ISSN 1453 – 7303 **"HIDRAULICA" (No. 1/2019)**
Magazine *of Hydraulics, Pneumatics, Tribology, Ecology, Sensorics, Mechatronics*

Heavy Duty Machine Tools – Hydraulic Balancing of the Kinematic Chains

Prof. PhD Eng. **Dan PRODAN[1],** Prof. PhD Eng. **Anca BUCUREȘTEANU[1],**
Assoc. Prof. PhD Eng. **Adrian MOTOMANCEA[1]**

[1] University POLITEHNICA of Bucharest, prodand2004@yahoo.com, ancabucuresteanu@gmail.com, adrian.motomancea@deltainfo.ro

Abstract: *The authors of this paper introduce some of the hydraulic balancing systems for the large rams, cross-rails and housings which travel vertically. These systems are used for the feed and/or positioning kinematic chains of the heavy-duty machine tools. The paper presents the hydraulic basic diagrams, the methodology of calculation and the manufactured units. These low-cost systems are equipped with usual hydraulic devices. The balancing hydraulic units can be used for DC or CNC machines.*

Keywords: *Heavy duty machine tools, feed/positioning kinematic chain, hydraulic balancing*

1. Necessity of balancing

In the case of heavy duty machine tools (vertical lathes, gantry type milling machines, boring and milling machines), there are different sub-assemblies such as housings, rams and cross-rails that are moved vertically by means of the feed/positioning kinematic chains [1, 2, 3]. The mass of these sub-assemblies ranges from 500 to 15000 kg. Their drive kinematic chain is shown in Figure 1.

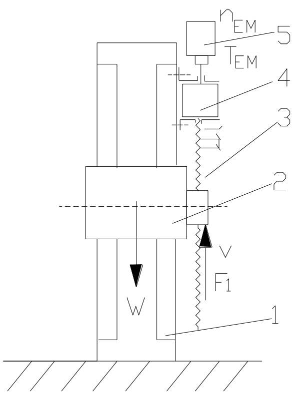


Fig. 1. Kinematic diagram of the vertical feed kinematic chain without balancing

The load 2 is moved on the guideways 1 by the feed kinematic chain which includes the leading screw 3, the reducer 4 and the electric motor 5. The transfer ratio of the reducer 4 is i (i<1) and the screw pitch is l [2, 3, 4].
For going up, if the electric motor develops the rotational speed n_{EM} and the torque T_{EM}, taking into account the weight of the entire assembly as W and neglecting the frictions, in stationary conditions, it is possible to determine the force F_1 and the speed v by means of the relations:

$$F_1 = W = \frac{2\pi}{il} T_{EM} \tag{1}$$

$$v = il n_{EM} \tag{2}$$

For example, for W = 50000 N, i = 1/2, v = 9 m/min, l = 20 mm, the motor should have the rotational speed n_{EM} = 900 RPM and the torque T_{EM} = 80 Nm. The power required in this case is P = 7.5 KW.
In order to unload the feed kinematic chain, which entails the use of feed motors with reduced torque and reduced power implicitly, it is necessary to use mechanical balancing, with

counterweight or hydraulic balancing. Mechanical balancing involves the use of counterweights as shown in Figure 2.

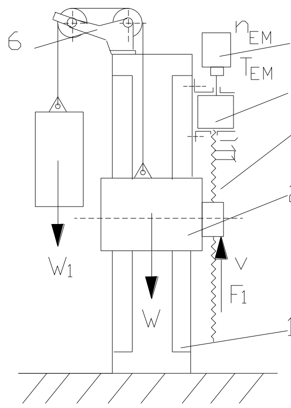


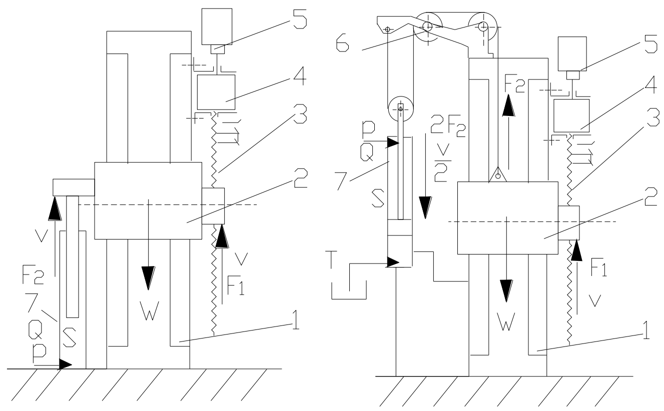
Fig. 2. Kinematic diagram of the vertical feed kinematic chain balanced with counterweight

