

## USING OF ACTIVE CLAMPING DEVICE FOR WORKPIECE VIBRATION SUPPRESSION

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**Abstract.** *Vibrations, which always appear in the machine tool - cutting process dynamic system (MT-CP) are very undesirable. Their negative influence on productivity, tool life and surface quality is well known. These facts cause that new methods of counteracting these phenomena are still developed and improved. The starting point of the investigations presented here was the statement that a workpiece is responsible for loosing stability of dynamic MT-CP system, and uncontrolled self-excited vibration can develop to an unacceptable level. In the paper the method of counteracting chattering through the application of a special workpiece clamping system is described. The idea presented at the 10<sup>th</sup> International Conference on Vibration Problems was to use active elements in the workpiece holder construction. This allows to modify dynamic properties of the workpiece and actively control its vibration. In this work piezo-actuators are used as active elements. The controller is used to shape the dynamic characteristic of the machine tool – cutting process system. Thus the damping factor in the vicinity of natural frequencies is significantly increased and the stable machining can be achieved at higher cutting parameters. The model of the active clamping system with a fixed workpiece is presented in the paper. Numerical simulations for different cutting parameters were conducted. In all cases lower vibration amplitudes were observed in comparison to the machining without the active clamping system. The time plots of selected signals are presented for machining of a workpiece sample. The experimental validation of the proposed solution was conducted for a wide set of machining parameters (spindle speed, feed rate and depth of cut). Experiment was executed on DMU 60 Monoblock milling machine with laser measurement system. A chatter noise processing helped to detect vibration induced by a slender cutting tool.*

## 1 INTRODUCTION

Self-excited vibration that develops during machining of materials, called „chatter”, is a phenomenon deteriorating the machining precision and quality of the workpiece surface. This kind of vibration, due to high energy, can cause unrecoverable damages to the workpiece, reduction of life time or breakage of a cutting tool, holders or components of a machine tool. Chatter vibration could also be a source of an oppressive noise. The simplest way for avoiding chatter is the reduction of cutting parameters. This, however, decreases the output of the machine. Therefore, other methods for chatter suppression are needed. They are usually connected with the search for other cutting conditions which guarantee stable cutting or influencing the machine – tool – holder – workpiece system. The latter approach can be effective not only at the machine tool, tooling or holder design stage but also when controlling the machine tool on-line, during cutting. This influence can have passive or active character. A characteristic of different counteractions applied against chattering can be found e.g. in the review [1].

In this paper the development and investigation of an active system that controls additional movement of a workpiece during milling is presented. The workpiece is clamped on a special table which motion is actuated by piezo-stacks. The system can be applied for shifting the cutting stability limit to higher values of the width of cut but only for rather light workpieces since heavy ones require high energy to control their motion, especially at higher frequencies.

Algorithms implemented in vibration monitoring systems, applied for improving on-line the cutting stability, are usually performed on the following stages: chatter detection through the analysis of selected chatter symptoms, examination of the system state aimed at the preparation of a suitable action for cutting process stabilization, and controlling of actuating devices. The system developed in this work that controls the motion of a workpiece does not classify the vibration nor evaluates its magnitude but, through the forces acting on the workpiece table, suppresses the vibration of the workpiece independently of the vibration nature and level [2]. It is assumed here that chatter develops due to insufficient stiffness of the workpiece and its vibration is measured. In order to provide the possibility of detection of instability caused by highly flexible cutting tool the system was equipped with microphones and sound signal processing system catching the increase of chatter vibration and its frequency.

