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Using the Pressure Intensifiers in Hydraulic Units of Heavy Duty Machine Tools

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Abstract: *This paper presents some applications of the pressure intensifiers used in the hydraulic units of the clamping/unclamping systems and unloading systems for the guideways of HBM and GANTRY type heavy duty machine tools. The paper makes a comparison between the systems with and without pressure intensifiers. There are presented mathematical models for the pressure intensifiers and the results obtained after a simulation using specialized programs.*

Keywords: *Pressure intensifiers, machine tools, guideways unloading systems*

1. Introduction

The operating principle of the pressure intensifiers [1, 2, 3] is shown in Figure 1.

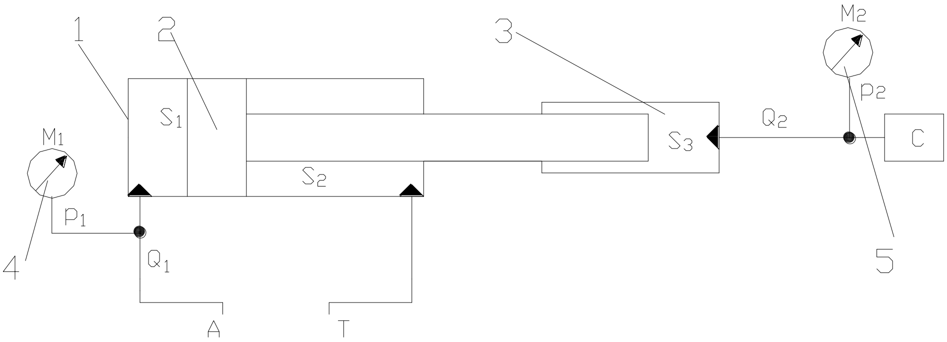


Fig. 1. Operating principle of the pressure intensifiers

If the cylinder 1 is supplied with the flow Q_1 at the pressure p_1 on the path A, the piston 2 moves in the pressure chamber 3 providing the consumer C with the liquid flow Q_2 at the pressure p_2. The pressures are read on the pressure gauges 4 and 5.
The pressure and the flow obtained depending on the surfaces S_1, S_2 and S_3 are the following ones:

$$p_2 = p_1 \frac{S_1}{S_3} \tag{1}$$

$$Q_2 = Q_1 \frac{S_3}{S_1} \tag{2}$$

In the absence of losses, the flow-pressure characteristic in stationary mode for the intensifier of Figure 1 is shown in Figure 2.
The two rectangles in Figure 2 have the same area equal to power P, where:

$$P = p_1 Q_1 = p_2 Q_2 \tag{3}$$

Consumer C can be a cylinder; in this case the flow Q_2 will feed a chamber of limited variable volume or an open circuit such as the systems of liquid jet cutting, in which case the flow Q_2 should be continuous.

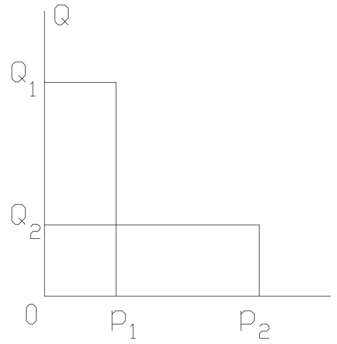
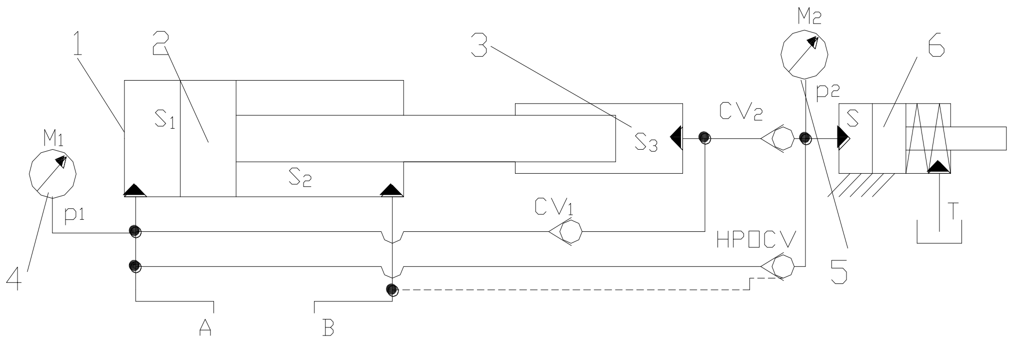


