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SOLUTIONS FOR HYDRAULIC BALANCING OF THE POSITIONING AND FEED KINEMATIC CHAINS IN PORTAL MILLING MACHINES

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Abstract: *The authors of this paper introduce hydraulic balancing systems for vertically moving heavy loads. The proposed system is suitable for heavy duty machine tools with crossrail that can be positioned depending on the size of the working piece. The positioning of the crossrail is relatively rare. Both crossrail and rail heads have big weights that can exceed 10–15 t. The entire weight of the assembly, friction and inertial forces are taken over by the positioning kinematic chain. In order to reduce the overall size and price of the kinematic chain elements, the solution is the crossrail balancing. The hydraulic balancing is made by means of two hydraulic cylinders symmetrically positioned. One shows the principle hydraulic diagrams, methodology of calculation and results of some simulations. For the crossrail balancing, a separate hydraulic unit for each of the two cylinders will be available. Experimental achievements are shown for a portal milling machine of FLP 2000 type. The specific feature of this system for crossrail balancing is that it uses the regular hydraulic equipment and having low prices. The hydraulic units for such balancing can successfully replace the complex and expensive units used for feed kinematic chains, in the case of CNC heavy duty machine tools. As well, the authors present a balancing hydraulic system for feed kinematic chains which is acting on crossrail heads of the heavy gantry machine tools, usually used for axes Z.*

Key words: *heavy machine tools portal type (gantry), positioning kinematic chain, feed kinematic chain, crossrail, crossrail heads, hydraulic balancing.*

1. GENERALITIES

In the case of heavy duty machine tools provided with crossrails [1, 2], such as portal milling machines (with table or portal movable), vertical lathes and guideways grinding machines, the positioning of the crossrail is relatively rare, when the height of the workpiece is changed. The crossrail and the rail heads too have big weights that can exceed 10–15 t. In these cases, the kinematic chain for crossrail vertical driving, with the key kinematic diagram shown in Fig. 1, is a positioning kinematic chain [3, 4].

The actuation will be not frequently performed, outside the cutting process. From the study made by the authors throughout the collaborations of the last 5 years with machine tool producers on the machines belonging to the family of vertical lathes included in SC 14–43 range and to the family of gantry milling machines [1, 2 and 5] it resulted that the kinematic chain for crossrail positioning is used 1–2 times a day in over 50% of the cases and it operates even more rarely than once a month in over 10% of the cases.

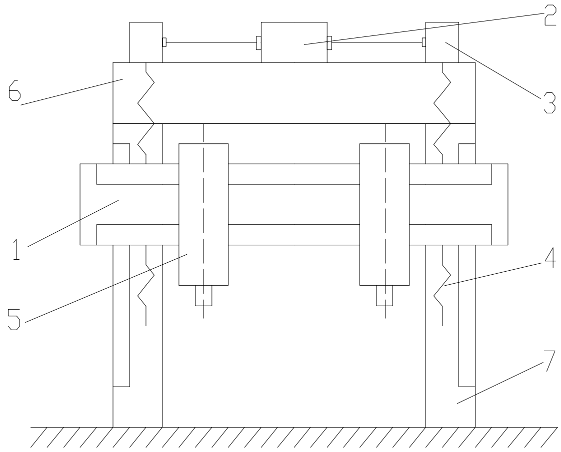

Fig. 1. Crossrail block diagram of portal / gantry milling machine or of vertical lathe: 1 – crossrail, 2 – two outputs electric motor, 3 – reducers, 4 – leading screws (usually trapezoidal ones), 5 – crossrail head (heads), 6 – cross-beam, 7 – columns.

In order to actuate the positioning kinematic chain, the crossrail head (or heads) 5 is (are) moved on the crossrail 1 to a position that ensures a symmetrical taking-over of the loads by the two leading screws 4 [1]. One performs the unlocking and the unindexing of the crossrail 1 [3, 4], the motor 2 starts and drives the screws 4 by means of the reducers 3.