## 2 MODEL OF ACTIVE CLAMPING SYSTEM

A wide variety of technical solutions could be used to reduce vibration level during machining. The source of the problem determines the type of used vibration control system. In the case when high level of vibration is caused by a slim, compliant cutting tool the methods with an adaptive spindle speed control could be successfully used [3]. Otherwise, the dynamic properties of the machine tool-cutting process can be affected by a workpiece with very low stiffness. In the paper [4] authors use an active vibration absorber attached to the workpiece. This 1-DOF method works effectively when the workpiece exhibits only one dominant direction of vibration. Figure 1a presents time plot of displacement in two perpendicular axes X and Y during milling simulation. Vibration control system is active only in Y axis. It is clear that in this case loss of stability could be connected with vibration in X axis. In Figure 1b a vibration control system is active in both X and Y axes and time plots are smooth with no signs of the regeneration phenomenon. For this reason, designing an effective vibration control system of the workpiece requires the use of 2-DOF devices.

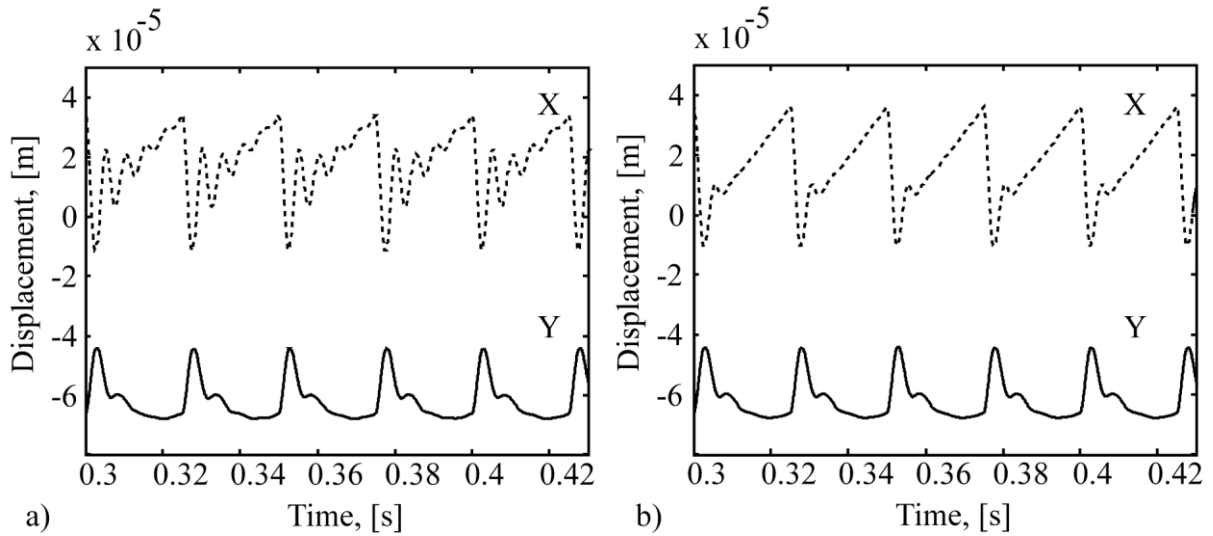


Figure 1: Displacement of the workpiece during milling simulation with vibration control system active in a) Y axis, b) X and Y axis.

On this basis in paper [2] authors developed 2-DOF active clamping system. This device allows to mount the workpiece on the active plate which can be driven by piezo-actuators in the range of  $\pm 50 \mu\text{m}$  in both X and Y axis. Figure 2 shows a model of the analyzed system.

