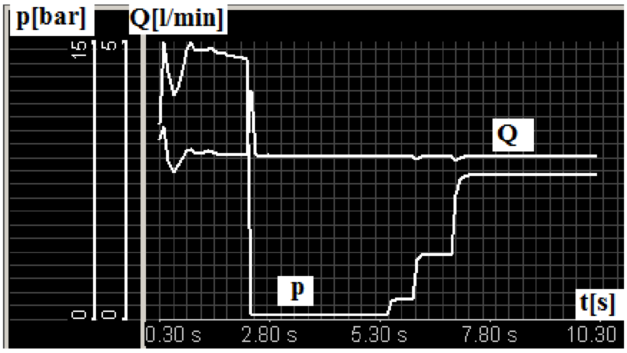


Fig. 6. Influence of pressure variations on the flow rate in the pocket when the flow is adjusted by a flow control valve

The time required for stabilization at start-up is of 3s approximately and is not a problem for a machine-tool in this range.

4. Setting the hydrostatic diagram

The hydraulic diagram in Figure 7 was developed in order to achieve the hydrostatic unit.

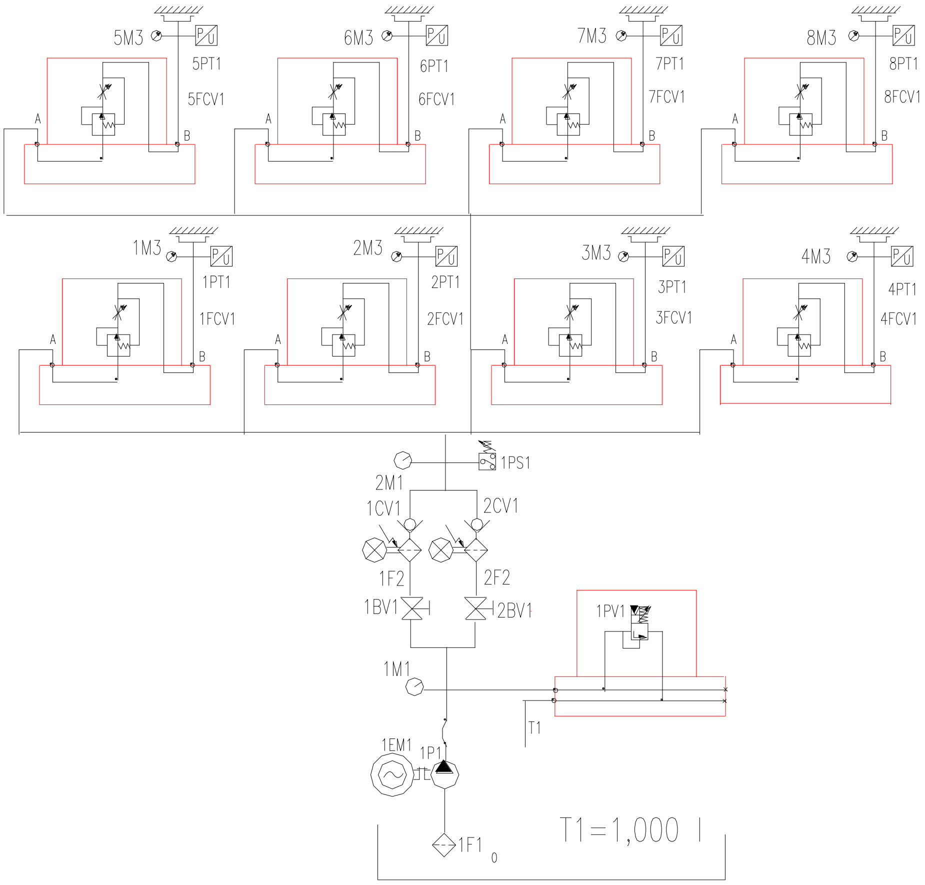
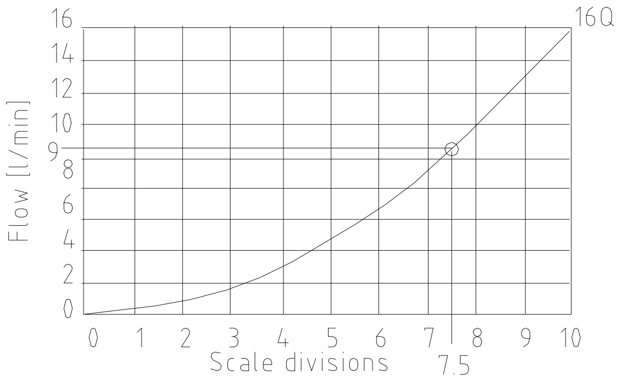


