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Marcin WOŹNIAK*, Arkadiusz PARUS
Faculty of Mechanical Engineering and Mechatronics, West Pomeranian University of Technology Szczecin, Poland
* Corresponding author: marcin.woźniak@zut.edu.pl

ACTIVE VIBRATION CONTROL OF A SLENDER WORKPIECE DURING TURNING

Key words: chatter vibration, vibration control, machining, vibration eliminator.

Abstract: The article presents a concept of using a controlled vibration eliminator to reduce vibrations generated during the turning process. The eliminator is embedded into the active holder supporting workpiece. A model of mechatronic system consists of a hydraulic actuator, a workpiece, and the machining process. A model of the system is used in the simulation of turning long, slender object. In the paper, the results of simulation for machining with and without vibration eliminator are presented. Based on the simulation, it was found that the designed system can limit vibration amplitude and increase the vibration stability in the turning process.

Aktywny układ tłumienia drgań smukłego przedmiotu obrabianego na tokarce

Słowa kluczowe: drgania samowzbudne, eliminacja drgań, obróbka skrawaniem, eliminator drgań.

Streszczenie: W artykule przedstawiono koncepcję sterowania eliminatora do redukcji drgań powstających podczas procesu toczenia. Eliminator drgań został wbudowany w podtrzymkę przedmiotu obrabianego. Model systemu mechatronicznego uwzględnia hydrauliczny układ wykonawczy, przedmiot obrabiany oraz proces skrawania. Zasymulowano, a następnie porównano zachowanie się obiektu z oraz bez układu sterowania. Na podstawie symulacji stwierdzono, że zaprojektowany układ jest w stanie efektywnie ograniczyć amplitudę drgań oraz zwiększyć wibrostabilność procesu skrawania na tokarce.

Introduction

The typical approach during modelling a machine tool is the assumption that it can be modelled as a Mass-Dissipative-Spring system (MDS), i.e. it has properties of an oscillation system with subcritical damping [1]. During machining, the system of machine tool–holder–workpiece–cutting tool has feedback by the machining force. As a result of incorrect selection of machining parameters, self-excited vibration (chatter) may occur. This leads to an uncontrolled increase of the vibration amplitude of the workpiece and cutting tool [2, 3].

The formation process of self-excited vibrations is directly related to the phenomenon of trace regeneration, i.e. the change of the cutting layer thickness $\Delta a(t)$ associated with the variable cutting force $\Delta F_{MP}(t)$ and vice versa [4]. The main effects of these vibrations are the poor surface roughness of the workpiece (lower accuracy of finished surface), the shortening of the life of the cutting tool and the entire machine tool, and increasing environmental noise. In summary, a high level of vibration causes an increase in productivity costs. In order to predict the vibrostability of the system, stability lobs are determined [5, 6]. In order to increase vibrostability and suppress vibration during machining, a wide range of active and passive methods can be used [4]. The passive methods involve an additional MDS system attached to the basic system. The goal of the added MDS is to receive and dissipate the energy coming from disturbances. In the turning process, a famous method used for increasing vibrostability is machining with a holder or tailstock support. A modern solution is the silent tool series produced by the Sandvik Company. The silent tool contains oil inside the tool that increases the damping ratio of vibration. Another solution of engineering is to implement a dynamic vibration eliminator like in the skyscraper "Taipei 101," which protect the building against winds, earthquakes, etc. [8]. Active methods involve supplying additional energy into the basic system by using an external energy source. In the active system, the force generated by actuator is calculated using control law. The ideal case is when the external force $F_{external}$ coming from the control system is equal to the basic force system F_{system} with the negative sign (phase is equal -180 degree).

$$F_{external} = -F_{system} \tag{1}$$

Forces acting on the system cancel each other out, which leads to a reduction of vibration. In practice, this case is impossible to realize in a wide range of frequencies, due to properties of the actuators, because every actuator has a limited passband. The implementation scope of the active method is limited to construction machine control vibrations [9], buildings [10], machine operator's seats [11], vehicle suspension [12], active machining holders [13], etc. In this article, an active support holder with the hydraulic actuator has been proposed. Hydraulic actuators generate a force that counteracts workpiece vibrations.