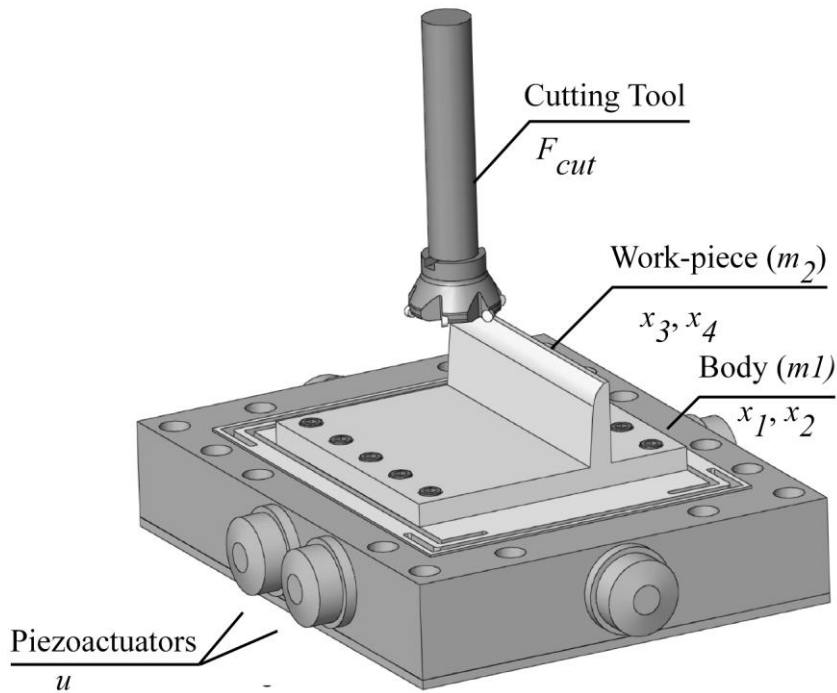


Figure 2: Model of the active clamping system.

Simulation was conducted for different sets of cutting parameters using the following constitutive equations describing the system in X axis:

$$\begin{aligned}
\dot{x}_1 &= x_2 \\
\dot{x}_2 &= \frac{K_p n d_{33} u}{m_1} - \frac{K_1 + K_p + K_2}{m_1} x_1 - \frac{C_1 + C_2}{m_1} x_2 - \frac{K_2}{m_1} x_3 - \frac{C_2}{m_1} x_4 \\
\dot{x}_3 &= x_4 \\
\dot{x}_4 &= F_{cut} + \frac{K_2}{m_2} x_1 + \frac{C_2}{m_2} x_2 + \frac{K_2}{m_2} x_3 + \frac{C_2}{m_2} x_4
\end{aligned} \tag{1}$$

where:

$K_p$  – stiffness of the piezoelectric stack,

$d_{33}$  – piezoelectric constant,

$n$  – number of piezo layers,

$K_1, C_1, m_1$  – stiffness, damping and mass of the active clamp body,

$K_2, C_2, m_2$  – stiffness, damping and mass of the workpiece with movable part of active clamp,

$u$  – control signal of piezoactuator,

$F_{cut}$  – cutting force,

$x_1, x_2$  and  $x_3, x_4$  are displacement and velocity of the body and workpiece respectively.

The model in Y direction is determined in the same way as in X. Significant reduction of the workpiece vibration level was achieved when using LQG controller. On this basis the real active clamping device was manufactured and tested on the CNC machine [5].

### 3 CONTROL SYSTEM

A similar procedure to the one shown in [2] has been applied in order to examine the effectiveness of developed solutions. A workpiece mounted in the active clamping system instead on the machine table directly exhibits smaller value of static stiffness which influences also the stiffness of the whole machine tool-cutting process system. Moreover, depending on the workpiece dynamic properties new resonant frequencies can be observed. Therefore, the model described by Eq. (1) couldn't be directly adopted in the LQG synthesis and additional identification procedure is required. The aims of the control system are: compensate the reduced stiffness and suppress the workpiece vibration.

#### 3.1 Identification of control object

Identification of the model consists in fitting the observed and controlled linear model with the experimentally determined transfer functions by applying to piezoactuators in X and Y axis supply voltage with linearly changing frequency from 10 to 1000 Hz for 10s with a constant amplitude. The experiment was conducted for several various values of signal amplitude and the results are presented in Figure 3.