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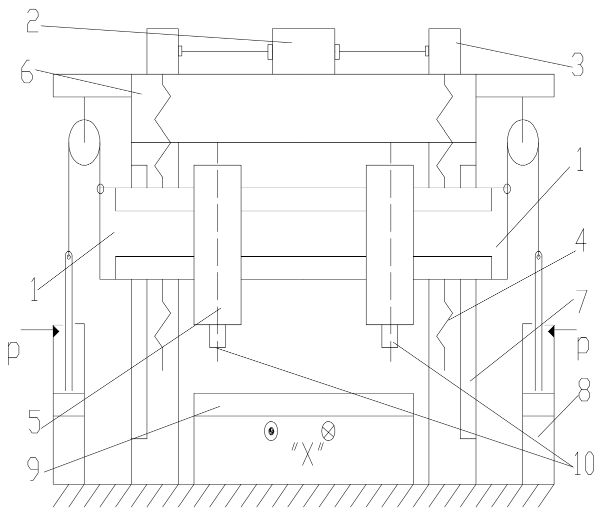
In order to reduce the motor power and implicitly to reduce the overall size/price of the kinematic chain elements, the solution is the crossrail balancing. This can be made mechanically by means of counterweights or hydraulically [3, 5].

Taking into account the complications specific to the mechanical balancing but also the current price of the metal (a counterweight of 10 t costs about 15000 €), the solution recommended by the designers and manufacturers of heavy duty machine tools is the hydraulic balancing.

The hydraulic balancing is made by means of hydraulic cylinders symmetrically positioned, as in Fig. 2. Most important elements are indicated on the block diagram (Fig. 2, a) and also on the view of FLP 2000 portal milling machine (Fig. 2, b). The positioning kinematic chain will have the same structure, but due to the distribution of efforts on the screws and also on the cylinders 8 it will have a smaller size motor 2 and reducers 3, and also leading screws 4 with smaller diameters. In Fig. 2, it was denoted by 9 the movable table on axis X and by 10 the slides that are going to be hydraulically balanced.



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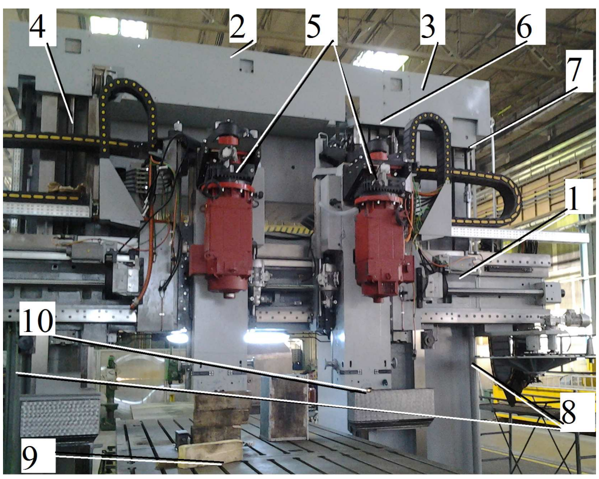


b

Fig. 2. FLP 2000 - portal milling machine: *a* – block diagram; *b* – machine front view; 1 – crossrail, 2 – two outputs electric motor, 3 – reducers, 4 – leading screws (trapezoidal preferably), 5 – crossrail head (heads), 6 – cross-beam, 7 – columns, 8 – crossrail balancing cylinders, 9 – movable table on axis X, 10 – rams (subject to hydraulic balancing).

The force developed in steady regime by each of the two cylinders is expressed by:

$$F = p \cdot \frac{\pi \cdot (D^2 - d^2)}{4}. \qquad (1)$$

where: D – piston diameter, d – rod diameter, p – pressure in the hydraulic unit.

Depending on each machine, the forces developed by the two cylinders can be smaller than the resistance forces (under-balancing) or may exceed them (over-balancing).

2. BALANCING HYDRAULIC SYSTEMS

Usually, balancing systems with reduction valves, pumps with pressure regulators or closed loop systems and accumulators [7, 8] are used for the vertical feed kinematic chains in the case of housings, rams and even crossrails if it is about working axes. These systems use expensive devices and because of the working mode (full time) they raise problems in terms of heating of the respective units. The working axes, controlled by the control unit, must be driven even during STOP phases as the position is maintained by controlling the size read on the scale by the transducer.