In this case, the counterweight W_1 balances totally or partially the weight W through the agency of the pulley system 6. The force developed in the feed kinematic chain, in stationary regime, will be equal to the difference between W and W_1.
The mechanical balancing has a series of disadvantages:

- it doubles the mass of the system, so that the performances are diminished in dynamic conditions. The acceleration of a mechanical system is inversely proportional to its mass. When the mass is doubled, it is obvious that the acceleration decreases accordingly;
- the structure of the machine has the size necessary for a large system, so it will be a solid one, made of a large amount of material;
- the counterweight requires a guiding and safety system;
- faulty pulley system entails the risk of an uncontrolled fall of the counterweight that could damage the entire machine.

2. Hydraulic balancing

In the case of a hydraulic balancing, the counterweight shown in Figure 2 will be replaced by the hydraulic cylinder 7 with the useful cross section S as in Figure 3. This one is supplied from a hydraulic source at a pressure p with the proper flow, marked with Q. The other notations in Figure 3 are similar to the notations used in the previous figures.



a. b.
Fig. 3. Balancing systems with hydraulic cylinders

In both cases, the sub-assembly 2 is moved by the feed kinematic chain formed of the screw 3, reducer 4 and motor 5 but also by the hydraulic cylinder 7 [1]. The possible losses of oil are recovered by the drainage on path T.

2.1 Mathematical models in stationary regime

Figure 3a shows the case in which cylinder 7 is directly connected to the balanced sub-assembly **2**. In this case, we shall consider:

$$F_1 = W - F_2 = \frac{2\pi}{il} T_{EM} \tag{3}$$

$$F_2 = pS \tag{4}$$

$$Q = Sv = Siln_{EM} \tag{5}$$

If the balancing is made as shown in Figure 3b, the relations above become:

$$F_1 = W - F_2 = \frac{2\pi}{il} T_{EM} \tag{6}$$

$$F_2 = S\frac{p}{2} \tag{7}$$

$$Q = S\frac{v}{2} = S\frac{iln_{EM}}{2} \tag{8}$$

If the travels are shorter, usually up to 1000-2000 mm and the weights are smaller (the rams of the vertical lathes or Gantry-type machines) it is possible to use the variant shown in Figure 3a. In the case of the machines with large masses to be balanced along travels over 2500-3000 mm, such as the boring and milling machines, it is recommended to use the balancing variant shown in Figure 3b. Thus, it is possible to use cylinders with halved lengths in comparison with the cylinders used in the Figure 3a variant.

In stationary conditions, the good operation of the machine requires a hydraulic unit able to provide the necessary flow Q and the working pressure p. The velocity is ensured by the feed kinematic chain and the load (or part of it) is taken over by the hydraulic system. As for the CNC machines, the velocity is commanded and controlled by the equipment.

Regardless the balancing variant selected, the necessary power of the feed kinematic chain is significantly reduced thanks to the hydraulic unit with an available theoretical power P_H:

$$P_H = pQ \tag{9}$$

2.2 Mathematical models in dynamic regime

It is considered that the feed kinematic chain can ensure a maximum velocity v_{MAX} in an acceleration time t_A according to the characteristics shown in Figure 4.

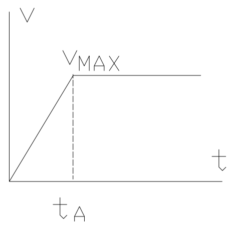


Fig. 4. Time dependence of velocity ensured by the feed kinematic chain

Taking into account the travel shown above, the loading diagram is the one presented in Figure 5.

The mass of the system is obtained by dividing the weight W by the gravitational acceleration (g), the friction forces in the two guideways are considered equal ($F_F/2$). The acceleration of the system is the second derivative of space; the first derivative is the velocity.