Fig. 7. Hydraulic diagram for the lathe with eight pockets

From the tank T1 the pump 1P1, driven by the electric motor 1EM1, sucks the oil through the suction filter 1F1. The oil is sent through the valves 1BV1 and 2BV1 to the pressure filters 1F2 and 2F2. These ones are provided with clogging electrical indicators. The eight flow control valves 1-8 FCV1 are supplied through the check valves1CV1 and 2CV1. For each pocket there is one manometer 1-8 M3 and one pressure transducer 1-8 PT1. The source pressure is regulated by means of the pressure valve 1PV1, and the value of the maximum pressure is displayed on the manometer 1M1. The filter that operates at a given moment is chosen with the help of the valves 1-2V1. If this filter is clogged, it will be separated by means of the valves and the stand-by filter will be coupled instead. The replacement of the clogged cartridge can be made without having to turn off the machine. The supply pressure of the pockets can be viewed on the manometer 2M1, located near the operator. The pressure required to operate is confirmed by the pressure switch 1PS1. Taking into account the characteristics and performances of the machine, it was chosen a pump with the capacity of q_p = 48 cm^3, driven by an electric motor with the rated speed of n_{EM} = 1470 RPM and a power of P_{EM} = 4 kW.
In these conditions, two-way flow control valves [6] were selected for the eight pockets, with the main features shown in Figure 8.



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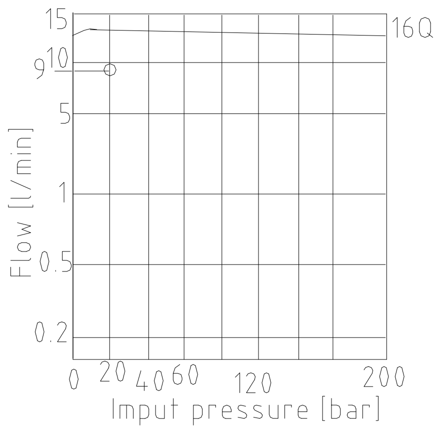


b

Fig. 8. Operational characteristics of the flow control valve used

In the theoretical case where all eight pockets are identical and the workload is evenly distributed, as per Figure 8a, the flow control valves allow a controllable adjustment from 0 to 7.5 divisions out of 10 at the most. It is estimated that the maximum working pressure shown on the manometer 2M1 will not exceed 20 bar, in which case, according to the characteristics in Figure 8b, it is possible that a maximum flow rate of 16 l/min is adjusted on each pocket. Under these conditions, in conformity with the relation (12), it is estimated that the maximum theoretical thickness of the film will be: $\mathrm{h_{Max}} = 0.6\mathrm{mm}$. In fact, a film of 0.15 - 0.2 mm evenly made is the maximum allowed.
To determine the surface *S* in the relation (3) and the constants K_W and K_Q in the relations (1) and (2) we consider the expressions [1]:

$$S = \pi(R_E^2 - R_I^2) \tag{13}$$

$$K_W = \frac{1}{2(R_E^2 - R_I^2)}\left(\frac{R_E^2 - R_2^2}{ln\frac{R_E}{R_2}} - \frac{R_1^2 - R_I^2}{ln\frac{R_1}{R_I}}\right) \tag{14}$$

$$K_Q = \frac{\pi}{6}\left(\frac{1}{ln\frac{R_E}{R_2}} + \frac{1}{ln\frac{R_1}{R_I}}\right) \tag{15}$$

In the relations above there were noted: R_E - outer radius of guideways, R_I - inner radius of guideways, R_1 - inner radius of pockets, R_2 - outer radius of pockets, as in Figure 9.