The balancing hydraulic units used in these cases are operating permanently, which involves adopting specific measures to reduce heating. Among these measures one can mention [5]:

- use of reduction valves, usually for smaller loads;
- use of variable flow pumps with pressure regulator, option that gives great results at acceptable prices up to pressures of 100–150 bar;
- use of a battery of pneumo-hydraulic accumulators, in which case the price is rather high and the balancing is made with variable forces, depending on the position of the moved slide;
- use of complex electro-hydraulic units involving using proportional hydraulic equipment and/or systems of electric recovery of the energy. These systems are the most expensive ones and are suitable for very large machines (AFP 230).

In the case of the kinematic chain used for positioning exclusively, the hydraulic balancing should not be permanent. It is actuated before each positioning and becomes usable only after ensuring crossrail unindexing [3].

3. BALANCING HYDRAULIC SYSTEM FOR POSITIONING KINEMATIC CHAINS OF HEAVY DUTY MACHINE TOOLS

The balancing unit shown in Fig. 3 is intended to supply one of the balancing cylinders 8 (Fig. 2). A separate unit for each of the two cylinders will be available for the crossrail balancing. The two units can be placed on the same tank or on two different tanks.

The pumps 3 and 4, actuated by the motor 5, suck the oil out of the tank 1, through the strainers 2, and send it to the unit. The high flow pump 3 supplies the balancing circuit through the check valve 6. The low flow pump 4 feeds the balancing circuit directly. The pressure in the balancing circuit is confirmed by the pressure relay 7.

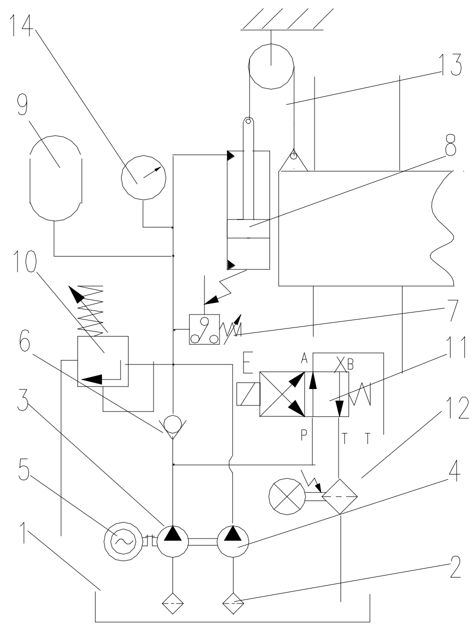


Fig. 3. Hydraulic balancing unit: 1 – tank, 2 – suction strainer, 3, 4 – pump, 5 – electric motor, 6 – check valve, 7 – pressure switches, 8 – balancing cylinder, 9 – accumulator, 10 – pressure valve, 11 – solenoid directional valve, 12 – return filter,13 – pulley system, 14 – pressure gauge, *E* – solenoid.

The maximum value of the balancing pressure is regulated by means of pressure valve 10. The flow of the big pump 3 can be discharged back to the tank, without supplying the balancing circuit, by means of the distributor 11. The pressure can be maintained in the circuit under certain conditions after the pumps stopping by means of the accumulator 9 [6] supplied by the balancing circuit in the same time as the cylinder 8. When the pump 3 flows directly to the tank, the oil filtering is performed through the return filter 12. The pressure in the balancing circuit is read on the pressure gauge 14.

There are three phases that can be determined in the operation of the diagram above:

1. **Lifting stroke**. The electric motor is started and as soon as the relay 7 confirms that the set pressure is reached, the voltage is set in the electromagnet *E* of the distributor 11.
   At this stage, the pumps supply the flow rates Q_1 and Q_2 ($Q_1 > Q_2$), with the condition that the resulting flow must verify in steady mode the condition:

$$Q_U = S \cdot v < Q_1 + Q_2 . \tag{2}$$

The following notations were used in the relation (2): v – crossrail travel velocity, S – useful area of the cylinder, Q_U – flow required for rise.
The flow discharged by the pressure valve Q_{PV} is:

$$Q_{PV} = Q_1 + Q_2 - S \cdot v . \tag{3}$$

The lost power in the lifting stage through the balancing unit is:

$$P_{LU} = p \cdot Q_{PV} . \tag{4}$$

2. **Stop after lifting**. In the first stage, the voltage is set off in the electromagnet *E*; then the motor 5 can be stopped if there is no other positioning to follow and if the crossrail is locked.
   The power lost in this phase is null.
3. **Descent stroke**. In order to perform this phase, the electric motor 5 starts and the system waits for the confirmation of the pressure by the pressure relay 7**.** The electromagnet *E* is not actuated during this phase. If the travel is performed also at the speed *v*, one can consider:

$$Q_D = S \cdot v , \tag{5}$$

$$Q_{PV} = Q_D + Q_2 , \tag{6}$$

$$P_{LD} = p \cdot Q_{PV} . \tag{7}$$

where: Q_D – necessary flow at fall, P_{LD} – lost power under the form of heat. In this power, only the component ΔP results from the hydraulic unit, where ΔP is:

$$\Delta P = p \cdot Q_2 . \tag{8}$$

At the *Stop* command after descend, the desired position is confirmed and the positioning kinematic chain is stopped, achieving the indexing and blocking followed by the command for electric motor stopping.

In all these phases, the accumulator 9 ensures the start flow peaks and also maintains the volume of pressurized oil [6, 7].

Usually, the crossrail in such machine tools, after it is indexed in desired position, is locked on the guides. There are two types of blocking:

- locking with disc springs and hydraulic unlocking,
- hydraulic locking and unlocking maintaining the state *blocked* by using another accumulator.

Figure 4 shows the operation of the installation for lock / unlock the crossrail in the machine FLP 2000 [8].

Pressured 9 (not shown) feeds the cylinders used for blocking on the columns 2. At the reaching the adjusted pressure by the pressure relay 3.1, the source can be disabled. The accumulator 6 of volume V_0 loaded at pressure p_0 [6] maintains the cylinders in the blocking position. If the pressure drops to the adjusted value of 3.2 in the pressure relay, the source is reactivated. The tightness of the circuit is improved by check valve. The pressure in the blocking can be viewed on the gauge 7. If we want the crossrail unlock, the directional valve 4 is actuated by the solenoid and the accumulator and its circuit are discharged due to valve 5 opening. This state is confirmed by the pressure relay 3.3.

Figure 5 shows partially the real installation.

4. BALANCING SYSTEMS FOR RAMS IN HEAVY DUTY MACHINE TOOLS

For balancing the two rams in this machine tool two identical installations were used having the principle schematic shown in Fig. 6. The entire weight of the assembly ram, working head, gear box, electric motor and necessary specific devices mounted.

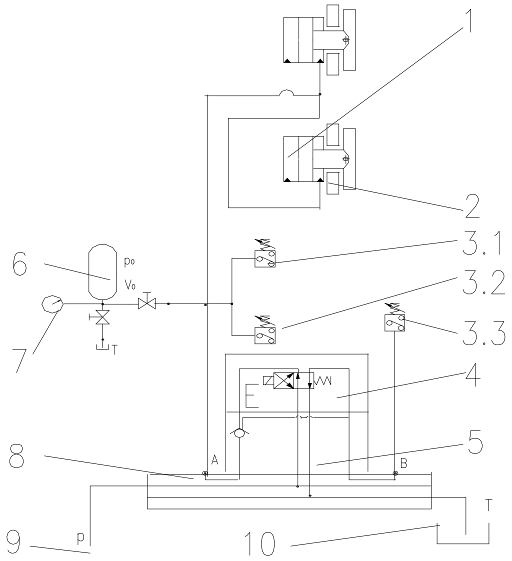


Fig. 4. Schematic of the installation for lock / unlock the cross-rail in the machine FLP in 2000: 1 – blocking cylinders, 2 – blocking zone on columns, 3.1, 3.2, 3.3 - pressure relays, 4 – distributor, 5 – pilot operated check valve, 6 - accumulator, 7 – manometer, 8 – basic plate, 9 – pressure source (not shown in the schematic), 10 - tank.