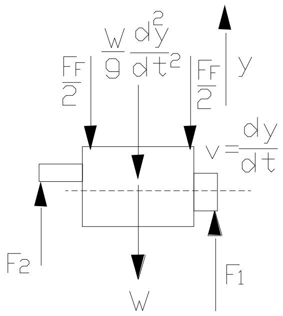


Fig. 5. The loading diagram

Regarding the diagram in Figure 3a, the following mathematical model can be taken into consideration in dynamic regime:

$$\frac{W}{g}\frac{d^2y}{dt^2}+b\frac{dy}{dt}+sgnvF_F+W=F_1+pS \tag{10}$$

$$Q=S\frac{dx}{dt}+ap+\frac{V_0+yS}{E_O}\frac{dp}{dt} \tag{11}$$

If the balancing system is the one shown in Figure 3b, the mathematical model in dynamic regime becomes:

$$\frac{W}{g}\frac{d^2y}{dt^2}+b\frac{dy}{dt}+sign(v)F_F+W=F_1+S\frac{p}{2} \tag{12}$$

$$Q=S\frac{1}{2}\frac{dx}{dt}+ap+\frac{V_0+\frac{y}{2}S}{E_O}\frac{dp}{dt} \tag{13}$$

Other notations used in the relations (10) - (13): b - linearization coefficient of force losses proportional to velocity (damping coefficient), sign - (signum function), a - linearization coefficient of flow losses proportional to pressure, V_0 - oil inactive volume in cylinder and feed pipe, E_O - elastic modulus of oil. On the basis of mathematical models, it is possible to make the analysis of the operation in dynamic regime by simulation, using specialized programs.
If the average volume under pressure in the cylinder is V_M = SC/2, where C is the cylinder stroke, it can be concluded that the own frequency of the cylinder is V_M = SC/2 in both cases:

$$f=\frac{1}{2\pi}\sqrt{\frac{E_O g S^2}{W V_M}} \tag{14}$$

In these conditions, it is recommended that the acceleration time t_A checks the condition [5]:

$$t_A \geq \frac{2.4}{f} \tag{15}$$

The control equipment of feed kinematic chain will provide acceleration consistent with the condition shown by the relation (15). So, in the case of a vertical lathe with the ram having the weight W = 20000 N, balanced by a cylinder with the useful surface S = 20 cm^2 and the total stroke C = 1000 mm, it results a minimum acceleration time of 0.4 s. The maximum acceleration will be a_{MAX} = 0.33 m/s^2 in order to reach the maximum velocity imposed v_{MAX} = 8 m/min.
The hydraulic source must provide the necessary pressure and flow in any phase: going up, STOP or going down. The necessary flow for the ram of the vertical lathe mentioned above, in maximum velocity regime, is Q = 16 l/min. Considering that a total balancing is made, the operating pressure in stationary regime is p = 100 bar.

In real conditions, the balancing is done so that the hydraulic unit does not take over exactly the balanced weight. If such a balancing was done when starting or stopping the operation, the movement would entail positioning and machining errors and also the wear of the kinematic chain elements because of the backlash occurred in the nut-screw mechanism. An over-balancing ($F_2 > W$) is recommended. In this case, the contact on only one flank of the lead screw must be maintained in order to reduce the return backlash.

3. Simulation of the balancing hydraulic systems running

In the design stage of a balancing unit, the simulation programs can be really helpful [5].
Let's consider a kinematic chain with the following values: W = 20000 N, i = 1/2, l = 10 mm, v_{Max} = 8 m/min. The actuation is made as in Figure 3a. Balancing cylinder has S = 20 cm^2, the stroke is C = 1000 mm.
Given these conditions, if the frictions are neglected and an acceleration time t_A is imposed, the necessary torque at the electric motor T_{EM} , for an unbalanced system, is:

$$T_{EM} \geq \left(W + \frac{W v_{MAX}}{g t_A}\right)\frac{il}{2\pi} \tag{16}$$

After making the replacements in the relation (16), for an acceleration time t_A = 0.5 s, checking condition (15), we shall obtain the condition $T_{EM} \geq$ 17 Nm. The maximum rotational speed of the motor will be:

$$n_{EM} = \frac{v_{MAX}}{il} \tag{17}$$

Following up the replacements, it will be obtained n_{EM} = 1600 RPM.
In the case of the above-mentioned kinematic chain, the balancing is considered actuated at the adjusted pressure of 110 bar, thus leading to the over-balancing.
Figure 6 shows the evolution of the pressure in the balancing cylinder and the evolution of the velocity of the subassembly 2 in Figure 3a. The acceleration time imposed at the electric motor is t_A = 0.5 s too and its maximum rotational speed n_{EM} = 1600 RPM.

