International Journal of Modern Manufacturing Technologies
ISSN 2067–3604, Vol. XI, No. 1 / 2019

APPROACH TO INERTIAL-DYNAMIC COMPENSATION OF THE CUTTING FORCES BY MILLING WITH INDUSTRIAL ROBOTS

Oleksandr Kuklevskyi [1,2], Rainer Gadow [1]

[1]University of Stuttgart, Graduate School of Excellence Advanced Manufacturing Engineering
Nobelstr. 12, 70569 Stuttgart, Germany
[2]University of Stuttgart, Institute for Machine Tools
Holzgartenstraße 17, 70174 Stuttgart, Germany

Corresponding author: Oleksandr Kuklevskyi, oleksandr.kuklevskyi@gsame.uni-stuttgart.de

Abstract: This article focuses on alternative method to improve the milling accuracy for industrial robots using inertial-dynamic forces. These forces, which are created by acceleration and deceleration of oscillating masses, counteract the milling cutting forces and thereby lower the amplitude of the vibrations as well as the load-related path deviations. In the article the mutual influence of the milling parameters and the parameters of piezo shaker on the compensation of the cutting force as well as limits of the application of this method are analyzed.

Key words: milling accuracy; vibration; industrial robot; piezo actuator

1. INTRODUCTION

Until 2019, more than 1.4 million industrial robots will be newly installed worldwide, and thus the total number will rise to 2.6 million units, [1]. Most of the newly installed robots will contribute to a further automation of production. Hence, they will probably be used in the handling and the assembly of workpieces as well as for performing joining tasks, such as e.g. welding, clinching, etc.

The machining of workpieces represents another up-and-coming field of application for industrial robots. Today, industrial robots are already being used for material removal processes in respect of workpieces with low quality requirements. Compared with machine tools, they show a great flexibility at a comparatively low price and thus represent a fully adequate alternative to machine tools in some areas. Products with higher quality requirements are, however, manufactured with classic machining processes by machine tools. This is because the static and dynamic stiffness of industrial robots is comparatively low, so that the manufactured products have an insufficient accuracy to size as well as an inadequate surface quality, [2]. The quasi-static loads resulting from the process here lead to path deviations, whereas alternating loads, for example, due to waviness excite vibrations of the structure. The natural frequencies and eigenmodes of a machining robot depend to a great extent on the chosen pose, i.e. the adjusted shaft angle. Hence, it is only possible to specifically avoid the resonant ranges with a great computing effort before the machining process and thus economically unjustifiable, [3].

The problems with accuracy can be addressed in various ways:

-increasing the stiffness of the machining centre, [4], damping of arising vibrations, [5];

-online/offline compensation of the deviations from the tool trajectory, [6, 7].

The first option mentioned requires a fundamental adaptation or alteration of the machine structure. As a consequence, a completely new machining system must be developed. This requires a lot of time and money as well as leaves questions open regarding flexibility.

The other two approaches to increase the accuracy are, however, universally applicable, as they can be implemented in the form of add-on modules and attached to existing machining systems. The systems passively damping the vibrations here take away vibrational energy from the vibrating structure, so that the vibration amplitudes on the tool centre point (TCP) are reduced. In the case of an active vibration damping, the vibration amplitudes are minimised due to an energy input. There is an approach to damp vibrations during milling by the generation of inertial-dynamic forces using electromagnetic shakers, [8]. It uses adaptive control, which determines and sets the necessary frequency and phase of shaker oscillations on the basis of signals from acceleration sensors. This method was applied to a milling machine and allowed to reduce the vibration amplitude during milling and to obtain a higher quality of the treated surface.

The compensating systems carry out path corrections of the cutting tools deviating from the specified trajectory as a result of the machining system's low

stiffness due to the machining forces. The correction parameters are made here either at the process planning stage or during the machining as well.
The systems used for increasing the attainable accuracy have in common that they counteract the effects and not the underlying causes. Hence, for example, deviations from the required tool trajectory are compensated with path corrections instead of reducing the process force or neutralising its effect.

2. INERTIAL-DYNAMIC COMPENSATION SYSTEM

An alternative approach is proposed here. It is aimed at reducing the vibrations as well as the low-frequency oscillations or the load-related path deviations which arise in milling with industrial robots, [9]. The principle of this approach is based on reducing the resultant forces acting on the end effector of the robot during milling. It is planned to perform this reduction by generating inertial-dynamic forces, which counteract the cutting forces in milling. For this purpose, the generated forces should have the same magnitude but the opposite direction to the cutting forces.
It is proposed to use piezoelectric preloaded stack actuators for the generation of inertial-dynamic forces, since they can operate under high loads and have sufficient operating speed.