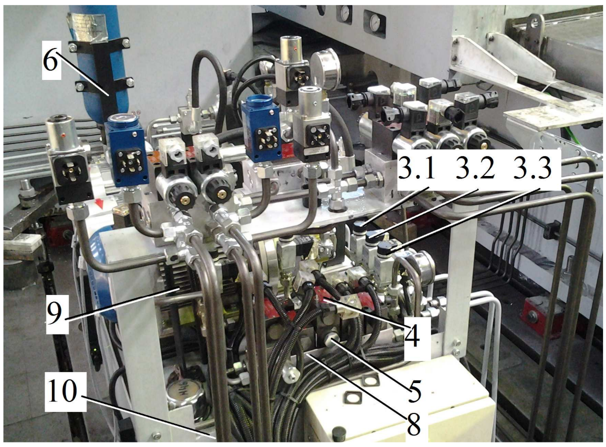


Fig. 5. Real balancing hydraulic unit corresponding to Fig. 4: 3.1, 3.2, 3.3 – pressure relays, 4 – distributor, 5 – unlockable check valve , 6 – accumulator, 8 – basic plate, 9 – pressure source, 10 – tank.

The variable flow pump having a pressure regulator 2 sucks the oil from the tank 1 and supply it through the check valve 12 to the cylinders 8. These balance the load 7. The pressures upstream and downstream the check valve are viewed on the gauges 11. The maximum pressure value is indicated (mandatory in CNC machine tools) by the pressure relay 9. The pressure valve 5 is set to a pressure higher than that adjusted in the pump 2 regulator. The installation also has for the start phases a pneumo-hydraulic accumulator 10. The pressure valve is fixed between the plates 4 and 6.

In the stop phase, the pump supplies a pressure adjusted by the regulator and a minimum flow, theoretical null. During this phase, the pressure valve 5 is completely closed. The energy consumption during this phase is negligible, the installation working as a blocking unit, being supported by the accumulator.

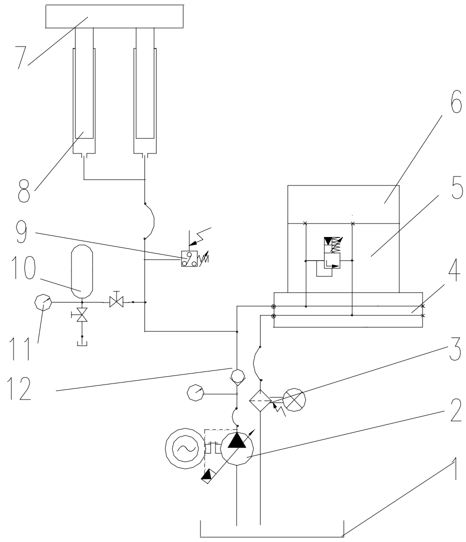


Fig. 6. Schematic of the ram balancing: 1 – tank, 2 – variable flow pump with pressure controller, 3 – filter, 4 – basic plate for apparatus, 5 – pressure valve, 6 – end plate, 7 – balanced ram, 8 – balancing cylinders (two for each ram), 9 – pressure relay, 10 – accumulator, 11 – gauges, 12 – check valve.

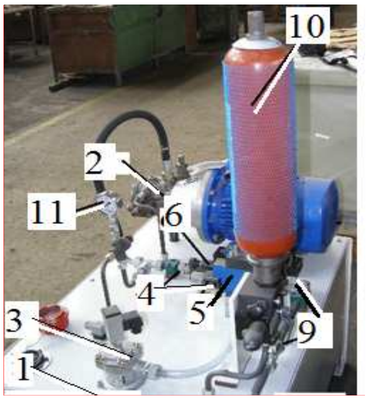


Fig. 7. Balancing hydraulic unit with pressure regulator pump.

The construction of the hydraulic unit for balancing one ram is shown in Fig. 7 where the notations from Fig 6 were kept.

4.1. Operation of the ram balancing installation

When the load 7 goes up by means of the feed kinematic chain, the pump sends the necessary flow, with the pressure set at the regulator. It can be considered stationary mode:

$$p_1 \cdot S = G^{'} , \tag{6}$$

$$p_1 \leq p_{RP} , \tag{7}$$

$$v \cdot S = Q < Q_P , \tag{8}$$

where: p_1 – pressure necessary for balancing, p_{PR} – pressure set at pump regulator, v – velocity imposed by the feed kinematic chain, S – useful surface of the balancing cylinders, Q – flow provided by the pump, Q_P – maximum flow provided by the pump, G' – balanced weight (greater, equal or smaller than G).

