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Heavy Duty Machine Tools - Specific Pneumatic Drives

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

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

Abstract: *This paper introduces some applications of the pneumatic drives for heavy duty machine tools. Only the machine pneumatic drives proper are presented, not their associated devices too. These applications are used for heavy duty machine tools like centre and vertical lathes, gantry type milling machines, boring and milling machines and horizontal boring and milling machines. The pneumatic systems presented hereby are an independent part of the machine since the manufacturing stage of this one or they have been attached to the machine during its remanufacturing. They are formed of typified elements mostly, supplied by specialized manufacturers.*

Keywords: *Machine tools, pneumatic drives, auxiliary pneumatic chains with pneumatic actuation*

1. Introduction

Almost every factory or workshop is equipped with central compressor stations and compressed air distribution networks. Therefore, it is necessary to design and build modern pneumatic systems of high productivity.
The most important advantages of the pneumatic drives that favored their use in machine tools building also are listed below:

- Achievement of clamping constant forces whose value can be easily controlled during operation;
- Accurate establishment of the developed forces value and the constant maintaining of these ones in order to make possible the machining of semi-finished products with thin and easily deformable walls without the danger of destroying them during the clamping;
- It is easier to maintain and operate the pneumatic drives than the hydraulic systems which are replaced – wherever possible – by the pneumatic drives;
- compatibility with the hydraulic systems within the hydro-pneumatic units, such as in the case of pressure intensifiers;
- possibility to ensure the cooling of the semi-finished products and tools during the cutting operations;
- the motors and devices included in the pneumatic drives are generally normalized components, leading to design labor savings and reduced cost price.

The heavy duty machine tools, by their intended use and construction, are generally equipped with main and feed kinematic chains electromechanically driven [1]. These kinematic chains develop large powers, highly accurate adjustable rotational speeds and translation speeds. In most cases, these requirements can not be met by the pneumatic systems. Consequently they can be found in the systems serving the non-generating kinematic chains [1] usually named auxiliary kinematic chains.

2. Pneumatically driven safety systems for operator and machine

In the case of the heavy duty vertical lathes [1, 2, 3] with table diameter larger than 1400mm, depending on their destination, these ones can be provided with safety covers that separate the work area from the machine outside. It is possible to have two or three covers. Their number is determined depending on the size of the machine and on the way the machine is fed with semi-finished materials. Generally, these covers have pneumatic actuation. They can be actuated independently or not, by manual control or by the machine program. Other specific requirements of these covers are listed below:

- their steady locking in upper position;
- prevention measures against accidental drop if the pressure supply fails;
- if it is the case, synchronous travel with the cross-rail.

Figure 1 presents the pneumatic diagram of the system that runs the two covers of a vertical lathe SC17 type.

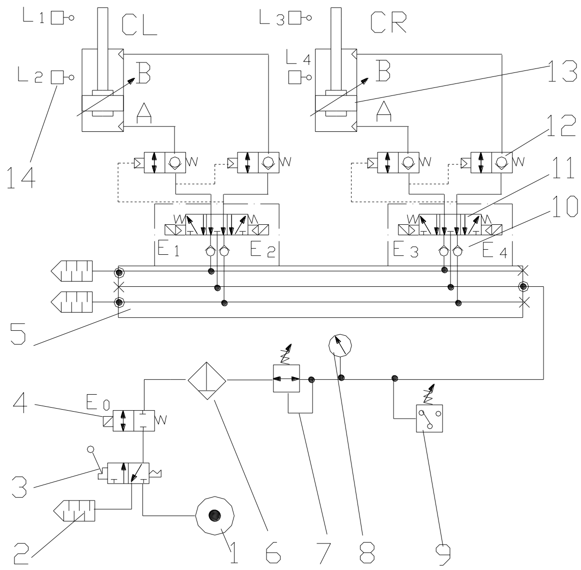


Fig. 1. Pneumatic diagram for the actuation of the covers of the vertical lathe SC17 type