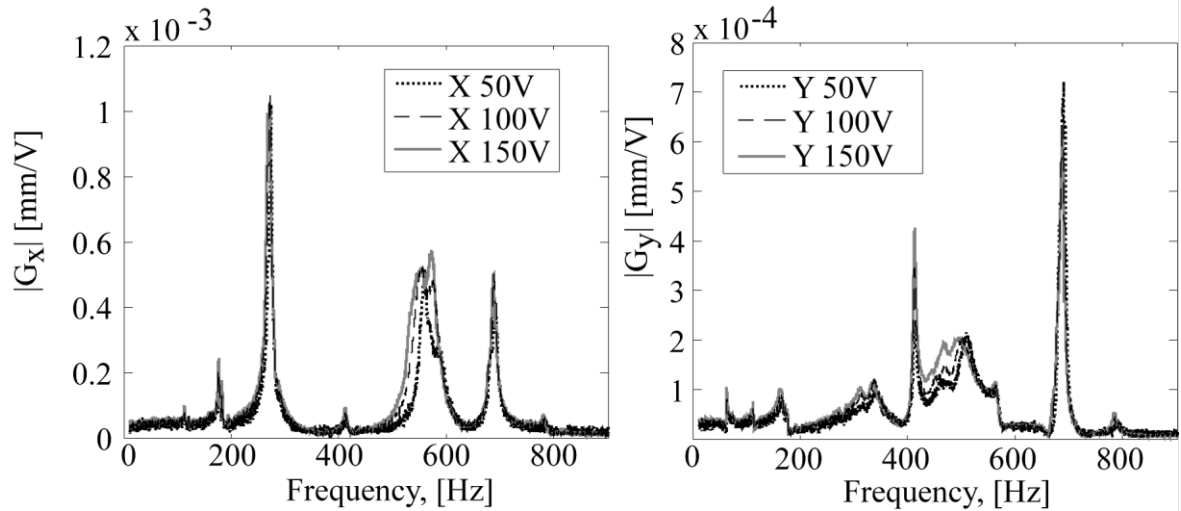


Figure 3: Modulus of experimentally determined transfer functions  $G_x$  and  $G_y$  for different values of voltage applied to piezoactuators.

Relatively small difference between characteristics could be observed in range 400-600 Hz but frequency of the dominant vibration remains the same for all applied voltage signal. The self-excited vibration usually arise at frequency close to one of the dominant natural frequencies. Taking this into consideration the identified model could be treated as linear and time invariant in frequency bands 0-400 Hz and 600-900 Hz, therefore an LQG controller can be applied. In Fig. 4 gray line represents the modulus of the experimentally obtained transfer function whilst black line corresponds to the transfer function obtained on the basis of an identified model. In both axes the amplitude of excitation signal was 100 V.

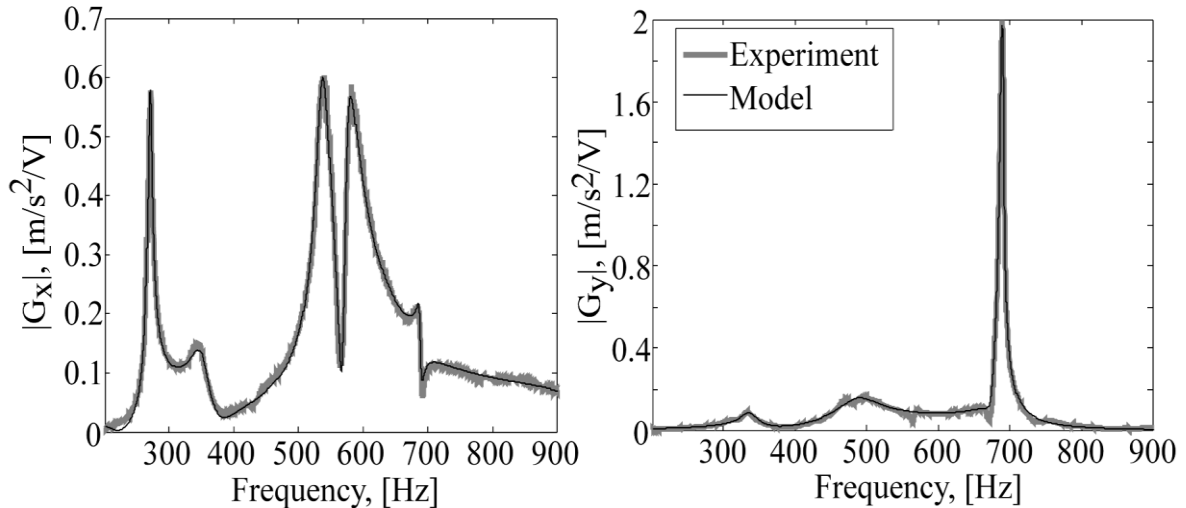


Figure 4: Modulus of the experimentally determined (gray) and identified (black) transfer functions  $G_x$  and  $G_y$ .