In this phase, the pressure valve 5 is completely closed. When the load 7 goes up, the energy consumption in the balancing system is not negligible and has maximum values in the fast movement phases.

When load 7 goes down, the check valve 12 is closed; the pump is actuated by the regulator towards the minimum flow value (theoretically zero). The pressure valve 5 is open; the fluid is discharged directly to the tank. The opening pressure of the valve must be superior to the pressure adjusted by the regulator. In these circumstances, it can be considered:

$$p_{PV} \cdot S = G'' , \quad (9)$$

$$p_{PV} > p_{RP} , \quad (10)$$

$$Q \to 0 . \quad (11)$$

The following notations have been used in the relations above: p_{PV} – pressure adjusted at the pressure valve, G'' – balanced weight (bigger, equal or smaller than G).

The energy consumption in this phase is negligible. A possible overbalancing can be seen at feed kinematic chain level by increasing the necessary torque of the electric motor of the feed kinematic chain.

5. SIMULATION OF BALANCING UNIT OPERATION OF THE POSITTIONING KINEMATIC CHAIN OF CROSSRAIL

Specialized programs were used for simulating the balancing unit in Fig. 3 [8]. The following values were taken into consideration for one of the two balancing cylinders: S = 40 cm^2, p = 175 bar, v = 2 m/min, Q_1 = 7.5 l/min, Q_2 = 4 l/min, G – weight to be balanced on a cylinder (7 000 daN). The power of the pumps driving motor is P_{EM} = 5.5 kW. The accumulator has the effective volume V_0 = 2.5 l and is preloaded with nitrogen at the pressure p_0 = 100 bar.

The characteristic in Fig. 8 was obtained for the crossrail travel upwards. After starting the pumps, one waits for the reaching of the pressure adjusted by means of the not actuated hydraulic distributor (E-). If the intended pressure is reached (confirmed by the pressure relay) the electromagnet can be switched on (E+). The flow discharged through the pressure valve Q_{PV} is the flow available for the crossrail travel upwards. The velocity imposed to the crossrail must verify the relation (2).

In the case of the crossrail travel downwards, the electric motor is started and the electromagnet E is no more actuated. As it can be seen in the graph of Fig. 9, the flow of pump 4 is discharged through the pressure valve. If the feed kinematic chain is started (START), the flow of pump 4 but also the flow expelled by the piston that descends with velocity v are discharged through the same valve.

The simulation of the operation is useful to the designer of hydraulic units allowing the selection of pumps and of the accumulator that can ensure optimal functioning.

In the case of such machine tools (heavy and very heavy duty ones), if the crossrail must be positioned, the

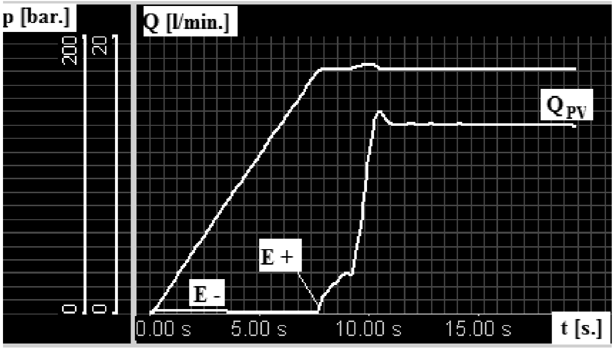


Fig. 8. Simulation of crossrail travel upwards.

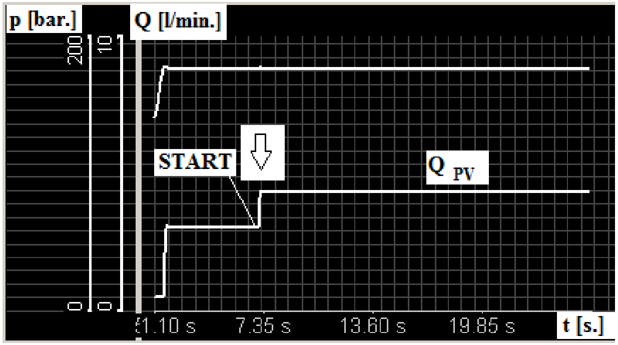


Fig. 9. Simulation of crossrail travel downwards.

times needed for the confirmations can reach values of the order 10 s. But this is not an impediment taking into account the low frequency of such operations.