1. Vibration eliminator

Further analyses assume that the workpiece is a weak element of the machine tool–holder–workpiece––cutting tool (i.e. the element with highest susceptibility). Vibration reduction occurs in the direction of radial force. This means that the occurrence of vibration is associated with the unfavourable dynamic properties of the workpiece. The dynamic properties of the machine tool and cutting tool have a small impact on the formation of vibration in comparison to the workpiece properties.

It should be emphasized that, in practical application, the holder support should be attached close to the cutting zone to achieve the best results in the minimization of workpiece vibration. However, in some cases, the distance is relatively large due to the geometrical properties of the cutting tool or machine tool construction. Using a passive workpiece holder does not provide sufficient improvement of vibrostability in this case.

Figure 1 shows the real object (b) with the block diagram (a). The concept of the vibration eliminator is as follows: Control signal $u_{fs}(t)$ is calculated based on the measurement of workpiece **(5)** accelerations $\ddot{x}$ and control law. The control signal is used to control the hydraulic distributor (flow servo valve) **(1)** which generates movements of the hydraulic actuator with additional mass **(2)** and the support of the holder **(3)** with jaws clamping workpiece **(4)**. The motion of the additional mass generated force leads to damping workpiece vibration. **(5)**. Workpiece vibrations are caused by the cutting force generated by the cutting tool **(6)**. In laboratory conditions, the cutting force is excited by a TIRA inductor – model 5220 – M. Details of the principle and design of the construction of a hydraulic holder can be found in article [14].

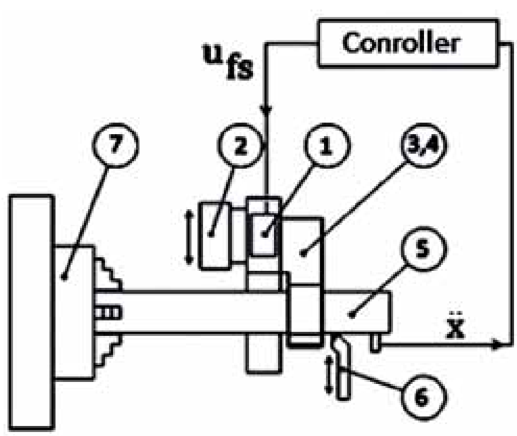
1.1. Modelling of MDS system

During modelling of the system, the necessary simplifications were made to linearize the mechatronic system. The mathematical model of the real object based on differential equations (2) was developed.

The differential equations of the described system are as follows [1, 4]:

$$\begin{gathered} m_{CT}\ddot{x}_{CT} + h_{CT}\dot{x}_{CT} + kx_{CT} = -F_{MP} \\ m_W\ddot{x}_W + h_W\left(\dot{x}_W - \dot{x}_B\right) + h_{W-S}\dot{x}_W + k_{W-S}x_W = +F_{MP} \\ m_B\ddot{x}_B + h_B\dot{x}_B + h_W\left(\dot{x}_B - \dot{x}_W\right) + h_E\left(\dot{x}_B - \dot{x}_E\right) + \\ + k_B x_B + k_W\left(x_B - x_W\right) = -F_{external} \\ m_E\ddot{x}_E + h_E\left(\dot{x}_E - \dot{x}_B\right) = +F_{external} \end{gathered} \tag{2}$$

a)



b)

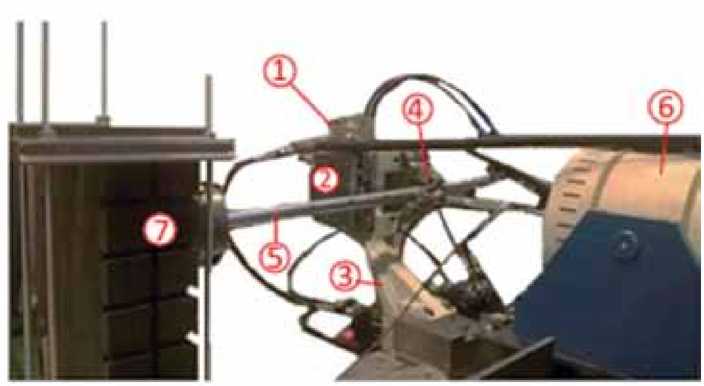


Fig. 1. a) Block diagram, b) Real object
Source: Authors.