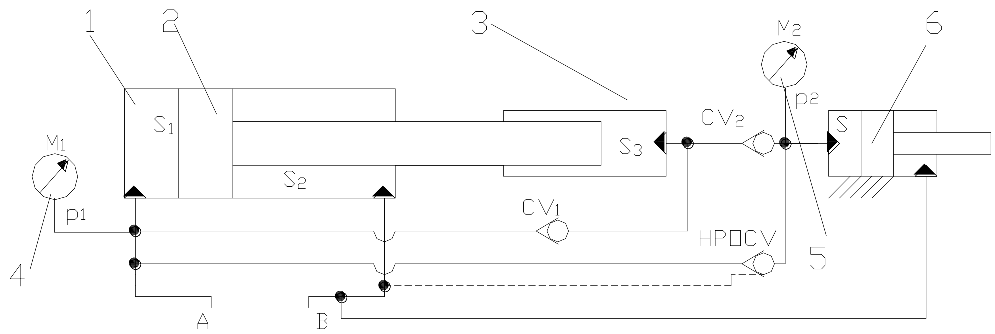
Fig. 2. Flow-pressure characteristic of the pressure intensifier

2. Hydraulic systems for guideways locking/unlocking and unloading

In the case of machine tools, the pressure intensifiers are used for powering the clamping/unclamping systems of the blanks or, in some situations, for powering the guideways unloading systems [1, 4]. In this case, the consumers C can be the single-acting hydraulic cylinders or, in the case of CNC machine tools [1], the double-acting hydraulic cylinders.
Figure 3 shows the actuation schemes for these systems.



a.



b.

Fig. 3. Using the pressure intensifiers in the hydraulic systems of machine tools

Figure 3 uses the same notations as in the previous figure, plus: 6 - consumer cylinders with useful area *S*, CV_1, CV_2 - one way check valves and HPDCV - Check valve hydraulically operated.
If the liquid supply is made on path A at a pressure lower than the value p_1, the liquid coming from chamber 3, but also directly, through the check valve CV1 and through the check valve hydraulically operated HPDCV, will get on the surface *S* of the consumers. When reaching the pressure p_1, before the cylinders 6 complete their entire travel, the pressure in chamber 3 gets to value p_2, the check valve CV1 closes and thus the locking or unloading of the guideways is

performed. If there are no losses and the oil compressibility ($E_{OIL} \rightarrow \infty$) is not taken into account, even after ceasing the supply with liquid on path A, the pressure p_2 is maintained on the surface S of the consumer thanks to the closing of valves CV_1, CV_2 and HPDCV.
By supplying the chamber B with liquid, the cylinders 6 will be discharged and will return to their initial position due to the spring for the variant shown in Figure 3a and thanks to the pressure (max. p_1) in the case of the consumers in Figure 3b. By supplying on the path B, the piston 2 of the intensifier return to its initial position, moving to the left up to the end of its travel.