In this paper we do not insist on the balancing installation of the ram.

6. EXPERIMENTAL RESULTS

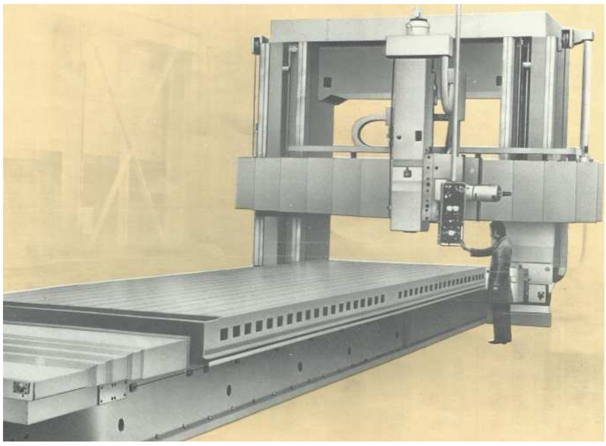
Figure 10 shows FLP 2000, a portal milling machine [1, 2] that required, on the occasion of its retrofitting, a new design of the hydraulic unit for crossrail balancing.

In Fig. 10,a the original machine tool is presented having a single milling unit. During refabrication some new requirements emerged with regard to the initial variant:

- displacement of the crossrail should be done only for positioning followed by indexing and blocking;
- on the crossrail two identical working units are considered, each being heavier than the original one;
- balancing unit of the crossrail works only in positioning phases;
- two additional tool magazines are provided.

Because the machine retrofitting involves also introducing the second working unit and adapting the balancing installation of the crossrail, the authors studied two possible balancing variants. In this paper only the first variant is presented. The second one is going to be achieved. Initially, the balancing system of the crossrail enables balancing specific to a feed kinematic chain [1, 5] for a crossrail loaded with a working unit equivalent to 5 500 daN.

The crossrail was extended and two tool magazines (each one with 24 stations) were added. Under these conditions, a new balancing unit was designed, consider-



a

b

Fig. 10. FLP 2000 machine tool: *a* – before retrofitting; *b* – after retrofitting.

ing that the crossrail movement is only a positioning one. The whole assembly is much heavier and the processing that is to be done on this machine does not need a motion of feed type in this assembly. As soon as the intended position is reached, the crossrail is indexed and locked. After this locking, the balancing unit is no more useful and it can be stopped until a new positioning.

The new hydraulic unit whose original balancing cylinders were kept includes a double gear pump driven by a 5.5 kW motor and by an accumulator of 4 l. The maximum working pressure is 180 bar. The entire unit is started only in the phases of crossrail positioning, fact that eliminates the risk of heating. The unit ensures the taking over of a static load of 13 000 daN; it is also helped by a counterweight system that is already placed on the machine.

Currently we try making simplified balancing equipment allowing permanent balancing of the rams and only during crossrail positioning to achieve their balancing.

7. CONCLUSIONS

If large masses are positioned on the heavy duty machine tools it is possible to create simple and cheap hydraulic units that can replace – in the case of a not frequent use – the conventional balancing units specific to the balancing units included in feed kinematic chains. The created hydraulic unit required hydraulic devices about 1 500 € cheaper than the devices usually used for such balancing.

If the crossrail travel is a positioning movement, it is recommended to be done with the rail heads parked symmetrically, so that the driving motor can achieve mechanically the simultaneity of screws rotation thanks to the rigidity of the kinematic chain. The positioning of the crossrail is achieved under optimum conditions when the system is also provided with mechanical locking and indexing and hydraulic unlocking.

The balancing systems studied are suitable for machine tools from the family of milling and grinding portal machines having movable portal or table and also of large vertical lathes.

If one wants the crossrail travel to be a feed movement, in any position of the rail heads, it is recommended to actuate the crossrail by means of two motors, one for each screw. In this case, the use of “GANTRY” function becomes mandatory, as well as the use of balancing units with pumps with pressure regulators, as for any vertical feed kinematic chain.

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