The unit is connected to the pneumatic source 1 by actuating the manual selector 3. The electric control is started by means of the electric valve 4 (E_0 powered). The compressed air is filtered and enriched with oil drops by the preparation group 6. The operating pressure (maximum 5 bar) is regulated by means of the pressure regulator 7. The value of the operating pressure is read on the manometer 8 while the pressure switch 9 confirms the presence of pressure in the unit. The check valves 10 and the electric valves 11 are assembled on the plate 5. These valves supply the two cylinders (CL- left side, CR –right side) 13 through the agency of the pneumatic pilot valve 12. The positions of the two cylinders are confirmed by the position limit stops L1-L4, 14. The noise during operation is reduced by means of the dampers 2 [4, 5, 6]. By actuating the electromagnets E_1-E_4 it is possible to move the cylinders CL and CR upwards and downwards, simultaneously or not. The commanded positions are confirmed by the limit stops L_1-L_4. If the electric valves E1-E4 did not receive any commands, the valves 12 with pneumatic actuation will lock the cylinders so that the accidental fall of the covers will be prevented. The check valves 10 ensure the independent operation of the two cylinders by impeding the re-entry of the eliminated air into the circuit. The cylinders are provided with braking systems at stroke end in both directions of the stroke.
Figure 2 shows a part of the pneumatic unit [5].

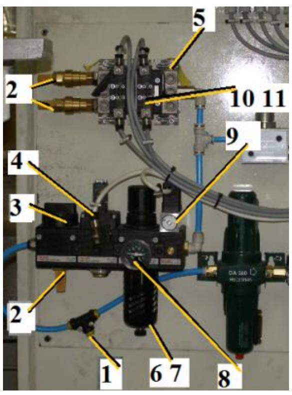


Fig. 2. Part of the pneumatic unit

In the manufactured unit, the pressurization system of the scales on X and Z axes of the machine is also supplied from the pressure intake port 1; this system is not represented in the diagram shown in Figure 1. The notations are the same as in Figure 1.
In Figure 3 is shown the pneumatic diagram for the actuation of the covers of the lathe SC33 CNC type. There are three covers for the bed, actuated by the cylinders CL, CC and CR 13 but also a 90º rotary cover at the disk type tool magazine. This cover is driven by the pneumatic oscillating motor 17 [5, 6].

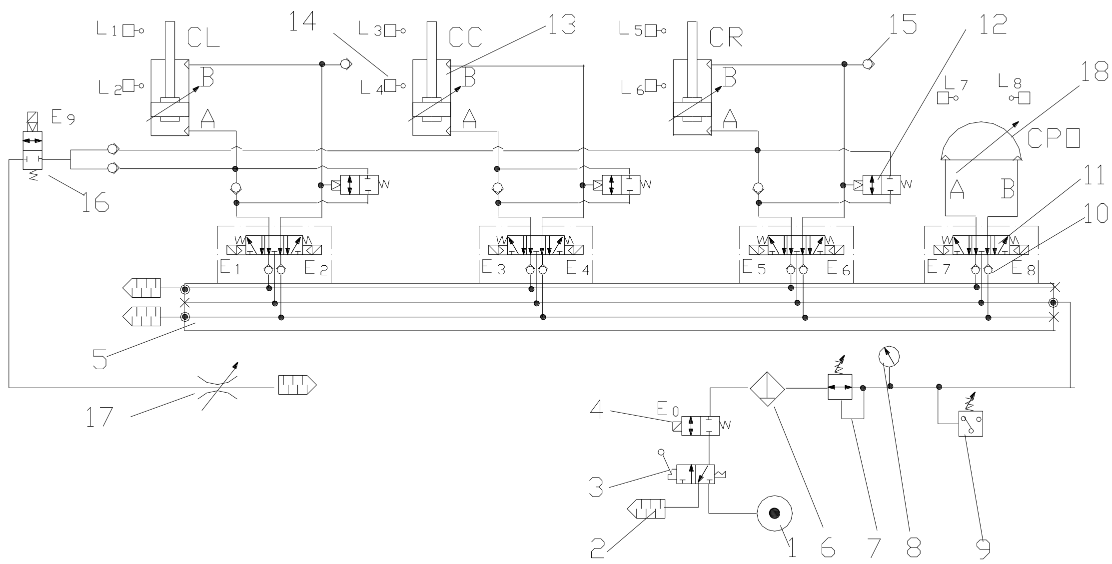


Fig. 3. Pneumatic diagram for the actuation of the covers of a vertical lathe SC33CNC type

The elements numbered from 1 to 14 in Figure 3 have the same significance as the ones in Figure 1, with the remarks below:
- valves 12 have another diagram,
- there are 13 check valves 10,
- there are three lifting cylinders: one on the left CL, one in the centre CC and one on the right CR.