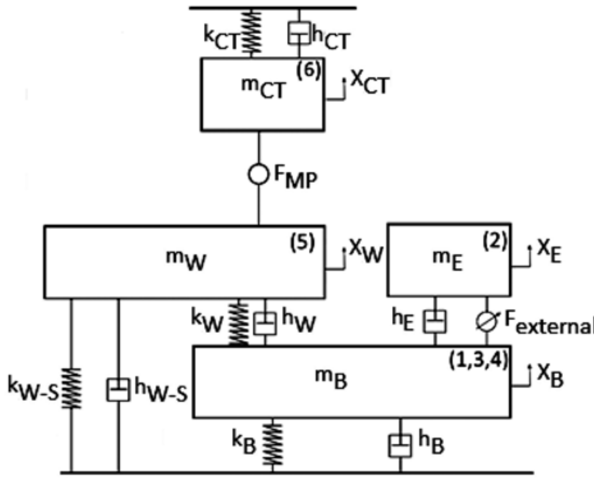


Fig. 2. Phenomenological model of the MDS system

Source: Authors.

k_{CT} – stiffness cutting tool – ground,
h_{CT} – damping cutting tool – ground,
m_{CT} – mass of cutting tool,
F_{MP} – disturbance force depends on machining process,
$F_{external}$ – external (hydraulic) force,
k_W – stiffness workpiece – body,
h_W – damping workpiece – body,
m_W – mass of workpiece,
k_B – stiffness body – ground,
h_B – damping body – ground,
m_B – mass of body,
h_E – damping eliminator – body,
m_E – mass of eliminator,
k_{W-S} – stiffness workpiecie – spindle (ground),
h_{W-S} – damping workpiece (ground) – spindle

The number in brackets refers to the symbols in Figure 1.

1.2. Actuator modelling

In active vibration control, a wide range of actuators are used (i.e. pneumatic, hydraulic, piezoelectric, and electromagnetic) [15]. In this article, as already mentioned, the medium of additional energy counteracting vibrations in the workpiece–cutting tool system is liquid. The conversion of hydraulic energy into mechanical energy takes place by means of hydraulic cylinders. The element controlling the flow of liquids in the system is the flow servovalve – MOOG series 760, whose internal structure is shown in Fig. 3b.

The general operation of the hydraulic system (Fig. 3a) is as follows: The flow of liquid in the hydraulic system is forced by a hydraulic pump **(2)**, which is driven by electrical motor **(1)**. By means of an external control system **(3)**, the distributor **(4)** controls the direction of the liquid flow and thus the direction of the movement of the hydraulic cylinder **(5)**. When liquid is supplied to one of a cylinder chambers, it returns to the tank from other chamber **(6)**. To protect the hydraulic system against overloads (pressure protection), the pressure reducing valve is built into the hydraulic system **(7)** [17, 18].

The principle of the servo valve operation (Fig. 3b) is as follows: The operation of the servo valve is divided into two parts: electrical and mechanical. First part is called an electromechanical converter. The current in the coils (1) operating in the differential configuration causes in the torque motor (2) rotation of the magnetized armature (3) that is connected to a thin-walled,

a) b)