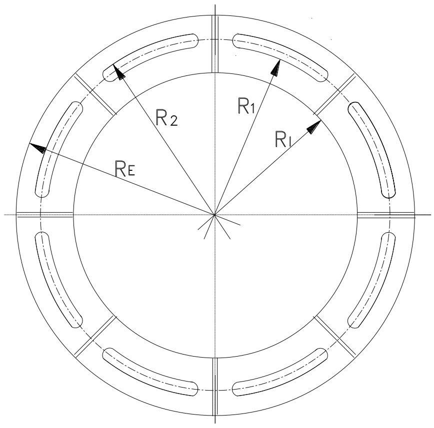


Fig. 9. Geometrical elements defining the hydrostatic pockets

5. Theoretical premises

After the measurement of pockets size of the remanufactured machine, the following values were obtained for the surface *S* and the constants K_W and K_Q: S = 75000 cm^2, K_W = 0.85 [-], K_Q = 21 [-].
The faceplate of the machine has a mass of 20 000 Kg and can bear a workpiece of 100 000 Kg. In these conditions, the required pocket pressures are determined by means of the relation (1) and the necessary flow rate for different values of the film is determined using the relation (2). The theoretical values obtained for two oils with different viscosities are listed in Table 1.

Table 1: The theoretical values obtained for two oils

	W [daN]	p [bar]	h [mm]	Oil v = 46 mm^2/s Q [l/min]	Oil v = 20 mm^2/s Q [l/min]
1	20000	0.4	0.15	4.4	10
2	20000	0.4	0.3	33	76
3	100000	2	0.15	21	48
4	120000	2.5	0.1	8	18
5	120000	2.5	0.15	26	60
6	150000	3	0.15	31	72

For the sizing of the pump 1P1 in the diagram shown in Figure 7 at these flow rates, there were also taken into account the flow rates required for lubricating the feed box, the pinion-ring gear mechanism and the central bearings.
For a certain adjustment of the flow control valves, corresponding to a constant flow rate, the following relation can be used for estimating the dependence of film thickness on the load:

$$p_P h^3 = ct. \tag{16}$$

As shown in Table 1, the necessary flow rate – depending on oil viscosity - for a maximum film of 0.15 mm, at a total load of 120 000 daN, is in the range of 26-60 l/min.
If the intention is to obtain a film of 0.15 mm for a load of 120 000 daN, in idle running without load, according to the relation (16) the film will have a thickness of 0.3 mm. Generally, during the machining operations, the film thickness should be as small as possible so as not to affect the machining accuracy because of the compressibility of the oil. In the case of this machine, it can be considered that the variation of oil thickness Δ*h* due to the variation of vertical forces (including those due to the variations of weight) has the expression below:

$$\Delta h = h \cdot \Delta F \cdot 9 \cdot 10^{-10} \text{ [mm]} \tag{17}$$

In the relation (17) the variation of force Δ*F* is expressed in daN.
According to Table 1, the capacity of the machine in terms of hydraulic unit makes also possible the takeover of bigger loads, 190 000 daN for example, if the pump flow rate is the necessary one.

In fact, at loads of more than 120 000 daN, the elastic deformations of the mechanical system affect the machining accuracy. Under these conditions, it is recommended to limit the weight of the workpiece at 100 000 daN.
The vertical travel of the faceplate is limited using a system of radial-axial bearings as in Figure 10.