Some additional elements are shown in Figure 3: 15 – check valves to avoid the depressurization, 16-electric valve for exhausting the air from the cylinders CL and CR if the left side and right side covers are pushed downwards by the cross-rail; in this case, the cover driven by the central cylinder CC is fully lifted, 17 – throttle valve for the adjustment of the speed of descent together with the cross-rail, 18- oscillating pneumatic motor with braking at stroke ends. The operation of the unit is similar to the one of the SC17 lathe. The essential difference is given by the fact that if the cross-rail of the lathe goes downwards for positioning or as work axis, the covers actuated by the left side cylinder CL or by the right side cylinder CR are pushed downwards by the cross-rail while the central cover keeps its position. When the cross-rail went downwards until the upper level of the side covers, the electromagnet E_9 of the electric valve 16 will be actuated. The cross-rail presses the cylinders CL and CR eliminating the air from the chambers A of these cylinders through the throttle 17. The chambers B of the cylinders suck the atmospheric air through the valves 15. The electric valve 16 is not actuated for the movements upward and downward of the covers. The tool magazine with 12 stations is located on the right side of the cross rail and is protected by a tiltable door driven by the rotary pneumatic motor (CPO) 18. This motor is actuated by the electromagnets E_7 and E_8. The positions of closed / open cover are confirmed by the limit stops L_7 and L_8.
The supply point on the plate 5 can also feed other pneumatic consumers of the machine, such as the tool blowing system, the clamping systems etc. In this case, the unit must meet the following conditions:
- the pressure source must cope with the maximum consumption;
- consumers must operate at a single pressure adjusted by means of regulator 7;

- the size of the construction (the DN) must be properly dimensioned.

3. Hydro-pneumatic pressure intensifiers used in machine tools

The pressure intensifiers are compact constructions which enable the transformation and intensification of a pneumatically developed pressure into a hydraulic pressure. This pressure can be sent to a consumer under the form of oil flow.
Figure 4 presents the operation diagram of a hydro-pneumatic pressure intensifier manufactured by specialized companies [5, 7].