3. Mathematical model of the pressure intensifiers in dynamic mode

Starting from the basic diagram of the intensifier shown in Figure 1, in dynamic mode, the following relations [3] can be taken into consideration:

$$M\frac{d^2x}{dt^2} + b\frac{dx}{dt} + c \cdot p_1 S_1 = p_2 S_3 \quad (4)$$

$$Q_1 = S_1\frac{dx}{dt} + ap_1 + \frac{V_{01}+xS_1}{E_O}\frac{dp_1}{dt} \quad (5)$$

$$Q_2 = S_2\frac{dx}{dt} - \frac{V_{02}-xS_2}{E_O}\frac{dp_2}{dt} \quad (6)$$

In the relations above there were also noted: M - mass of the piston, x - instantaneous movement of the piston, t - time, b - linearized coefficient of force losses proportional to the velocity (damping coefficient), c - friction coefficient of the piston operating in the low pressure zone, a - linearized coefficient of flow losses proportional to the pressure, V_{01} and V_{02} - initial volumes of liquid in the two champers, E_0 - elastic modulus of the liquid.
One can notice that the mathematical model is formed of non-linear differential equations. The solution of these equations by classic methods implies their linearization. The mathematical models allow understanding better the operation of the system. At the present moment there are specialized programs that enable the simulation of intensifiers operation without the need to elaborate the mathematical model. We hereby present the results obtained by simulation of a system provided with a pressure intensifier with power ration 3.8, supplied from a constant flow pump 6 l/min and a pressure p_1 = 80 bar. The consumer is similar to the consumer in Figure 3b.
The evolution of pressures p_1 and p_2 in the case of pressure intensifier powering is shown in Figure 4. In approximately 5s the high pressure circuit is supplied at the maximum pressure p_2.
Because the circuits are not sealed, for maintaining the pressure p_2 it is necessary to continue the supply of the low pressure circuit. In the case that the low pressure circuit supply (p_1, Q_1) is stopped, as shown in Figure 5, the high pressure circuit is discharged.

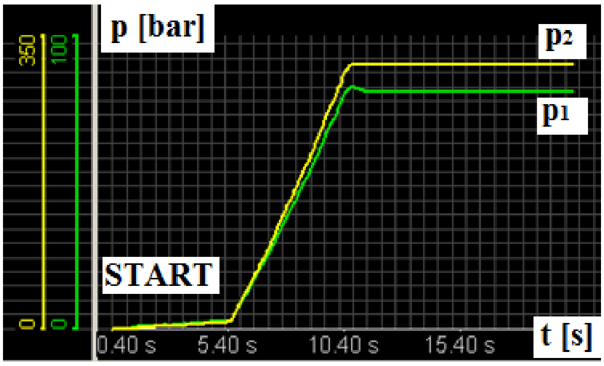


Fig. 4. Evolution of pressures p_1 and p_2

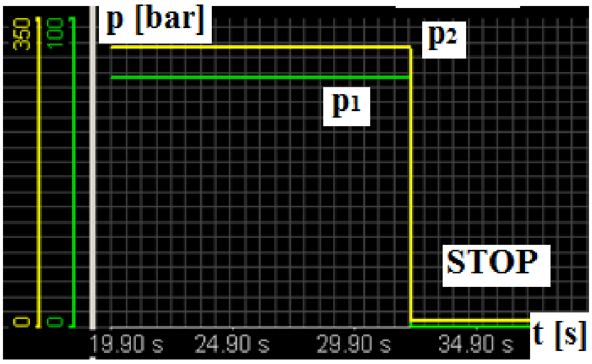


Fig. 5. Discharge of high pressure circuit

In order to avoid such situations but also to prevent the excessive heating of the oil it is recommended to use the pneumatic hydraulic accumulators [5]. Taking into account the pressure limits specific to the accumulators it is recommended to place them in the low pressure circuit.
If such accumulator is used, the evolution of the pressures p_1 and p_2 during the phase of intensifier

powering is shown in Figure 6.
Because the accumulator is charging, one can notice that the time needed to reach the maximum pressure is longer, 10 s approximately. In the case of heavy-duty machine tools, this delay does not affect decisively the auxiliary times. The presence of the accumulator makes possible to maintain the pressure in the high pressure circuit even after the discharge of the low pressure, at the source level, by using the pre-control systems [1].
Figure 7 shows how the pressure p_2 is maintained even if the low pressure circuit is not powered.