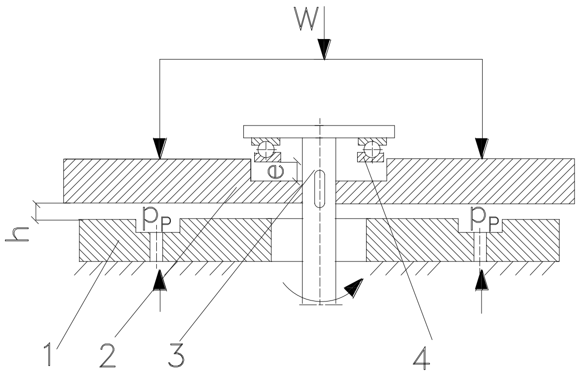
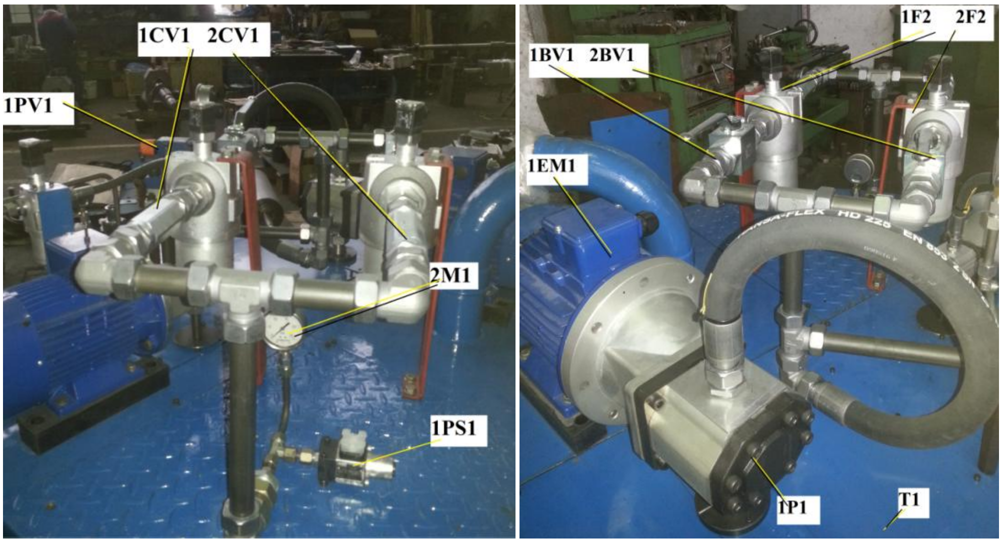


Fig. 10. The system of radial-axial bearings used for limiting the vertical travel of the faceplate

Thanks to pressure p_P in the pockets between faceplate 2 and bed 1, an oil film of thickness *h* is made in presence of the load *W*. The faceplate 2 is driven by the spindle 3. Between the bearing 4 and the faceplate 2 there is the size *e*. If the film *h* grows, at some point the gap *e* may become "0". From now on the surplus of pressure that exceeds the value required for the takeover of the load *W* leads to the emergence of an thrust force that will be discharged onto the bearing. It must not exceed the admissible value, in conformity with the catalogue [6].

6. Experimental results and achievements

The hydraulic unit for suspension was made in the form of a cylindrical tank T1 of 1000 l on which most of the devices were assembled as in Figure 11. The hydraulic devices shown in this figure have the same notations as those in the hydraulic diagram of Figure 7.



a. b.

Fig. 11. Hydraulic unit for suspension

The hydraulic devices in Figure 7, placed after the pressure switch 1PS1, were located on the machine bed as in Figure 12.

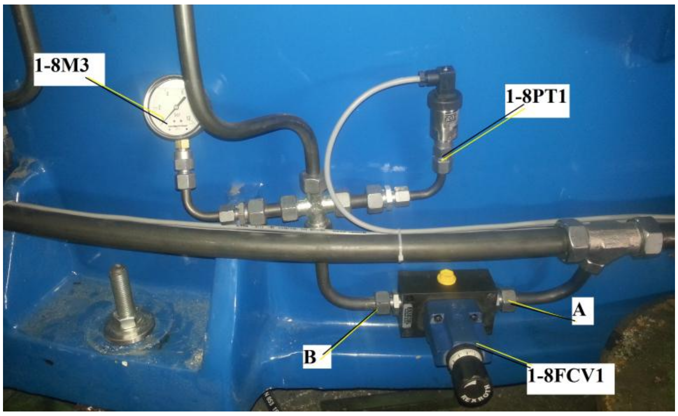


Fig. 12. Location of the control and regulation devices for the pressure in pockets

This equipment is represented by the flow control valves 1-8 FCV1, pressure switches 1-8 PT1 and manometers 1-8 M3. The oil film in the eight pockets can be regulated by means of this equipment. In Figure 13 is shown the location of the eight pockets on the bed of the machine.