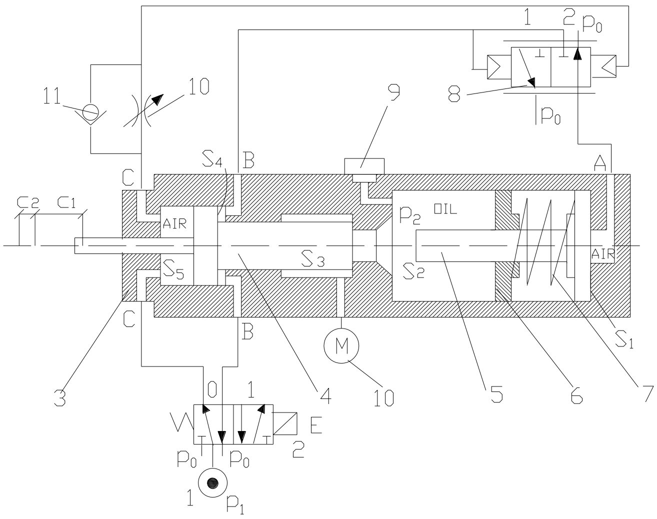


Fig. 4. Hydro-pneumatic intensifier

The system is supplied with compressed air at the maximum pressure p_1 = 2÷10 bar on path 1. If the electric valve 2 is not actuated (E is not supplied), the air presses the surface S_5 of the piston 4 which is moved to the right in body 3. The piston 5 too is moved to the right in the same time with both piston 6 and spring 7 that moves to the right as well. The left side chamber of piston 6 separates the oil in body 3 from the path A of body 3.This chamber was initially filled with oil by removing the plug 9.The supply pressure on path C of the body 3, through the check valve 11, actuates the pilot 8 by passing it to the position 2. The rod of piston 4 moved on the total travel C_1+C_2. All this time, the chamber B of body 3 is maintained at the atmospheric pressure p_0.
If the electromagnet E is powered, the electric valve 2 switches from position 0 to position 1. In this case, the chamber B of the body 3 will be supplied. The piston 4 will move to the left along the stroke C_1, where it opposes a resisting force. This one entails the increase of the pressure in chamber B up to the maximum value p_1. This pressure changes the status of the pilot 8 which will be switched to status 1. In this case, the pressure p_1 will act on path A and the piston 7 on the surface S_1, moving to the left. The surface S_2 of this one presses the oil and achieves the pressure p_2. This one drives the surface S_3 of the piston on the whole remaining stroke C2. Meanwhile, the pressure p_2 developed in the chamber with oil can be viewed on the pressure gauge 10. If the electric valve 2 is no more actuated, the entire travel to the right will be performed. The piston is returned to the initial position by the spring 7. The command air is evacuated from the right side chamber of the valve 8 in a supervised manner, by means of the throttle valve 10. If the frictions and the possible losses are neglected, the maximum pressure developed in the pneumatic chamber has the value:

$$p_2 = p_1 \frac{S_1}{S_2} \quad (1)$$

The maximum forces developed at the rod of cylinder 4 along the two strokes C1 and C2 are shown below:

$$F_1 = p_1 S_4 \quad (2)$$

$$F_2 = p_2 S_3 = p_1 \frac{S_1}{S_2} S_3 \quad (3)$$

The system presented hereby includes the pressure intensifier and the power cylinder joined in a single construction. But there are also some hydro-pneumatic pressure intensifiers that can serve several cylinders [7]. Figure 5 shows how the four cylinders are supplied. In their case, the stroke C1 and the return along the total stroke C_1+C_2 are achieved pneumatically. The power stroke C_2 is made hydraulically at an intensified pressure.

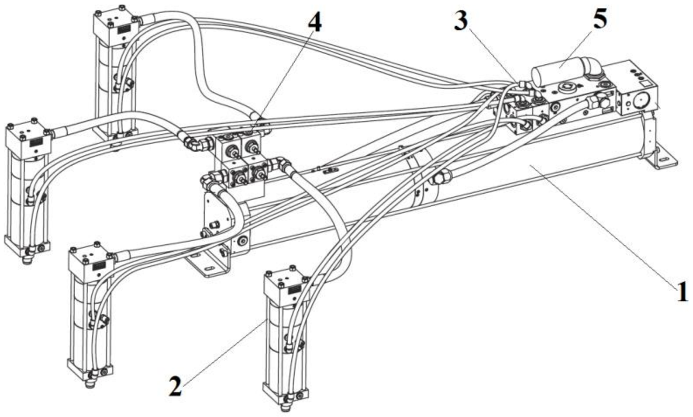


Fig. 5. Hydro-pneumatic intensifier that serves four cylinders

The hydro-pneumatic intensifier supplies the cylinders 2 through the valves 3 during the phases of approach (C_1) and retreat (C_1+C_2). During the phases that require high forces, stroke (C_2), the cylinders are hydraulically supplied by means of the valves 4. The silent operation is ensured by the noise suppressor 5. The hydro-pneumatic pressure intensifiers can provide a pneumatic pressure p_1 = 2÷6 bar and hydraulic pressures up to p_2 = 400 bar.
In the field of heavy duty machine tools, these intensifiers are used as systems for unloading the guideways [1, 8].
The sliding guideways enable the unloading of big loads, the accurate stop and locking on a commanded position. As for the heavy duty machine tools, the rolling guideways are preferred for the achievement of long travels. The rolling guideways are characterized by higher speeds and lower resisting forces. By using combined guideways and unloading systems, we can obtain systems that make possible positioning movements at high velocity on almost the entire travel; afterwards, the travel is made at lower velocity, on the sliding guideways only, followed by accurate stop and possible locking of axis [1, 2].
Figure 6 shows the operating principle of the unloading systems [8].