### 3.2 LQG controller

The objective of the control input  $u$  is to minimize the cost function

$$\mathbf{J}(u, y) = \int_0^{\infty} (\mathbf{y}^T \mathbf{Q} \mathbf{y} + \mathbf{u}^T \mathbf{R} \mathbf{u}) dt \quad (2)$$

where values of weight matrices  $\mathbf{Q}$ ,  $\mathbf{R}$  were determined using numerical simulation taking into account the machining process model and dynamic characteristics of the workpiece. The values in  $\mathbf{Q}$  and  $\mathbf{R}$  were tuned until the highest efficiency of the system was obtained, and control signal was kept within permissible limits ( $\pm 500$  V). Parameters determined that way were corrected after a few attempts of milling a real specimen. The influence of control on the system's dynamics can be observed in Figure 5.

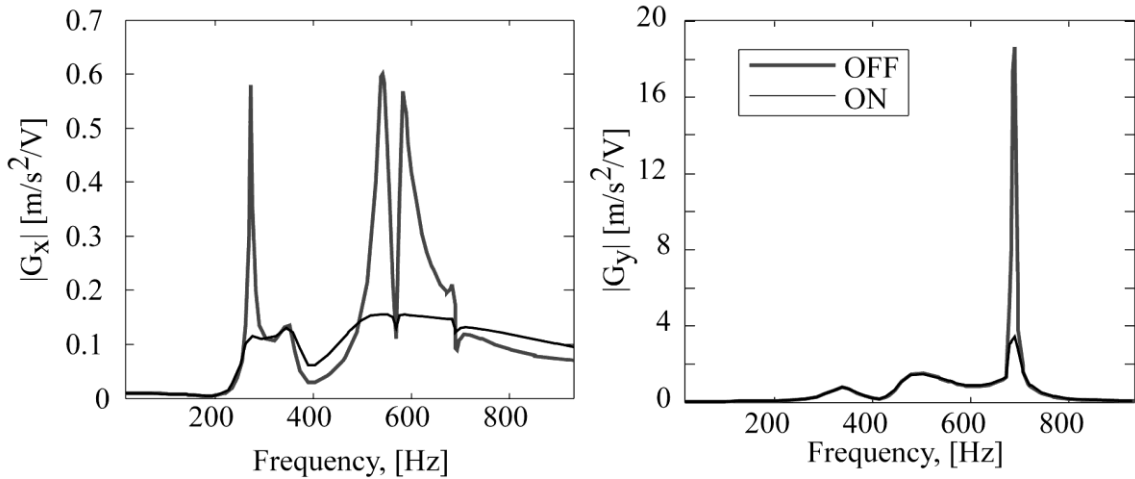


Figure 5: Vibration amplitude for the system with and without LQG controller.

Figure 6 shows transient workpiece response to a force impulse applied by a modal hammer Kistler 9726A20000 in X and Y without control system whilst Figure 7 with LQG controller. In the latter case significant increase of damping in the system is observed.