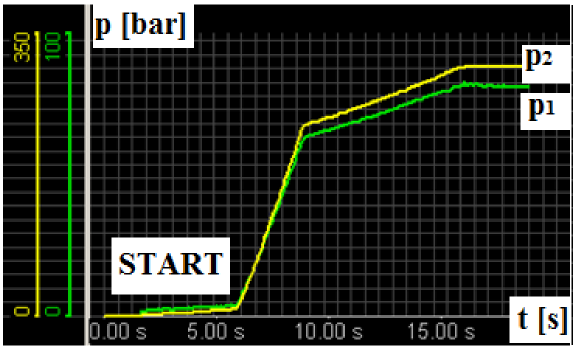


Fig. 6. Evolution of pressures p_1 and p_2 in circuit with accumulator

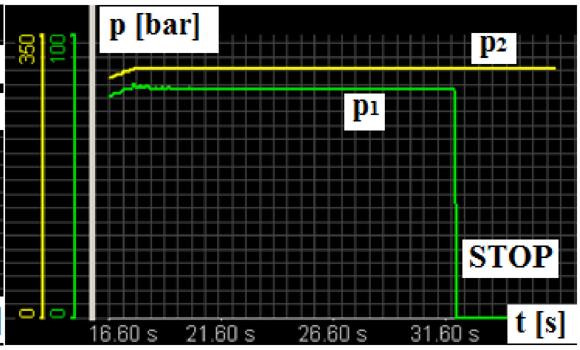


Fig. 7. Keeping the pressure p_2 after discharging the low pressure circuit

The time of pressure p_2 maintaining depends on accumulator size and on the circuit losses as well.
If pressure p_2 must be maintained for a longer time, the circuit will be charged again.
For discharging the circuit, it is necessary to discharge the accumulator low pressure circuit too. This situation is simulated in Figure 8.

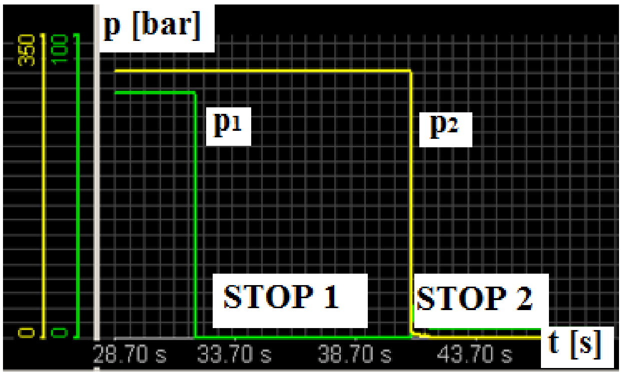


Fig. 8. Discharge of circuit when accumulators are used

In Figure 8, at STOP 1 control, the low pressure circuit is cut off (the pre-control [1] is disconnected); at STOP 2 control, all unit will be discharged, which corresponds to the supply through path B, according to Figure 3a.
The use of simulations in the design phase of hydraulic units allows determining their behavior in dynamic mode.

4. Using the pressure intensifiers in the unloading systems of heavy-duty machine tools guideways

The sliding guideways enable the unloading of large loads, the accurate stopping and locking in a controlled position. The rolling guideways are preferred for the heavy-duty machine tools if long travels must be performed. These guideways are characterized by higher velocities and low resisting forces. The use of combined guideways and unloading systems helps to obtain systems enabling the positioning movements made with high velocity on almost the entire travel, after which a travel with smaller velocity is made on the sliding guideways only, followed by an accurate stop and the eventual axis locking [2, 6].

Figure 9 shows the operating principle of the unloading systems.