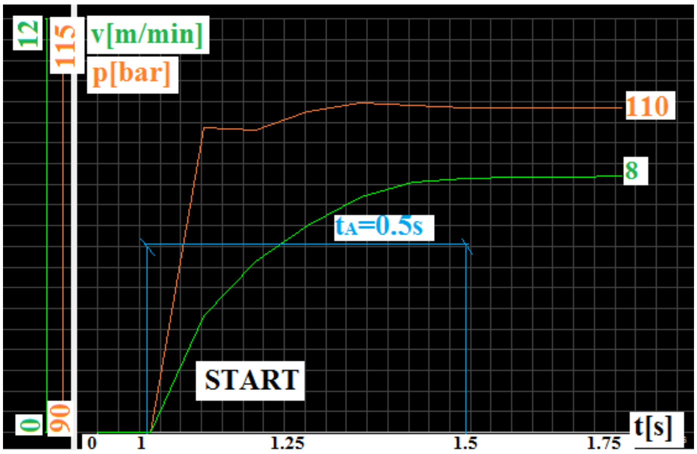


Fig. 6. Evolution of pressure and velocity in the balancing unit

The characteristics in Figure 6 were obtained for an electric motor that develops the theoretical maximum torque T_{EM} = 2 Nm. In these conditions, the system stabilizes after 0.5 s approximately. It is worth mentioning that the initial pressure in the cylinder must ensure the load taking over before the start of the feed kinematic chain.
By using a well-sized balancing unit, the efforts of the feed kinematic chain are diminished, therefore the behavior of the system is improved during rapid travel or when it operates in coordination with the travels on other axes (in the case of interpolation).

4. Balancing hydraulic units

There are several balancing diagrams used for the actuation of the vertical kinematic chains of the heavy duty machine-tools. The most frequently used three variants are presented hereby [6].

4.1 Balancing units with 3-way pressure reducing valve

The basic diagram of such unit is presented in Figure 7.

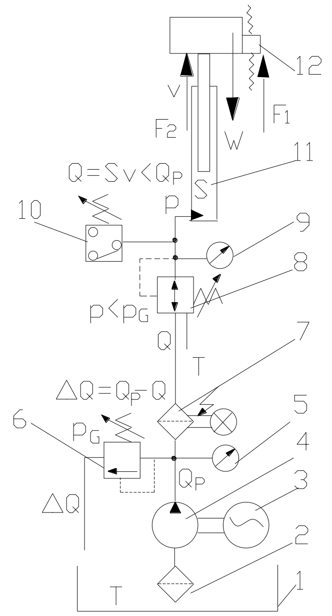


Fig. 7. Balancing hydraulic system with pressure reducing valve

The electric motor 3 drives the constant flow pump 4 which sucks the oil from the tank 1 (T) through the suction filter 2. The oil is filtered by the filter with clogging indicator 7. General pressure p_G is adjusted by means of the pressure relief valve 6. The balancing pressure p is adjusted at the pressure reducing valve 8. The two pressures p_G and p are read by means of the pressure gauges 5 and 9. Other notations used in Figure 7, in addition to the notations already presented, are:10 - pressure switch, 11 - balancing cylinder, 12 - feed kinematic chain, ΔQ - flow directed through the pressure relief valve 6. The flow supplied by the pump is Q_P and the flow used by the balancing system is Q.
The general conditions of operation are:

$$Q = Sv \tag{18}$$

$$v \in [-v_{MAX}, v_{MAX}] \tag{19}$$

$$Q \in [0, Sv_{MAX}] \tag{20}$$

$$\Delta Q = Q_P - Q \tag{21}$$

$$p < p_G \tag{22}$$

In the phases where the balanced assembly must not go upwards, the unit can also achieve other functions at the general pressure p_G. In the case of the CNC machines, the pressure switch 10 is the one that confirm the presence of the pressure required by the balancing. This diagram is relatively simple, it uses ordinary items and the price is low. The use of gear pumps is a simple and cheap solution, but has the disadvantage of a high consumption of energy even during the phases when the feed kinematic chain is not actuated. This energy consumption has also negative effects as for the system heating. For these reasons it should be used at pressures up to 40 bar and for flows smaller than 20 l/min. Therefore, some vertical lathes use this system that operate at the pressure p_G = 35 bar while the pump passes a flow Q_P = 16 l/min. In this case, 1.5 KW of the power supplied by the electric motor is used for the balancing. The hydraulic tank (with a volume of 220 l) is placed inside the machine bed and is used by the lubrication unit too.