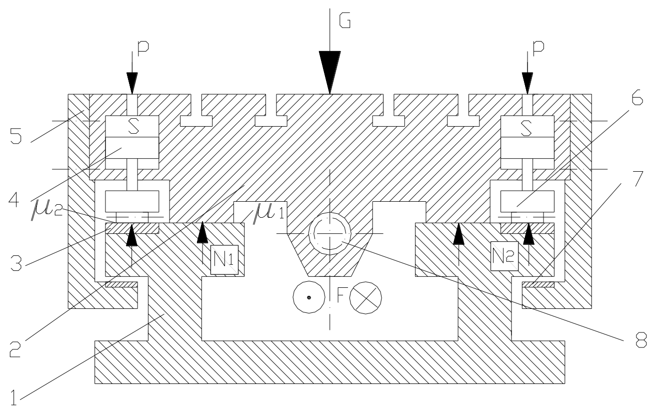


Fig. 6. Hydraulic system for guideways unloading

The saddle 2 travels on the guideways of the bed 1, with the frictional coefficient μ_1. The tougher plates 3 too are located on the bed. The intermediate elements 6 (with rollers) are introduced between these plates. The rolling frictional coefficient μ_2 occurs between the rollers and the plates 3. If the supply pressure p is missing, the system will operate without unloading. If there is a pressure p in the n pistons 4, with the active surfaces S, these ones press the elements 6 on the plates 3. The effect of the pressure action is the taking over of a part of the load G by the rolling guideways. The guideways are secured (closed) by the closing plates 5. On the lower part of these plates there are insertions of anti-friction material 7. In both cases, the force F required at the level of the final element of the feed device 8 will have the expressions as follows:
- without unloading

$$F = \mu_1 G \tag{4}$$

- with unloading

$$F = \mu_1 N_1 + \mu_2 N_2 \tag{5}$$

The following notations were used in the relation (5): N_1 - normal force at the sliding guideways level, N_2 - normal force at the rolling guideways level. These forces have the expressions:

$$N_1 = G - N_2 \tag{6}$$

$$N_2 = npS \tag{7}$$

The value of pressure p is so adjusted as to ensure a sufficient unloading at the associated kinematic chain but to not reach the total unloading, characterized by the critical value:

$$p_C = \frac{G}{nS} \tag{8}$$

Figure 7 presents the dependence of the force (F) developed by the unloading pressure (p).

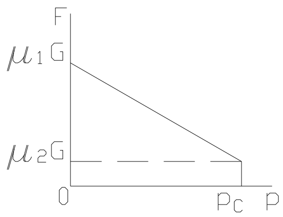
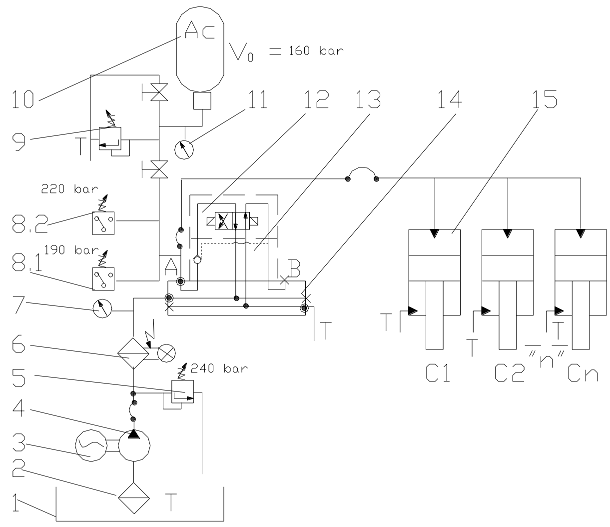
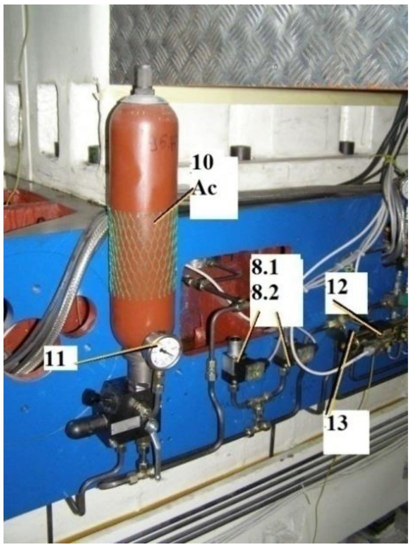


Fig. 7. Dependence of the force required by the unloading pressure in the kinematic chain

