

Pneumatic device of the preload and dynamic loads balancing to reduce the intensity of thermal processes in the metal cutting process

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Abstract. Improved reliability of the technological system «machine-tool-instrument-detail» is an important current task. Backlashes and insufficient stiffness of technological system lead to intensive wear of the cutting tool, increasing the heat in the cutting zone. Due to high temperature in the thin surface layers of the workpiece and tool thermal processes may occur which are similar to release and can cause the structural changes of the material. The current article presents the final design of the device which has been developed to reduce the intensity of thermal processes in metal cutting.

1. Introduction

Improving the quality of metalworking and reliability of Technological System «Machine-Tool-Instrument-Detail» (TS MTID) is always an important current task. There are some methods to solve this problem: to change the machining process mode (feed, spindle rate speed, depth of cut), installation of special devices to enhance the stiffness, vibration level reducing [1–5] etc. For example, due to the stiffness enhancing the durability of technological system elements significantly increases.

Insufficient stiffness of process system parts leads to overheating in the cutting zone which impacts directly on the resistance of the tool to weariness, the quality of process material and the cutting efficiency.

Impact of pressure and high temperature causes thermal processes (chemical reaction between the tool and workpiece materials), that changes the chemical composition and physical and chemical properties of the tool surface and the workpiece.

It is known that stiffness of the technological system is almost always the dominating factor determining precision at all types of mechanical machine processing which do not involve implementation of size or profile cutting tool [6].

One of the most effective measures to improve the stiffness of the machine-tool, is the elimination of backlashes [7]. The main method of elimination of backlashes is to provide a preload, which is included while designing the technological system by using the particular units design or using special mechanisms for the tension force generation F_t . In both the first and second cases, tension force F_t



results in a shift in ratio of deformation to force $\frac{\Delta_1}{\Delta F_1}$ downward (Figure 1), so that when the technological system operates, two zones of stiffness can be marked:

- «low» – without F_t ;
- «high» – with F_t .

In the area of «small» stiffness a backlash in the joints influence the technological system operation which downgrades the parts processing. There is no backlash in the area of «high» stiffness. In such a case, significant increase of precision and quality of the parts processing is attained as well as the increase of time for life of instrument operation and the reliability of the technological system operation [8-11].

2. Design of the pneumatic device

There known devices for tension implementation by load suspension. But this design increases the dynamic loads in the technological system by increasing the system inertia. There are solutions to create a preload by means of pneumatic springs however, due to the limited spring stroke and the inability to change the power characteristics of these pneumatic springs when required, but the implementation of such springs is partly neutral [12, 13].

The developed pneumatic device with a simple design and reliability for operation increases the stiffness of any TS MTID when these shortcomings are absent as well as dampens occurring vibrations.

Figure 2 shows the basic diagram of a pneumatic device.

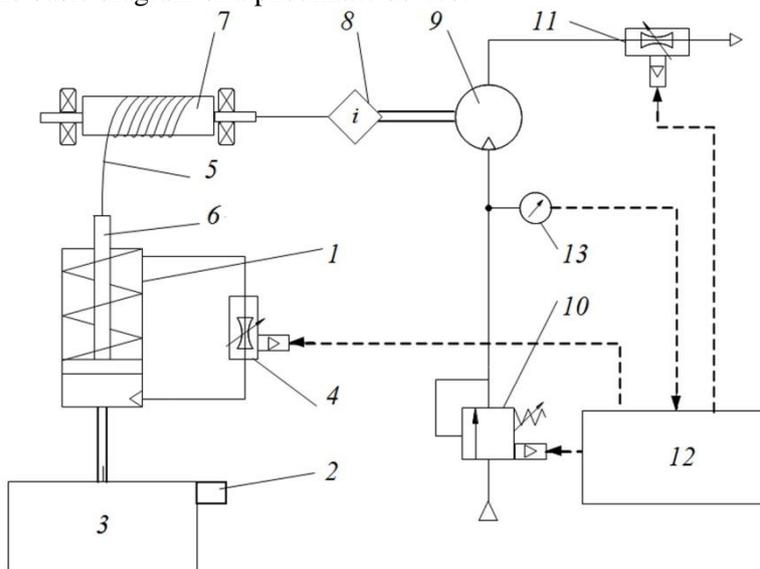


Figure 2. Structure diagram of the preload device and balancing of the dynamic loads in the drive unit of the movable part of the machine-tool.

Pneumatic device for a movable unit of the machine-tool contains a damper I , build in a form of a single stock pneumatic cylinder. There is a return spring fitted in the hydro-cylinder end. Vibration

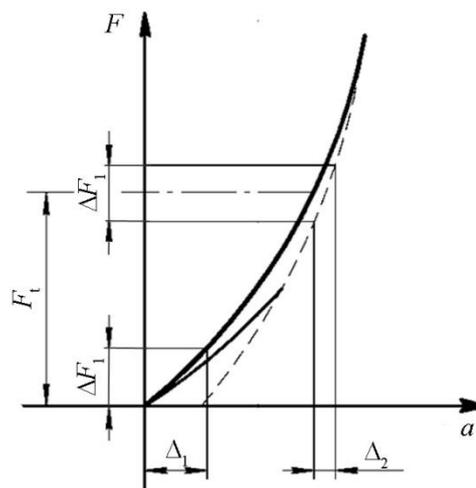


Figure 1. Graph of dependency between the applied force and deformation TS.

sensor 2 is fixed on the piston end of the hydro-cylinder which is jointed to a movable unit of the machine-tool 3, for example with a table.