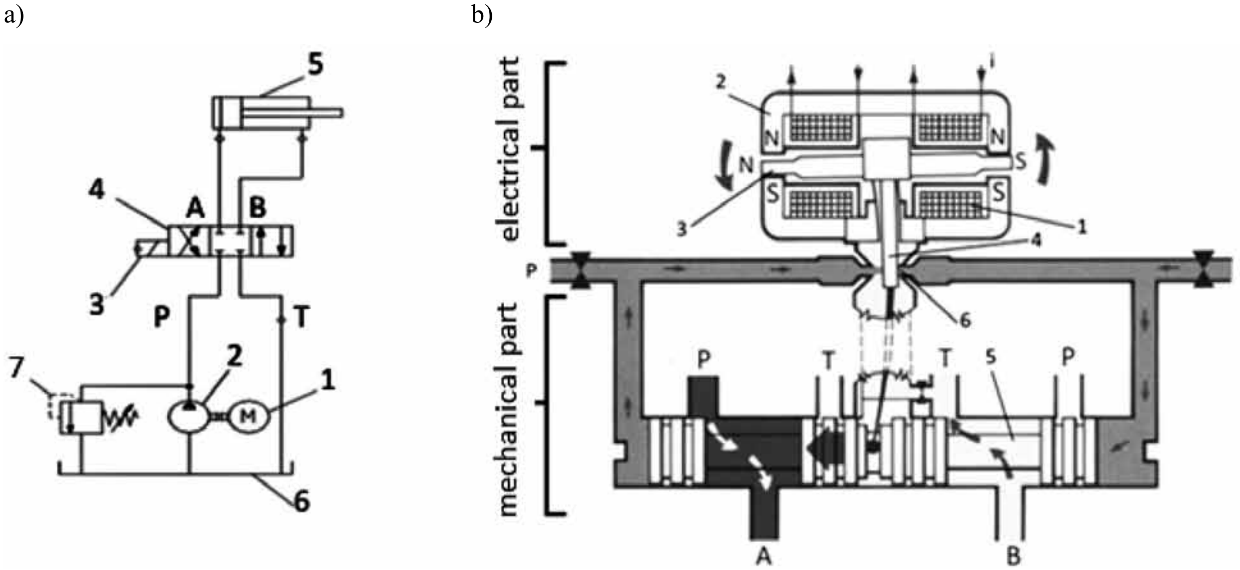


Fig. 3. a) Principle of the hydraulic system, b) Servovalve build

Source: [16].

deformable sleeve (4). The jumper (3) is suspended in the air gap by the magnetic field generated between the electromechanical transducer and the piston rod (5). The rotation of the armature (3) results in the obstruction of one of nozzles (6). The obstructed nozzle causes the extension of the piston rod (5), which properly directs the flow of liquid in the hydraulic system [17, 18].

Modelling of the hydraulic part of the system can be divided into the following two stages [18]:

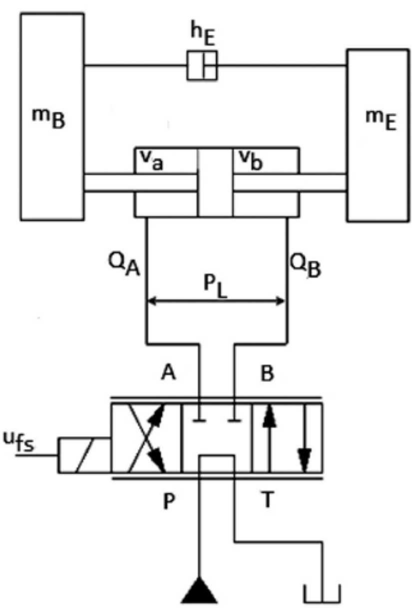
1) The electromechanical system is crucial for the dynamics (transfer function) of the slider distributing the liquid flow in the valve x_v and the voltage control signal u_{fs}.
2) The liquid flow system is described with the phenomena occurring in the distributor in the hydraulic – transfer function $Q(x_v, P_L)$ between the liquid flow value and the position of the distribution slider and the pressure occurring in the system.

1.2.1. Hydraulic system

The torque actuator generates a torque value, whose value is proportional to the current of the *RL system*. This value is determined by Equation (3). The displacement of the piston rod (Fig. 3b – element 5) due to the current in the electrical system is described by Equation (4).

$$\frac{i(s)}{u_{fs}(s)} = \frac{1}{Ls+R} \tag{3}$$

i – current [A], u_{fs} – control signal [V].



x_v – piston rod position of servovalve (fig. 3b – element 5) [*m*],
$A(x_v)$ – liquid flow cross-section area [m^2],
w – liquid flow linear gradient [*m*],
P – source pressure [*Pa*],
P_L – pressure depend of loading equals $P_L = P_A - P_B$ [*Pa*]

Fig. 4. Actuator model with subsection MDS (Fig. 2)
Source: Fluidsim.

The model of the dynamic piston rod is a second order system, which can be described as following transfer function:

$$\frac{x_v(s)}{i(s)} = \frac{\omega_n^2}{i_{MAX}\left(s^2 + 2\zeta\omega_n s + \omega_n^2\right)} \tag{4}$$

ζ – dimensionless damping ratio [1],
ω_n – radial frequency $\left[\frac{rad}{s}\right]$,
i – current, i_{MAX} – **max**imum current in RL system [A].
x_v – piston rod displacement [m],

The readout of the dynamic parameters of the piston rod, i.e. ζ or ω_n, can be read from the frequency characteristics provided by the manufacturer, e.g., [19].