4.2 Closed circuit balancing units

Figure 8 shows the hydraulic diagram of balancing in closed circuit for AF127 machine.

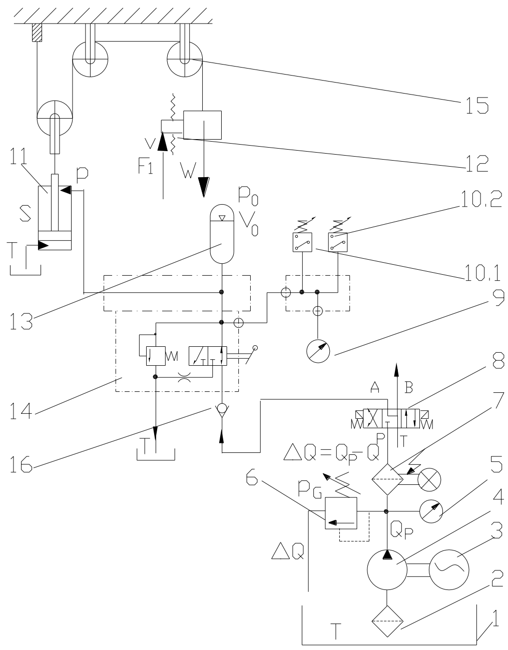


Fig. 8. Balancing system in closed circuit

The notations W, F_1, p, S, T, p_G, Q, ΔQ, 1 - 7, 9, 11 and 12 in Figure 8 are the same as the ones in Figure 7. Supplementary notations: 8 - hydraulic distributor, 10.1 – pressure switch for minimum pressure (p_m), 10.2 - pressure switch for maximum pressure (p_M), 13 - accumulator (accumulators), 14 - safety block, 15 - pulley system, 16 - check valve.

The closed circuit includes cylinder 11, accumulator 13, pressure switches 10.1 and 10.2 and check valve 16. Accumulator 13 has the volume V_0 and is loaded with nitrogen at the initial pressure p_0. After making the circuit with the cylinder piston in low position, the distributor 8 is clutched in the position that makes possible the circulation of the oil P→A and B→T. The closed circuit is fed through the check valve 16 up to a pressure equal to p_m. When this pressure is reached, the circuit is considered to be loaded, thus the distributor is declutched, path P is closed and paths A and B are connected to the tank (T).

The nitrogen was submitted to an isothermal transformation in the accumulator, according to the relation [7, 8];

$$p_0 V_0 = p_m V_M \tag{23}$$

An amount of oil entered the accumulator, entailing the decrease of the initial volume of nitrogen from the value V_0 to the value V_M.

In this phase, the feed kinematic chain is unloaded with a force F_2 expressed as follows:

$$F_2 = \frac{p_m S}{2} \tag{24}$$

If we consider that the balanced assembly 12 gets down by the quantity y, the nitrogen in the accumulator suffers an isothermal or adiabatic transformation. The transformation is an adiabatic one if the travel is made with high velocity, in the range of seconds. The isothermal transformation is made in a slow pace, in the range of minutes [7]. The relations that describe these transformations are listed below:

$$p_m V_M^{\gamma} = p_{(y)} \left(V_M - \frac{yS}{2}\right)^{\gamma} \tag{25}$$

$$p_m V_M = p_{(y)} (V_M - \frac{yS}{2}) \tag{26}$$

$$y \in [0,2C] \tag{27}$$

In these relations, C is the stroke of the cylinder 11. For nitrogen, it can be considered that γ = 1.4. We can notice that the balancing force is variable and it increases when the load goes downwards:

$$F_{2(y)=} \frac{p_{(y)}S}{2} \tag{28}$$

The maximum value of this pressure p_M is reached when the cylinder rod gets in the higher end of the travel. This pressure is signaled by the pressure switch 10.2. The unit is able to operate by the charge and discharge of the accumulator towards and coming from the cylinder, even if the pump does not supply this circuit. If the connections P→B and A→T are made, the pump can supply other consumers too, through the path B. When the pressure drops under the adjusted value p_m, as shown by the pressure switch 10.1, the recharge of the circuit (P→A and B→T) shall be commanded.
The unit is relatively simple: it includes usual devices; the pump has constant flow and the energy consumption is very low (it occurs practically only when the circuit is charged). These are the main advantages of such systems. There are also some disadvantages, like:
- only one accumulator is not usually enough; batteries of accumulators are needed; therefore, the price of the unit gets bigger;
- the balancing force (F_2) is variable. It has a minimum value when the load is up and a maximum value when the load is down. These variations are felt by the feed kinematic chain and entail variations of the motor torque;
- the failure of the accumulator (accumulators) leads to the instantaneous immobilization of the machine because the feed kinematic chain is overstressed.