The unloading with a pressure higher than the value p_C entails the loss of the sliding guidance, the instability and even the overstress of the closing elements 3 and 5.
Once settled the value of the operating pressure p, this one must be ensured and maintained in the n pistons as long as necessary. Usually, the pressure is necessary for the fast travels made for positioning. These travels are performed during much smaller times than the ones required by the machining operations. It is the case of the unloading of the X axes guideways in the gantry type milling machines and the heavy duty boring and milling machines [1].
The hydraulic unloading can be performed in three manners:
a- with hydraulic units specially built;
b- with hydraulic pressure intensifiers [8];
c- with hydro-pneumatic intensifiers.
Generally, the hydraulic units intended for the unloading provide pressures of 250 bar. Higher values of the pressure involve the use of more expensive components. Figure 8 shows the hydraulic diagram and some of the hydraulic elements used to unload the guideways of a horizontal boring and milling machine.





a. b.

Fig. 8. Hydraulic system for the unloading of X axis guideways

The pump 4 actuated by the electric motor 3 sucks the oil from the tank 1 through the suction filter 2. The maximum operating pressure is adjusted by means of the pressure relief valve 5. Afterwards the oil is filtered by means of the filter with clogging indicator 6. The adjusted pressure is read on the pressure gauge 7. The plate 14 is supplied on the path P. The electric valve 12 and the check valve hydraulically operated 13 are located on the plate 14. The actuation of the electromagnet E_2 leads to the charge of the circuit for guideways unloading. The pressure adjusted at the pressure relief valve 5 is ensured for all the n unloading cylinders 15. At this moment the pump can be stopped. The accumulator helps to maintain the pressure between the values adjusted at the pressure switches 8.1 and 8.2. The pressure switch 8.1 commands the eventual re-start of the pump, while the pressure switch 8.2 will command the stop of the pump. All this time, the pressure can be viewed on the pressure gauge 11. Regardless of whether the pump is running or not, the system will be discharged by actuating the electromagnet E_1.

The hydraulic unit includes quite many elements and the power consumed by these ones is about 3KW. If a higher pressure is needed, hydraulic pressure intensifiers [4, 6, 7] shall be used. In this case, the hydraulic diagram includes one or several intensifiers besides the elements used above.

The use of the hydro-pneumatic intensifiers simplifies very much the construction. The pump, the electric motor and many components are eliminated. In this case, if a maximum number of 6 cylinders is needed, it is possible to use a unit similar to the one shown in Figure 4. The ordinary hydraulic diagrams use elements that run at pressures of 320 bar at the most. In case of higher pressures, the coupling elements (pipes, hoses, fittings) are special, much more expensive than the usual ones. Another advantage of the hydro-pneumatic pressure intensifiers is the fact that the oil tank, the filtering unit etc. are no more needed.

4. Balancing pneumatic systems

The balancing systems are used for the feed kinematic chains that operate vertically, in order to reduce the necessary power and to avoid the oversizing of the basic components (electric motor, reducer, screw and ball nut mechanism). These systems operate in parallel with the feed kinematic chain but take over a part of the weight of the live elements G [1, 9]. Depending on how they perform this function, the balancing kinematic chains can be mechanical (with counterweight), hydraulic or pneumatic ones.

The mechanical ones have a simpler structure but they involve the increase of the machine mass, require specific safety systems and they diminish the performances of the kinematic chain, especially in transitory regime. The hydraulic systems are efficient, do not increase the mass of the machine, ensure a proper dynamic behavior of the feed dynamic chain but they also involve the

presence of a complex hydraulic unit with generally expensive hydraulic elements. Another disadvantage is the need of large oil tanks (required by the high installed power) and, in some cases, cooling systems for the oil.
The pneumatic balancing can be used for smaller travels than the ones of the hydraulic systems; it also can be used when it is possible to balance the movable weight by means of one or two cylinders of acceptable size, for operating pressures of 5-6 bar.
Figure 9 shows the balancing diagram for the cover of a milling machining centre.