1.2.2. Hydraulic system

In this section, the function liquid flow in the servo valve and hydraulic cylinder was linearized.

The following simplifications are assumed [16]:

- No external or internal leakage occurs in the system $\left(Q(xv=0)=0\right)$.
- The value of return pressure to the tank is equal to 0
- The actuator cylinder is a double piston rod.
- The movements of the actuator are small, i.e. volume of chamber A and B are equal $V_a = V_b = \frac{V_0}{2}$.
- Laminar flow occurs.
- There is a zero cover in the servo valve $Q(xv \neq 0) \neq 0$.

Using the hydrodynamics equation and considering, that $v = \frac{Q}{A}$, the flow function can be described as [16, 18]:

$$Q(x_v, P_L) = A(x_v)C_d\sqrt{\frac{2}{\rho}(P-P_L)} = wx_vC_d\sqrt{\frac{2}{\rho}(P-P_L)} \tag{5}$$

Analysing Equation (5), the linearization of the liquid flow (assuming constant working pressure) in the system can be described by following equations [16]:

$$\Delta Q = K_q\Delta x_v + K_c\Delta P_L \tag{6}$$

$$K_q = \frac{\partial Q}{\partial x_v} = wC_d\sqrt{\frac{2}{\rho}(P-P_{L0})}, P_L = const \tag{7}$$

$$K_c = -\frac{\partial Q}{\partial P_L} = \frac{wx_{v0}C_d}{2\sqrt{\frac{2}{\rho}(P-P_{L0})}}, x_v = const \tag{8}$$

K_q – the gain factor related to the displacement of the piston rod x_v, $\left[\frac{m^2}{s}\right]$,
K_C – the gain flow – pressure factor – it describes how the change in the load influences the change of the flow rate at $x_v = 0$.

In the article it is assumed that there are no external loads in the system $P_{L0} = 0$, and the piston rod is in the central position $x_{v0} = 0$ for $u_{fs} = 0$. The liquid flow equation is as follows:

$$Q(x_v) = K_q x_v \tag{9}$$

The pressure value in the actuator chambers A and B is described by equations (10) and (11):

$$Q_A = A_p \dot{x}_p + \frac{V_A}{\beta} \dot{P}_A \tag{10}$$

$$-Q_b = -A_p \dot{x}_p + \frac{V_B}{\beta} \dot{P}_B \tag{11}$$

A_p – cross-section area of actuator **p**iston rod [m^2],
$\dot{x}_p$ – relative speed eliminator – hydraulic holder $\left[\frac{m}{s}\right]$.

The force acting on the MDS system depends on the product of the actuator's working area and the pressure difference between the A-B chambers and the surface.

In addition, the flow Q in the system is equal to $Q = \frac{Q_A + Q_B}{2}$.

$$\dot{P}_L = \dot{P}_A - \dot{P}_B = \frac{4\beta}{V_0}\left(K_q x_v - A_p \dot{x}_p\right) = C_{mb}\left(K_q x_v - A_p \dot{x}_p\right) \tag{12}$$

β – compressibility module [Pa^{-1}].

The force of the actuator is equal (13) – refer to Equation (2):

$$F_{external} = A_p P_L \tag{13}$$

In summary, the executive system can be presented in diagram form:

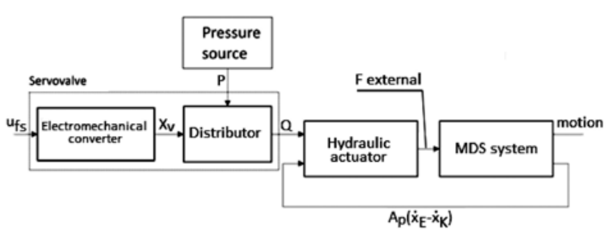


Fig. 5. Block diagram of executive system
Source: Authors.