4.3 Balancing units with pumps with pressure regulator

A balancing unit with pressure regulator pump used for modern vertical lathes, Gantry-type machines and also for horizontal boring and milling machines (HBM-s) is presented in Figure 9.
The motor 2 actuates the pump with variable flow 3 which sucks the oil from the tank 1 (T). The general pressure p_1 is adjusted by means of the pressure switch of the pump. The pressure p_2 is regulated at the pressure relief valve 6 and it is usually higher by 5-10 bar than pressure p_1. The two pressures p_1 and p_2 can be read on the pressure gauges 4 and 8. The one-way check valve 5 closes if the pressure downstream exceeds the value p_1. The pressure switch 7 is set at a pressure p_3, smaller by 5 bar at the most related to p_1. This switch confirms the presence of pressure in circuit and in cylinder 9 too, making possible the actuation of the feed kinematic chain 10. The circuit also includes an accumulator 11 and the safety block of this one 12. The filtering of the oil in circuit is made by the return filter 13. When the unit starts operating, the circuit reaches the pressure p1. If no movement is commanded, the pressure regulator sets the pump in null flow position. In this case, the accumulator is charged at the pressure p_1. If the load travels upwards, the pump supplies the necessary flow depending on the velocity. When the load travels downwards, the one-way check valve 5 closes, the pump regulator passes the pump to the null flow position and the oil of the cylinder runs into the tank through the pressure relief valve 6 and the return filter 13. In this phase, the balancing is made at the pressure p_2. Actually, the pump supplies the circuit only during the moving upwards phases. In STOP phase and moving downwards phase, the pressure regulator declutches the pump, thus the consumed energy and the unit heating are reduced considerably.

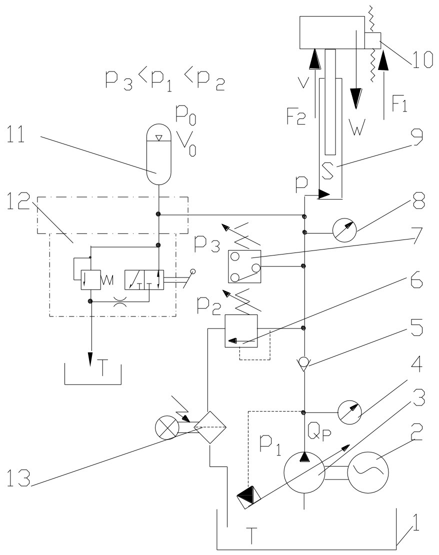
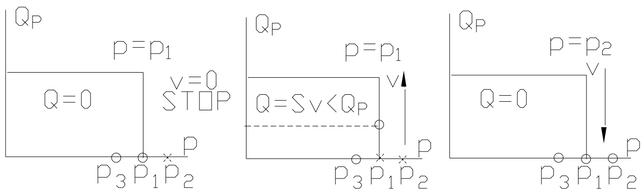


Fig. 9. Balancing unit with pump provided with pressure regulator

The pressure achieved in each one of these phases and the respective flows are shown in Figure 10 [5, 6].



a. STOP b. Going upwards c. Going downwards

Fig. 10. Flow-pressure characteristics in the three phases of operation

If the STOP phase occurs after the going upwards phase, the pressure is p_1 and the flow ensured by the pump is 0. If the STOP phase occurs after the going downwards phase, the pressure downstream the check valve and in the accumulator has value p_2; the pressure upstream the check valve has the value p_1 and the flow supplied by the pump is 0. In the going upwards phase, the pressure in the entire unit is p_1 and the pump supplies enough flow to achieve the imposed velocity but smaller than the maximum flow. In the going downwards phase, the pressure downstream the check valve and in the accumulator has value p_2; the pressure upstream the check valve has the value p_1 and the flow supplied by the pump is 0. The accumulator 11 operates between the pressures p_1 and p_2, making possible the flow peaks required by the acceleration phases. Due to this accumulator, the balancing unit operates as a locking unit too during the STOP phases. Figure 11 shows such a unit manufactured with usual devices [5].