Stock end and head end of the pneumatic cylinder are interconnected by tubes by means of controlled throttle 4. One end of the wire 5 is fixed to the rod of the pneumatic cylinder 6. The other end of the wire is coiled on the wheel 7, and fixed thereon.

The wheel 7 is fixed to a stationary part of the machine-tool and through a planetary gear reducer 8 is connected to the drive shaft of the pneumatic motor 9 which is connected with a pressure source, such as a pump by the tube through a reducing valve 10. There is a pneumatic accumulator (on the structural diagram are not shown) before reducing valve 10. The regulated throttle 11 connecting to the pneumatic system is mounted on the outlet of pneumatic motor 9. Controlled throttle 4 and the vibration sensor 2 are connected to the numeric control device 12.

Gravity force G and the cutting force F_c are in charge when machining the details which carry out the movement of the movable machine units (screw gear, the gear of rack bar - tooth gear, etc.):

$$F_c = F_{\text{const}} \pm dF \approx F_{\text{const}} \pm \sum_{i=1}^n (dF_i \cdot \sin \omega_i t + \varphi_i),$$

where F_{const} – constant cutting force component;

$\pm dF$ – variable cutting force component;

$dF_i, \omega_i, \varphi_i$ – amplitude, frequency, phase of a i harmonic.

The component force $\pm dF$ is caused by variable cross section of cut or intermittent nature of the cutting process (milling). And when contour machining on the numeric control device there is a significant dynamic force dF_{in} related to the presence of the inertial mass of the movable unit – actuator of machine tool:

$$dF_{\text{in}} = m_a \frac{d^2 z}{dt^2},$$

where m_a – movable unit mass, H;

$\frac{d^2 z}{dt^2}$ – movable unit acceleration, m/s^2 .

These variable components of the forces have a negative impact on the accuracy of machining.

3. Calculation method and operating principle of the device

Pneumatic device operates as follows. When operating, these forces act on the damper body 1. The effect of the variable component of the forces $\pm dF$ gives rise to a pulsing air flow dQ between the stock end and head end of the damper 1 cavities, which passes through a controlled throttle 4 generates the damping force dF_d

$$dF_d = S \frac{dQ^2}{K_{\text{th}}^2} S_{\text{th}}^2,$$

where S – stock area of damper, m^2 ;

dQ – pulsating gas flow rate, m^3/s ;

K_{th} – throttle ratio (determined by the design), $\text{m}^3/\text{N}\cdot\text{s}$;

S_{th} – through area of controlled throttle, m^2 .

It's necessary to do the ratio for optimal damping:

$$\frac{C_d}{K_d} = f_{\text{nf}} = \sqrt{\frac{C_a}{m_a}}, \quad (1)$$

where f_{nf} – the natural frequency oscillations of the movable unit of the machine, Hz;

C_a – stiffness of the movable unit of the machine, N/m;

m_a – mass of the movable unit of the machine, kg.

C_d – damper stiffness, N/m:

$$C_d = \frac{E_{rm} \cdot S_{pr}^2}{W_C} + C_s,$$

where E_{rm} – reduced modulus of the bulk elasticity for damper cavities, N/m²;

W_C – the total volume of air under pressure, m³;

C_s – the stiffness of the return spring damper, Nm;

K_d – damping coefficient, Ns/m:

$$K_d = \frac{dF_d}{V_{pr}},$$

V_{pr} – the rate of movement of the damper piston relatively to its body, m/s.

The equation below is based on the assumption that non-extensible rope (or other rod) or it's reduced value of deformation in comparison with the deformation and backlashes of the movable machine parts.

$$V_{pr} = V_a \text{ and } a_{pr} = a_a,$$

where V_a – conveying speed;

a_a, a_{pr} – accelerations of the movable operating element and the damper piston.

When the condition (1) is met the pneumatic device will perform as a high-frequencies filter, which leads to a significant reduction of the dynamic and impact forces on the movable machine unit by damping. Rotational moment on the wheel 7 is generated by pneumatic motor 9. The required force F_r on the wire 5, is:

$$F_r = \frac{q}{2\pi} (p_0 - p_{th}),$$

where q – the work volume of pneumatic motors 9, m³/v

p_0 – the pressure on pneumatic accumulator, H/m²;

p_{th} – pressure before the adjustable throttle 11, H/m²

The generated force is greater than the weight of the movable unit of the machine G

$$F_r > G = m_a \cdot g, \tag{2}$$

where g –gravitational acceleration m/s²

When the condition (2) is met the force F_d on the wire 5 is directed coincidentally the same way as the cutting force F_c . This will create a permanent tension Δ of the joints of movable unit of the machine-tool as well as compression on the magnitude

$$\Delta = \frac{F_c + F_r}{C_a},$$

thus, neutralizing the backlashes magnitude in mechanical transmission which perform the movable units of machine-tool.