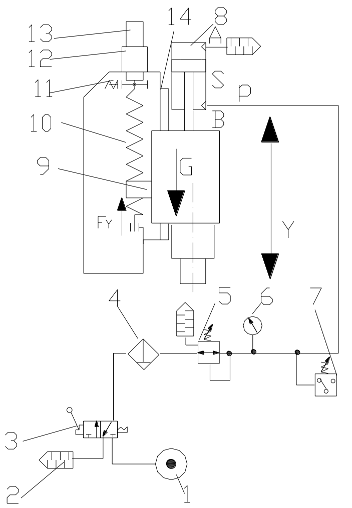


Fig. 9. Balancing unit for the cover of a milling machining centre

In order to move the cover 9 on the guidewats 14 of Y axis, the feed/positioning kinematic chain is formed of the following elements: electric motor 13, reducer 12 and ball screw 10. On the ball screw is placed an electromagnetic brake 11 that prevents the case from falling down if an electrical failure occurs. Cover 9 is permanently supported by the pneumatic cylinder 8 supplied with air under pressure on the surface S. The necessary air is taken over from the source 1 and, after the actuation of the manual valve 3, is passed through the filter 4. The pressure required for balancing is adjusted by means of the pressure regulator 5. The adjusted pressure is read by means of the pressure gauge 6. The pressure switch must confirm the presence of the pressure p and the brake 11 must be unlocked in order to enable the operation of the feed kinematic chain. To avoid the depressurization of the chamber A of the cylinder, a noise suppressor 2, identical with the one on the return of the valve 3, was assembled.
When going upwards, in stationary regime, the following relation can be considered:

$$G + F_F + F_A = F_Y + F_E \tag{9}$$

The relation (9) used the following notations: G - weight of the cover, F_F - sum of the friction forces in the system, F_A - cutting force (for feed travel only), F_Y - force developed by the ball screw, p - pressure adjusted at the pressure regulator 5, F_E - force developed by the balancing system. In this case $F_E = p \cdot S$. If we consider that the balancing of the weight G is totally made, we can notice that the force developed by the feed kinematic chain will compensate the friction forces and possibly the cutting ones in the direction Y.
In dynamic regime, during the phase of rapid going upwards for positioning, the following relation can be taken into consideration:

$$M\frac{dv}{dt} + bv + G + F_F = F_E + \frac{2\pi T}{ip_{BS}} \tag{10}$$

In the relation (10) it has been also noted: M - mass of the mobile assembly(cover), v - instantaneous velocity along Y axis, t - time, b - linearization coefficient of force losses proportional to velocity (damping constant), T - torque at the drive motor, i - transfer ratio of the reducer 12

(subunit or =1), p_{BS} - pitch of the leading screw 10. The relations (9) and (10) show to which extent the presence of the balancing system diminishes the value of the torque T at the drive motor level.
The role played by the balancing is highlighted by the comparison of the variants of actuation with and without balancing on the occasion of the remanufacturing of a milling machining centre with vertical shaft. The drive motor of the feed kinematic chain on Y axis has the nominal torque T_{NEM} = 20 Nm and the nominal rotational speed n_{NEM} = 3000 RPM. The leading screw has the pitch p_{BS} = 10 mm and the reducer has the transfer ratio i = 1.The values shown in the graph of Figure 10 are obtained for the torques required at the motor without balancing T_{NCB} and the motor with balancing T_{CB} for the mass of cover M = 750 kg, maximum imposed velocity v_{Max} = 15 m/min and the acceleration time t_{AC} = 0.1 s.

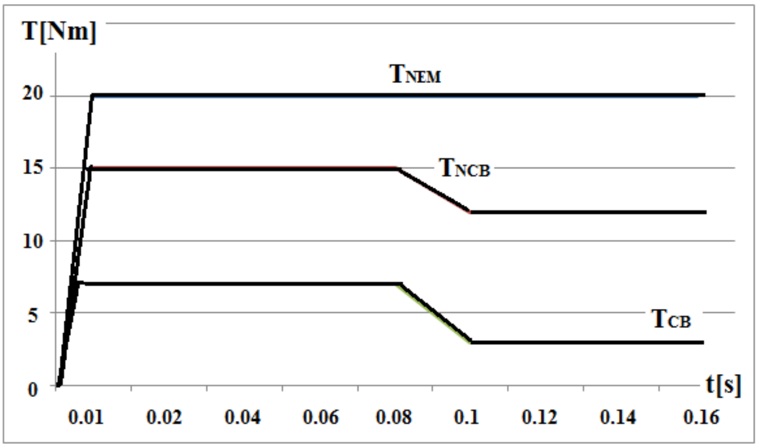


Fig. 10. Torque necessary for the Y axis drive motor with and without balancing

The calculations indicate in both cases an acceleration of a = 2.5 m/s^2 necessary for reaching the maximum imposed velocity. In both cases, the developed torque has acceptable values under the maximum value. One can notice that the feed kinematic chain is less stressed in the presence of balancing. The balancing system performs the unloading of the kinematic chain permanently, even if the electric actuation is stopped.
During operation, the real charge, measured as percentage of electric current, has values in the range of 42÷78% without balancing and in the range of 27÷62% with pneumatic balancing, values close to the ones calculated and shown in Figure 10. Figure 11 shows two details of the balancing pneumatic system [5, 7, 10].