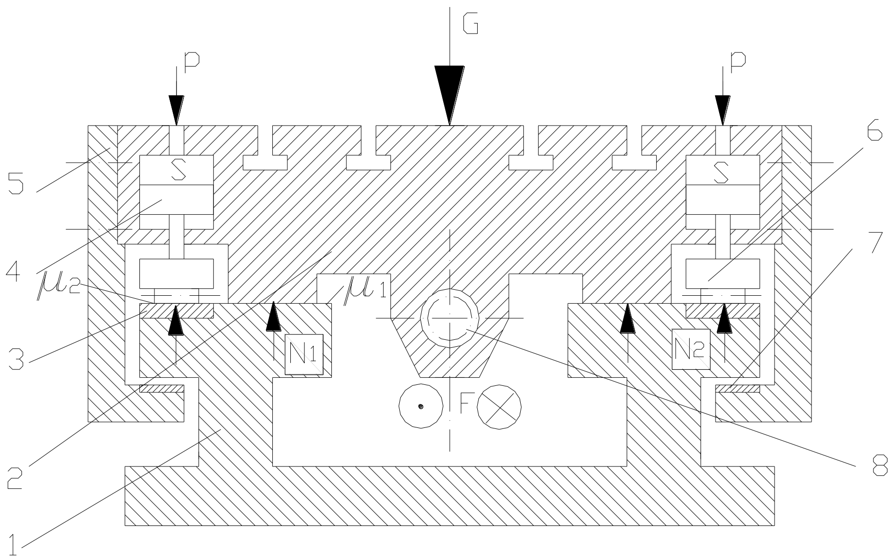


Fig. 9. Hydraulic system for guideways unloading

The saddle 2 with the sliding friction coefficient μ_1 travels on the guideways of the bed 1. Also the harder plates 3 are applied on the bed. Between these plates and the intermediate elements (with rollers) 6 there is the sliding friction coefficient μ_2. In the absence of the supply pressure *p*, the system will operate without unloading. If a pressure *p* is present on the n pistons 4 with the active surfaces *S*, these ones press the elements 6 on the plates 3. The pressure action results in the taking over of a part of the load *G* by the rolling quideways. The guideways will be secured (closed) by means of the closing plates 5. On the lower zone of these ones there is plating made of antifriction material 7. In these two cases the force *F* required at the final component of the feed mechanism 8 will have the expressions:

- Without unloading

$$F = \mu_1 G \tag{7}$$

- With unloading

$$F = \mu_1 N_1 + \mu_2 N_2 \tag{8}$$

In the relation (8) it was noted: N_1 - normal force at the sliding guideways level, N_2 - normal force at the rolling guideways level. These forces have the following expressions:

$$N_1 = G - N_2 \tag{9}$$

$$N_2 = npS \tag{10}$$

The value of pressure *p* is specially established to enable a sufficient unloading at the related kinematic chain but there will be not reached a "total unloading", characterized by the "critical value":

$$p_C = \frac{G}{nS} \tag{11}$$

The dependence of the force (F) developed by the unloading pressure (p) is shown in Figure 10.