A certain pressure in the stock end of the hydraulic cylinder of damper 1 and the compression of the return spring is attained. This provides the normal operation of the damper 1 at a large stroke magnitude and increases the stiffness of elements of the movable unit of the machine-tool. The control valve 10 and throttle 11 are used to calibrate the required torque and rotational speed of the pneumatic motor shaft 9.

The digital controller 12 controls the value f_{th} of the controlled throttle 4 area, depending on the movement and vibrations rates level occurring in the mobile unit of machine-tool and measured by the vibration sensor 2. The digital controller passes control signals to the controlled throttle 4 of the damper 1.

The differential equation that describes the operation of the pneumatic damper, is the follows:

$$m_a a_a + \alpha V_a + C_d x_a = F, \tag{3}$$

Where the variation low F takes values from:

$$F = \begin{cases} F_c, & \text{in the zone of low stiffness} \\ F_t + F_c, & \text{in the zone of higt stiffness} \end{cases}$$

The mathematical model (simulation) made it possible to conduct theoretical study on the pneumatic device operation.

Structural diagram of the pneumatic device by which calculations were carried out in the Lab View National instruments software product, is shown below (Figure 3).

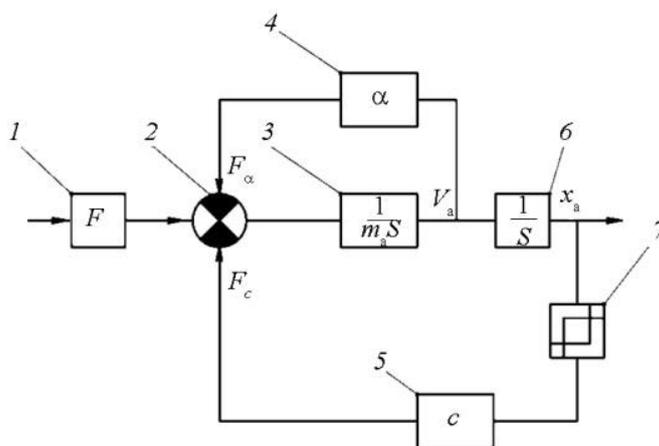


Figure 3. Structural diagram.

- 1 – tension; 2 –summation unit; 3 – 1/S Laplace operator and mass of the technological system unit;
- 4 – friction ratio; 5 – stiffness of the technological system unit; 6 – Laplace operator;
- 7 – element, describing the backlash of the mechanical unit.

The obtained results of the research are presented in the form of spectrograms (Figure 4).

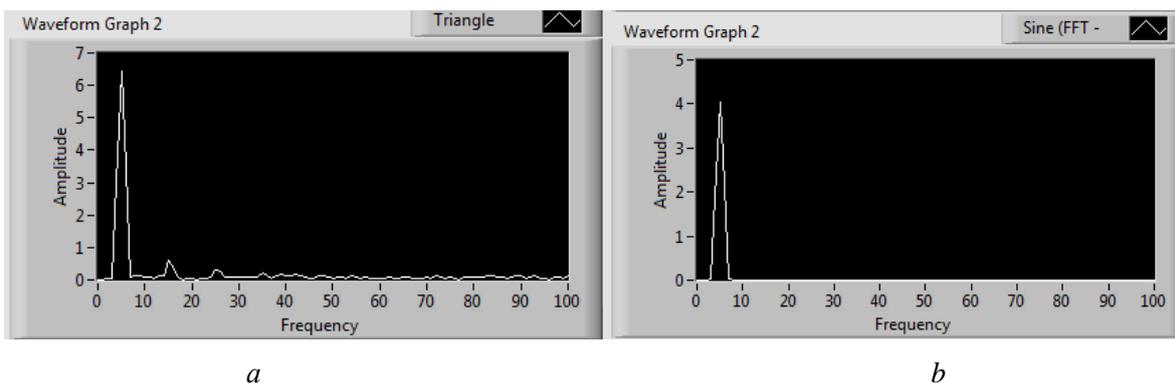


Figure 4. Spectrograms of the process obtained by simulating the operation of the damper
 a – at a «low» stiffness of the system and backlash presence;
 b – at a «high» stiffness of the system and backlash absence.

The first diagram (Figure 4a) shows the presence of background noise, which can be explained by the presence of backlash in the joint.

The second diagram (Figure 4b) built in the presence of a damping device (at backlash absence) and which shows the absence of noise.

4. Conclusion

The results of the simulation conducted in the Lab View National instruments, confirmed the adequacy of the structural diagram (Figure 3) and the promising use of the damping device.

Basing on further comparison of spectrograms we made a conclusion that the presence of backlash between the operating body and the machine-tool leads to an increased oscillations amplitude that negatively affected the processing quality of the parts [14-16].

As a result of applying preload, the reduction in vibration and noise in the technological system is attained, the quality of treatment increases, tool life increases by reducing the intensity of thermal processes in metal cutting. It is confirmed by experimental data.

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