2. Control system and disturbance model

The system is built of the following:

- The machining process with a disturbance model including the feedback between the workpiece and cutting tool,
- MDS executive system and the hydraulic component, and
- Control law block – controller and measurement system.

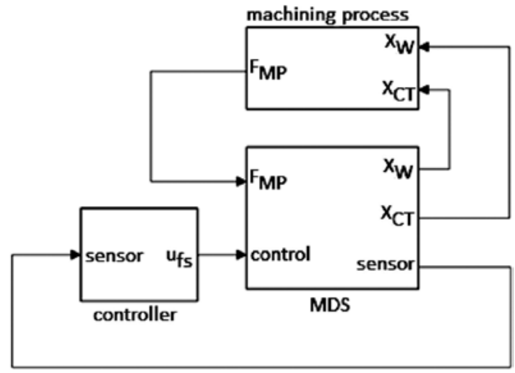


Fig. 6. Block diagram of active damping control system
Source: Authors.

The vibration stimulating signal is a cutting process that can be modelled with the following equation (referring to Equation 2) [4, 5]:

$$F_{MP} = b_0 k_s \left[a_0 - \left(x_W(t) - x_{CT}(t)\right) + \left(x_W(t-\tau) - x_{CT}(t-\tau)\right)\right] \tag{14}$$

k_s – coefficient resistance of cutting tool $\left[\frac{N}{mm^2}\right]$,
b_0 – width of the machined layer [mm],
a_0 – the thickness of the machined layer [mm],
τ – turnover period $= \frac{60}{rpm}$ [s],
rpm – revolution per minute $\left[\frac{1}{minute}\right]$.

For active damping control, a cascade PID control was used. The outer loop is for workpiece acceleration $\ddot{x}_W(t)$ and inner loop is for the current coil of the servo valve $i(t)$ control.

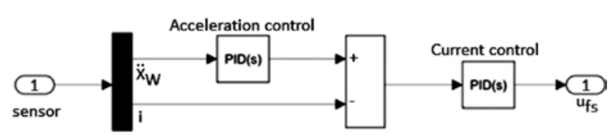


Fig. 7. Block diagram of active damping control system
Source: Authors.

Results and conclusions

The results of machining process simulation are presented in two scenarios – turning with active and inactive controller PID's. The diameter of the slender workpiece is 40 mm, and it is 1000 mm long. In the upper part of Figure 8, displacements of the workpieces for both scenarios are presented. The middle part presents the force of the vibration absorber, and the lower part presents the cutting force during machining with and without an active control system.

Referring to Figure 8, it was observed that the use of an active vibration eliminator attached to the hydraulic holder can reduce vibrations of the workpiece. Maximal displacement of workpiece has almost the same value in

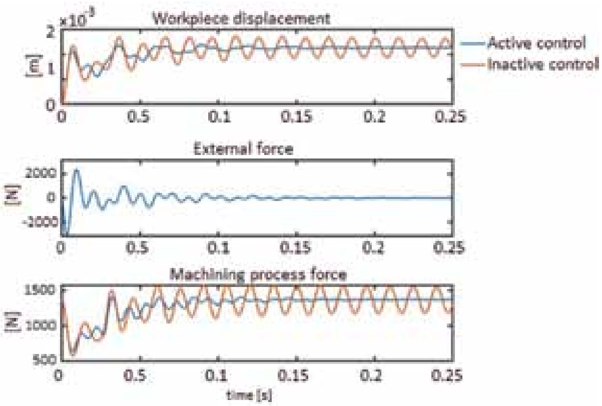


Fig. 8. Displacement of workpiece $x_W(t)$, Control force $F_{EXTERNAL}(t)$ and cutting force $F_{MP}(t)$
Source: Authors.

the active and inactive scenarios. The main difference is that active system can reduce oscillation faster, which makes the machining process is more stable. This means that the boundary layer width is increased relative to the system without an active vibration controller.

This paper presents the concept of reducing the vibration level during the machining process by using a vibration eliminator connected to hydraulic holder. A model of the system consisting of hydraulic subsystems (executive element and servo valve) and a mechanical system (with resonance frequencies and amplitudes) was developed. Details on the synthesis of the control system will be presented in a separate article. In a further stage of the works, object identification is planned along with the reduction of the model [20, 21] and then the validation of the developed control system.

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