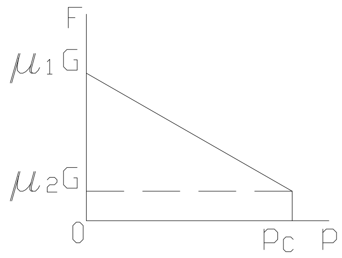
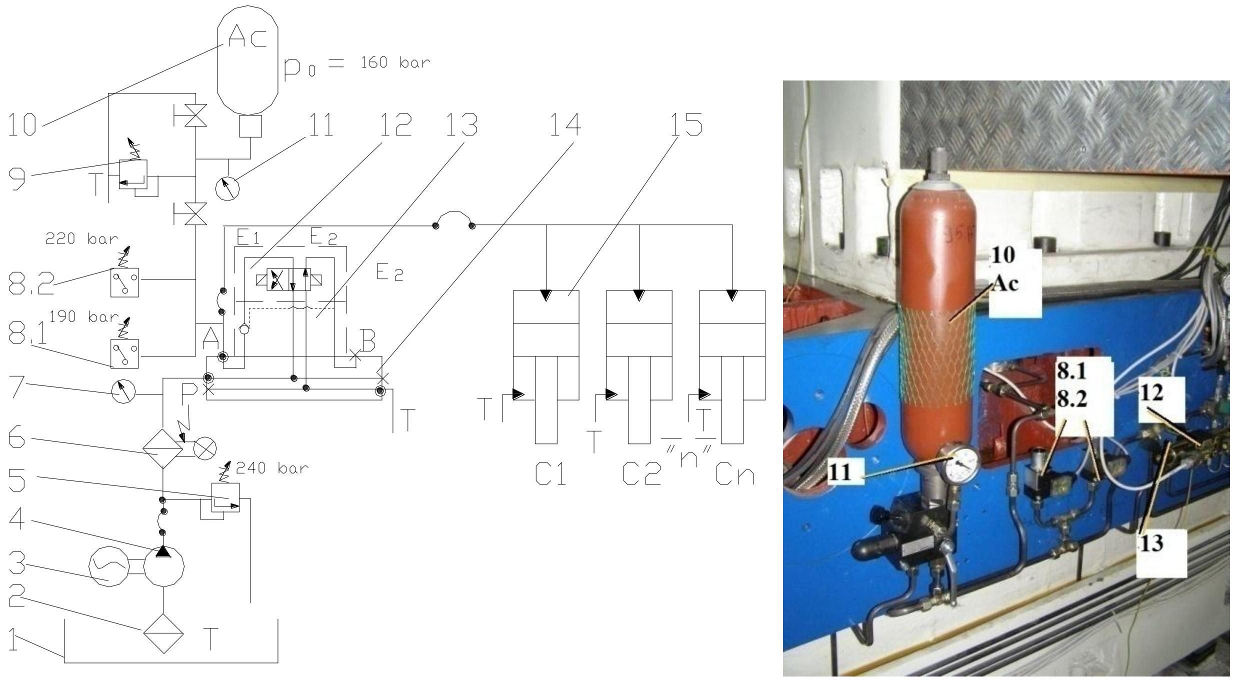


Fig. 10. Dependence of the force required by feed kinematic chain on the unloading pressure

The unloading with a pressure higher than the value p_C leads to the disappearance of the sliding guiding, to unsteadiness and can even overstress the closing elements 3 and 5.
Once determined the value of the operating pressure *p*, this one must be obtained and maintained at the n pistons as long as needed. Usually the pressure is necessary for rapid travels for positioning. These travels are performed in much smaller times that the times required by the machining operations. It is the case of the unloading of X axes guideways in the Gantry type milling machines and the heavy-duty boring and milling machines [1, 6, 7].
The hydraulic units for unloading make pressures of 200 bar usually. Higher values of the pressure entail the use of more expensive components. Figure 11 shows the hydraulic diagram and a part of the hydraulic elements used to unload the guideways of an AFP type machine [1, 6, 7].



a. b.

Fig. 11. Hydraulic system of guideways unloading on X axis

The pump 4 driven by the electric motor 3 sucks the oil from the tank 1 through the suction filter 2. The operating maximum pressure is adjusted by means of the pressure relief valve 5. Then the oil is filtered by means of the filter with clogging indicator 6. The adjusted pressure is viewed on the pressure gauge 7. Plate 14 is supplied on path P. The electric valve 12 and the check valve hydraulically operated 13 are placed on the plate 14. The actuation of the electromagnet E_2 leads to the loading of the guideways unloading circuit. The pressure at the pressure relief valve 5 is ensured in all n unloading cylinders 15. In this moment the pump can be stopped. The pressure is maintained by means of the accumulator 10 [5] within the values adjusted at the pressure switches 8.1 and 8.2. The switch 8.1 controls the eventual restart of the pump, while the switch 8.2 controls

its stopping. All this time the pressure can be read on the pressure gauge 11. The system discharge is performed by actuating the electromagnet E_1 whether the pump operates or not.
The hydraulic unit includes rather many elements and the power consumed by it is about 3 KW. If a higher pressure is required, it is possible to use hydraulic pressure intensifiers. In this case, the hydraulic diagram includes one or more intensifiers, besides the elements mentioned above. These intensifiers allow operating with a lower pressure of the pump for the same cylinders (in this case under 100 bar). Thus the necessary power of the electric motor decreases (~1.5 KW) and the hydraulic elements of the low pressure circuit are less stressed. The new hydraulic diagram is shown in Figure 12. The notations are the same as the ones used in the previous figure; we mention that the electric valve 12 has another diagram and the item 16 is the pressure intensifier in Figure 12 which serves the consumers C, namely the cylinders 15, in this case.