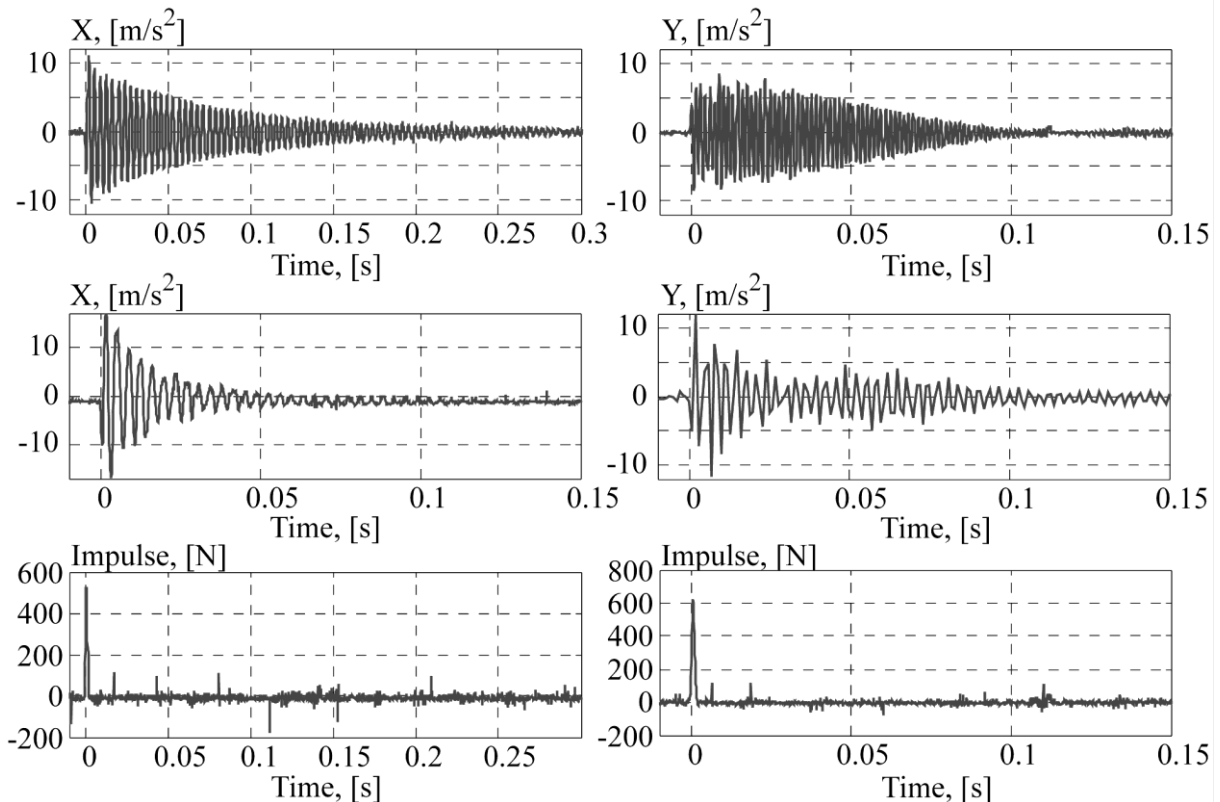


Figure 6: Transient response of workpiece to a force impulse without control system (at the top) and with LQG controller (in the middle).

#### 4 EXPERIMENT

The results of vibration reduction when an active clamping system with piezoactuators was used were verified by tests carried out on a numerically controlled DMU 60 Monoblock machine tool. The milling was executed with cutting tool DIN 845 B-25 K-N HSS with nominal parameters without cooling. Acceleration and velocity of the workpiece were measured with three-axial PCB 356A01 sensor and Polytec PSV 3D Laser Vibrometer respectively. Identification procedure and LQG controller were implemented in dSpace 1006 real time rapid prototyping system. In Figure 8 sample time plots of acceleration, velocity and voltage of the piezoactuator are presented. At time equal 5 s the controller was turned off and significant increase of vibration can be observed on acceleration and velocity signal. It is worth to notice that voltage of the piezo is bounded in permissible range. Figure 9 shows a surface of the workpiece. The worsening of surface quality could be observed in the right-hand part of the picture where controller was turned off.