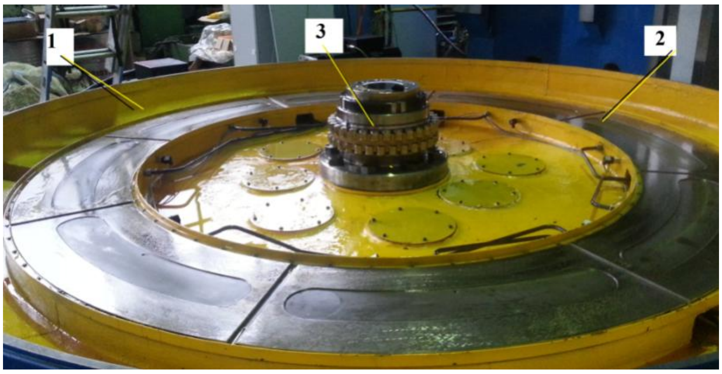


Fig. 13. Location of the real pockets on the bed of the machine

There are eight pocket-type plates 2 clamped to the bed of the machine 1 (made of cast iron); these plates ensure the thrust load discharge. The supporting on bearings 3 makes possible the radial centering of the table.
Initially, a primary adjustment of the flow rate was made in the eight pockets as in Figure 14.

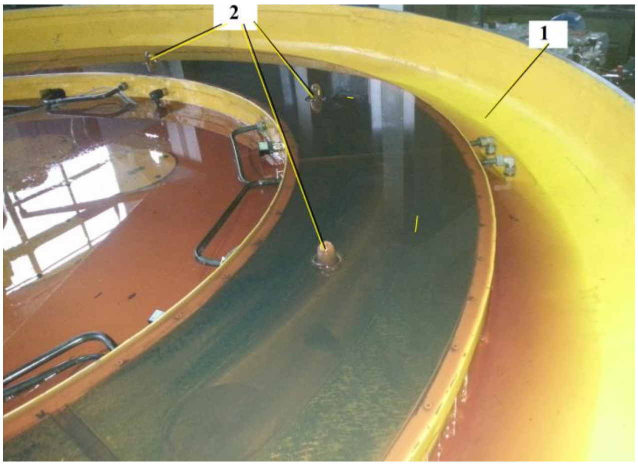


Fig. 14. Primary adjustment of the oil flow rate in the pockets

The unit was started in the absence of the faceplate; then it was checked if columns of liquid 2, approximately equal in height, are made in the eight points controlled. The resulted oil is recovered using the trough 1.

After making equal columns, the hydraulic unit was stopped and the faceplate of the machine was assembled as in figure 15.