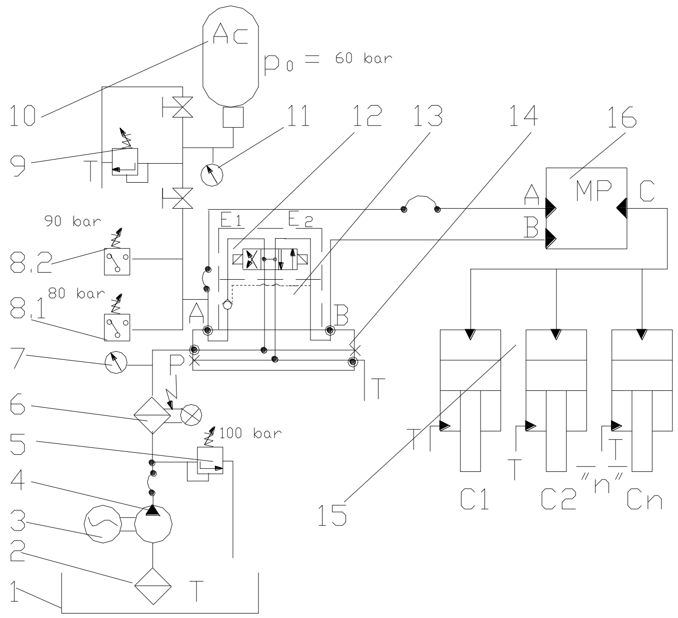


Fig. 12. Hydraulic unit for unloading with pressure intensifier and accumulator

Thanks to the pressure intensifier, the pressure adjusted at the pressure relief valve drops up to 100 bar; in its turn, the accumulator charges at a pressure of 60 bar only. In these conditions the power of motor 3 too decreases by 50%.
In some situations it is possible to use hydro-pneumatic pressure intensifiers. With this type of intensifiers, the pressure p_1 of the primary circuit is obtained from a pneumatic source but usually this one does not exceed the value of 10 bar. The operating principle of these intensifiers is similar [2]. At the same intensification ratio, the maximum pressures obtained are smaller than the pressures obtained by hydraulic intensifiers. We mention some advantages of the hydro-pneumatic intensifiers: small volumes of liquid, compact overall size, no need of pressure hydraulic sources (pumps, devices, tanks etc.).

5. Conclusions

The pressure hydraulic intensifiers are a preferable solution if high clamping/unclamping forces are required and obtained with pressures superior to 300 bar. These intensifiers serve the consumers that need small amounts of liquid. In the case of heavy-duty machine tools, they are recommended for the locking/unlocking systems but also for the unloading systems of the guideways.
As a general rule, the obtained pressures do not exceed 500 bar. These pressures develop only after getting out from the intensifier. Therefore, the devices used in the low pressure floor are not special ones and the hydro-pneumatic accumulators too can be used up to pressures of 250 bar.
For the same consumers and the same pressure (higher than 200 bar), the units that include intensifiers will have electric motors with a smaller power than the units that do not have such

intensifiers. Taking into consideration that the pressure in the low pressure circuit drops even if the tanks volume is diminished, the heating of the unit is reduced if intensifiers are used. The intensifiers entail the increase of the times intended for the performance of the respective functions. Usually, this increase of the auxiliary times in the case of the heavy-duty machine-tools does not interfere with their productivity. For these machines, the auxiliary times, even in the range of minutes, are much smaller than the machining times which are in the range of hours.
The calculation of the systems equipped with intensifiers can be easier if simulations programs are used, making possible the determination of the behavior in static mode and in dynamic mode as well.

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