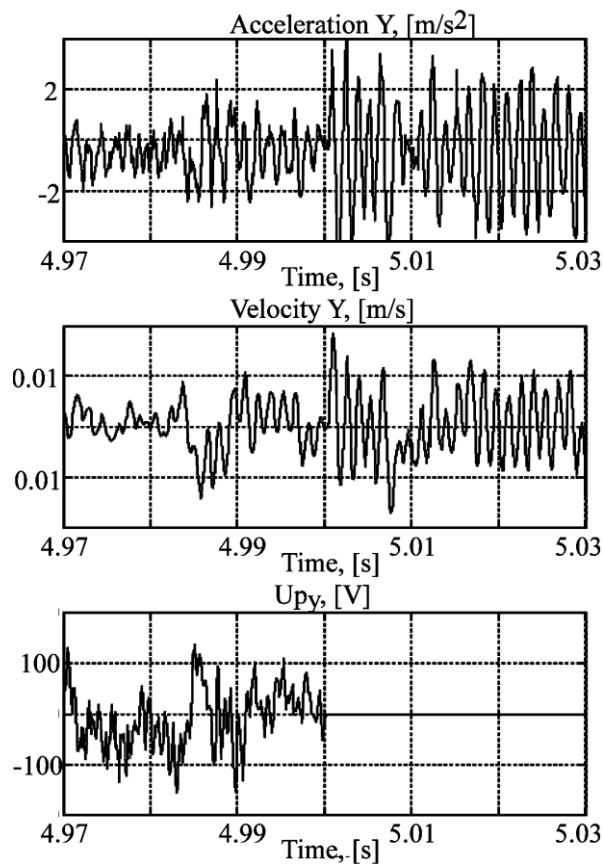


Figure 8: Acceleration and velocity of the workpiece and control signal of the piezoactuator during milling experiment with active clamping system.

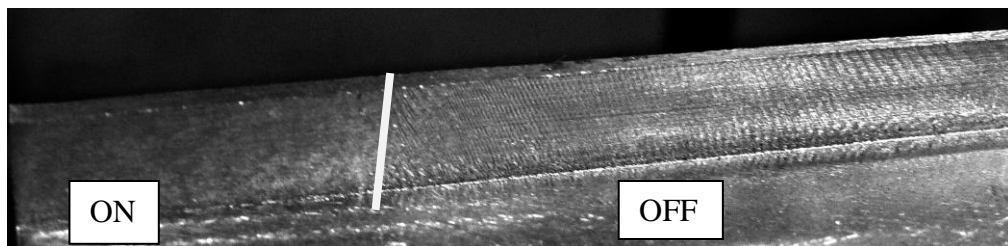


Figure 9: Machined surface obtained with the control system on and off.

## 5 SUPPLEMENTARY SYSTEM AND CONCLUSIONS

Inputs to the machine tool vibration monitoring system should provide information necessary for building chatter symptoms. Cutting force components, vibration (usually measured with accelerometers) or noise (from microphones) are mostly used as inputs. Acoustic emission signals or “sensorless” systems that measure e.g. servomotor current are used occasionally. The main advantage of a force signal is its direct dependence on the difficult to measure relative vibration between the cutting tool and the workpiece. However, the application of dynamometers in production environment is inconvenient and risky. The measurement of acceleration is cheap, accurate and easy but the sensors often cannot be placed close enough to the cutting area. Therefore, they are rather insensitive to vibration of a slender tool or compliant workpiece. Those vibrations are not able to excite heavy elements of the machine tool. Microphones are almost ideal for this application. The problem lies, however, in the disturbance of the effective signal by environmental noise uncorrelated with chatter. In this work two microphones were used and STFT for the detection of chatter development. For the improvement of S/N ratio in the sound signals a simple band-pass filtration was applied.

The main disadvantage of the active workpiece clamping system is the application of laser for measuring the workpiece vibration. In some conditions it is not sufficiently reliable and cannot be used when a coolant is applied during cutting. A newly designed workpiece clamping device table and a new configuration of piezo-actuators should improve the efficiency of chatter suppression. The new system is now under tests.

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