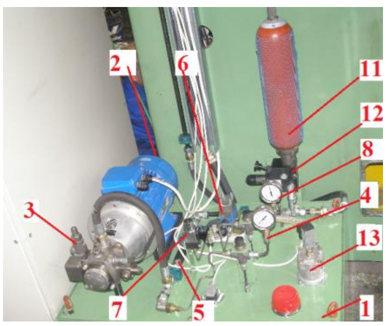


Fig. 11. Balancing hydraulic unit with pump with pressure regulator

Figure 11 kept the same notations used in Figure 9. The tank has a volume of 230 l and is equipped with a temperature probe and an electronic level gauge. The pump ensures a maximum flow Q_P = 20 l/min; the adjusted pressures are p_1 = 75 bar, p_2 = 85 bar and p_3 = 70 bar. The electric motor has the power 4 KW. The cylinder – not shown in Figure 11 – has an active cross-section of 20 cm^2. The maximum velocity imposed in the feed kinematic chain is v_{MAX} = 8 m/min and the balanced weight is W = 14000 N.
Because these units use the pressure regulator and also an accumulator, it is recommended to use them for balancing big loads, over 100000 N, with maximum velocities up to 10 m/min. Thus, in the case of a machine AFP 200 type, such balancing unit ensures the vertical displacement of a housing with W = 150000 N with maxima velocity of 8 m/min. Pump has a maximum flow Q_P = 100 l/min and the working pressure p_1 = 105 bar, p_2 = 115 bar, p_3 = 100 bar.

5. Conclusions

In the case of the vertical feed kinematic chains for heavy duty machines it is recommended that the load (slide, housing, ram) is balanced hydraulically. The hydraulic balancing should be preferred instead of the mechanical one given its higher performances, especially dynamic regime. In order to choose the hydraulic balancing variant, a study of the necessary power should be done. In a first phase, this power can be estimated as equal to the product of the weight to be balanced (W) and the maximum velocity of the respective kinematic chain. For small powers, up to 2-3 KW, it is possible to use balancing systems with pressure reducing valves and debit constant flow pump. The systems in closed circuit or the pumps with pressure regulator could be used in the case of higher powers, over 20000 N.
The systems in closed circuit use pumps with constant flow but need high volume accumulators. The balancing force in these systems is not constant and depends on the load position. The systems with regulator pumps can achieve the balancing of big loads but they need more expensive devices. The presence of an accumulator in this unit improves its behavior in transient regime and also ensures the operation of the system for locking the load during the STOP phases.
Regardless of the variant selected, the kinematic chain will be driven in such way to be in dynamic compliance with the hydraulic unit, providing a proper acceleration time.

References

[1] Prodan, Dan. *Heavy machine tools. Mechanical and Hydraulic Systems/Maşini-unelte grele. Sisteme mecanice si hidraulice*. Bucharest, Printech Publishing House, 2010.
[2] Perovic, Bozina. *Machine tools handbook/Handbuch Werkzeug-Maschinen*. Munchen, Carl Hanser Verlag, 2006.
[3] Joshi, P. H. *Machine tools handbook*. New Delhi, McGraw-Hill Publishing House, 2007.
[4] Teixido, C., J-C. Jouanne, B. Bauwe, P. Chambraud, G. Ignatio and C. Guerin. *Guide de Construction Mecanique*. Paris, Delagrave Edition, 2000.
[5] *** Catalogues and leaflets BOSCH REXROTH, BERARMA, ATOS.
[6] Prodan, Dan, Anca Bucuresteanu, Adrian Motomancea and Emilia Balan. "Balancing Hydraulic Units. General Considerations." *International Journal of Engineering and Innovative Technology (IJEIT)* Vol. 3, Issue 5, (November 2013): 234-237.
[7] Bucuresteanu, Anca. *Pneumatic Hydraulic Accumulators. Use and Modeling/Acumulatoare pneumohidraulice. Utilizare si modelare*. Bucharest, Printech Publishing House, 2001.
[8] Prodan, Dan, Anca Bucuresteanu, Tiberiu Dobrescu and Adrian Motomancea. "Mathematical Models on the Closed Circuit Balancing Systems in Machine Tools." *Applied Mechanics and Materials.* Vol. 859, (2017):170-176.