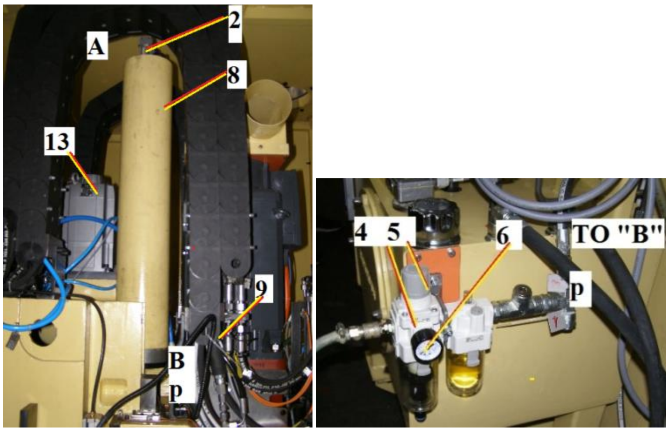


Fig. 11. Details of the balancing pneumatic system

The notations are the same as the ones used in Figure 9. Pneumatic cylinder 8 has the diameter of the piston D = 100 mm and the diameter of the rod d = 16 mm. Its active stroke is c = 500 mm.

5. Conclusions

As a general rule, the machine tools are equipped with a large range of drives of all types: electromechanical, hydraulic and pneumatic ones. At the present moment, most of the generating kinematic chains (main chains and feed ones) are electromechanically actuated. The hydraulic and pneumatic drives, with rare exceptions, can be used in the construction of the auxiliary kinematic chains or of some systems associated to the machine tool.

The pneumatic drives, compared to the hydraulic ones, have a series of advantages: simpler sources of energy (compressed air network); low operating pressure (usually 5÷10 bar) related to the pressure in the hydraulic systems (60÷200 bar); they can achieve higher translational and rotational speeds than the ones obtained hydraulically; easier to maintain; cleaner.

Some of the disadvantages of the pneumatic drives are listed below: their rigidity is inferior to the rigidity of the hydraulic drives because of air compressibility; at similar overall size, they develop forces(torque) inferior to the ones developed hydraulically; in the absence of a local source of compressed air, it is necessary to purchase compressors.

Given these conditions, the pneumatic drives are used in many types of machine tools where they perform some functions as follows: tools cooling, classic one or by means of VORTEX systems; pressurization of pressure transducers in order to protect them; actuation of some clamping systems, feeding with tools and semi-finished material in the case of small and average size machines.

The elements listed below are specific to the pneumatic drives for heavy duty machine tools: actuation of the safety systems; use of hydro-pneumatic pressure intensifiers; actuation of the balancing systems for the masses moved by the feed kinematic chains vertically.

The pneumatic drive of the safety systems is used in most of the modern CNC heavy duty machine tools: lathes, machining centers, boring and milling machines etc. The pneumatically actuated covers can operate horizontally or vertically; they are provided with safety systems against the loss of the commanded position; their velocity is high but does not affect the auxiliary times.

The hydro-pneumatic pressure intensifiers can be the optimal solution for the unloading of the guideways for horizontal feed kinematic chains used in the heavy duty machine tools like gantry type machines and floor type HBM-s. They are simpler and easier to actuate than the hydraulic ones that require a high pressure source, usually over 200 bar.

The pneumatic balancing of the feed kinematic chains that operate vertically is a simple solution to be preferred instead of the hydraulic balancing. Compared to the hydraulic balancing, the pneumatic one deals with shorter strokes (under 1000 mm) and is generally applied to much smaller masses because of the pressure of max 10 bar and the air compressibility at this pressure.

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