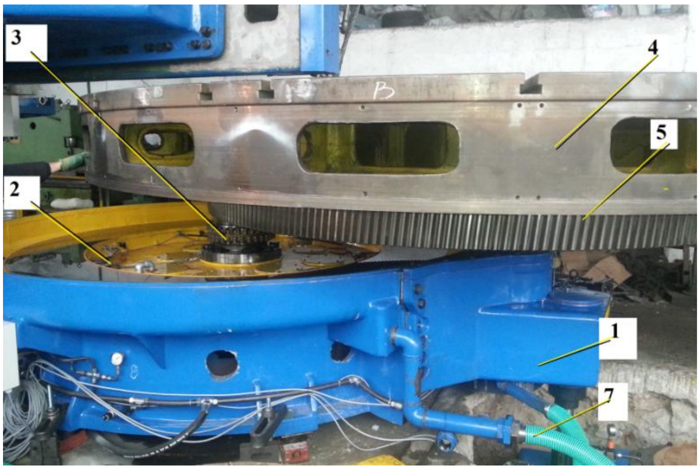


Fig. 15. Assembling of the machine table

On the bed 1 are located the gear box and the driving system of the faceplate 4. This is provided with a driving gear 5 and is centered radials by means of the bearings system 3; it is seated and unloaded axially on the pockets 2. As soon as the faceplate was positioned, the oil film thickness in pockets was adjusted using the eight flow control valves as shown in Figure 16.

a. b.

Fig. 16. Adjusting and checking the thickness of the oil film

The supports of the four dial gauges 3 (positioned every 90^0) were clamped on the bed 1. The dial gauges feel the faceplate 2.
An initial adjustment was made on the occasion of the tests; the faceplate was the only load and it was obtained a uniform film with thickness of 0.15 - 0.17 mm.
The results of the real measurements are shown in Table 2.

Table 2: The results of the measurements

W [daN]	p_{Med} [bar]	h_{Med} [mm]	Q [l/min]
20000	1.8-2.2	0-15-0.17	31

The size *e* was made at the value of 0.2 mm. In this case, the bearing in Figure 10 begins to be stressed only if the pressure of ~2 bar is exceeded under conditions of use, in idle running, of a flow rate higher than 50-55 l/min. This value corresponds to the scale division of ~7.5 on the eight flow control valves. The force taken by this bearing, with no workpiece on the faceplate, has the approximate value:

$$F = 75\ 000(p_P - 2)K_W \text{ [daN]} \tag{18}$$

In the relation (18) it was noted: p_p - average pressure in pockets [bar], F - thrust force transmitted to the bearing [daN].
A specific problem for these machines is the accidental shutdown of the power supply voltage. In this case, if the machine is operating, because of the inertia of the faceplate – workpiece system, there is the risk of seizing occurred due to the disappearance of the oil film. To avoid such accidents, the machine is equipped with an emergency power system that will power the electric motor 1EM1 if needed. There is also a variant where the system is provided with pneumatic-hydraulic accumulators so dimensioned to be able to supply oil to the pockets long enough to allow the faceplate to stop completely [9, 13].

7. Conclusions

The hydrostatic suspension systems are usually used for the heavy duty vertical lathes with faceplate bigger than 5 000 mm.
These systems can operate at constant pressure or constant flow rate. The variant of using flow control valves, namely those operating at constant flow rate, is the preferable option taking into consideration the increased possibilities of adjustment and also the system stability. It should be mentioned that this variant is more expensive than the one which involves adjustments at constant pressure.
In the design phase of the hydrostatic units for these machines, account must be taken of as many factors as possible which may influence the system size. These factors include: minimum and maximum loads that will be put on the faceplate, the intended thickness of the film, maximum revolutions per minute (speed).
Even using specialized programs of calculation and simulation, certain safety margins must be provided due to the complexity of phenomena and variations in sizes as well.
Thus, the viscosity and density of oil – important parameters of calculation – are influenced by the inherent variations of temperature. Exceeding the imposed maximum oil thickness may result in the instability of the system or even its destruction. Thus, forcing the unit by increasing the flow rate in the absence of the load can entail the destruction of the axial - radial bearing located in the centre of the faceplate. The assembling of this bearing is very accurately closed with a chain of sizes determined during the tests phase of the machine.
Special attention must be paid to the oil filtering systems because of the specificity of the hydraulic equipment used. This one comes into contact with the atmosphere of the machining hall and is inevitably clogged. It is recommended to use oversized filters in terms of flow rate and to implement solutions that allow the replacement of filtering cartridges without having to turn off the machine.
The continuous monitoring of the general pressure and of each pocket pressure avoids the seizing which could have particular effects in the case of such machines.
The remanufacturing of this machine, made more than 30 years ago, resulted in a modern machine that is able to compete technically with a brand-new machine.

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