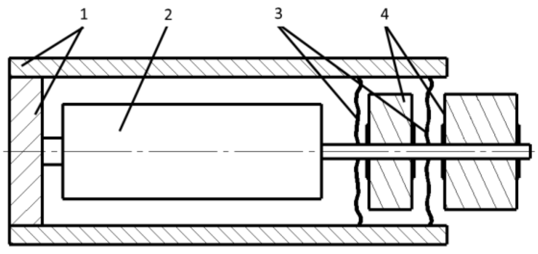

Fig.1. Electrodynamic shaker, consisting of housing (1), piezo drive (2), guide mechanism (3) and attached mass (4)

The device for creating inertial-dynamic forces using a piezo drive is shown schematically in Figure 1. It is called a piezo shaker and consists of a housing (1), a piezo drive (2), a guide mechanism (3) and an attached mass (4). The generation of the inertial-dynamic force occurs as follows: when a voltage signal is applied to the piezo drive, it moves with the corresponding dependence on time, also moving the attached mass, giving it a definite acceleration. The guide mechanism provides the straightness of the movement of the attached mass. When the moving masses of the piezo drive and the attached mass are accelerated, a force arises that counteracts the change in the amount of motion of these masses. This force is used to compensate for the cutting force. For that, it must have the same magnitude but the opposite direction. Thereafter, the drive with the attached mass returns to its initial position with the minimum possible acceleration.
A shaker of such design can compensate for the force only along its axis. To compensate forces in two or in three directions, it is therefore necessary to apply two or three shakers accordingly.
Figure 2 shows the location of the system actuators for the two-axes compensation.

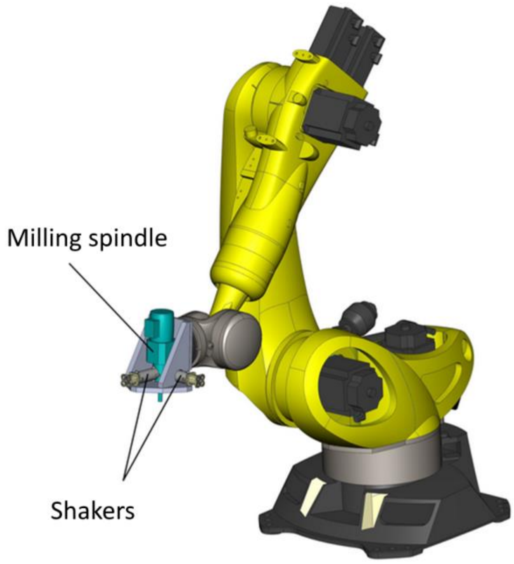

Fig. 2. Compensation system

This compensation system is used when the values of the axial forces of cutting during milling are much smaller than the cutting forces in the radial and tangential directions. For this design of the system, the following requirements must be fulfilled. Firstly, the distances from the axes of the shakers to the axis of the milling spindle and also to the zone of interaction between the cutting edge and the workpiece have to be minimal. This should reduce unwanted torques from the generated forces during system operations. Secondly, the design of actuators must have a minimal length so as not to restrict the mobility of the robot during milling.

3. CUTTING FORCES IN ALUMINIUM 6063 ALLOY MILLING

In this paper, the possibility of using a piezo shaker to compensate for the cutting force when milling the aluminium 6063 alloy will be considered. For this purpose, simulations of the cutting process for a single-edged end mill were performed in the Deform-3D software. The tool model had the following parameters: diameter of cut 14mm; radius of the cutting edge 0.01mm; corner radius 0.5mm; rake angle 5º; helix angle 0º; dish angle 5º. The cutting parameters in the simulations had the following values: axial depth of cut 1 mm; radial depth of cut 1,

2 and 4mm; feed per tooth 0.05, 0.1 and 0.2mm; tool rotational speed 3000, 6000 and 12000rpm.
The results of the simulation showed that the value of the projection of the cutting force along the Z axis (axis of rotation of the tool) is significantly lower and does not exceed 8% of the value of the projections on in X (feed) and Y (normal) axes. Therefore, only the last two projections of the cutting force on the X and Y axes are considered. The direction of the X axis coincided with the feed direction in the simulation model. The direction of the Y axis was perpendicular to the direction of feed.

4. PARAMETERS OF THE PIEZO SHAKER

The investigated parameters of the shaker are the effective mass and the necessary stroke which allow generating the necessary inertial-dynamic force to compensate the cutting force. The inertial-dynamic force while returning the shaker to its initial condition is also an important parameter. Its value should be as small as possible. The effective mass here is the sum of the movable mass of the piezoelectric drive and the attached mass. Motion parameters of the shaker were calculated for following values of the effective mass: 0.3kg, 1kg, 1.7kg and 2.4kg.
In terms of the generation of compensatory force, the movement of the shaker can be divided into active and passive phases. During the active phase the shaker creates an inertial-dynamic force equal in value, but opposite the direction of the cutting force. In the passive phase, the actor is brought to the initial condition before the next tooth of the milling tool engages the workpiece. The passive phase of the shaker movement can be subdivided into three parts: first deceleration (to a complete stop), acceleration in the direction of the initial position and second deceleration. The necessary stroke of the shaker is determined by its position at the end of the first part of the passive phase of motion. Equations of motion were created for all parts of the active and passive phases of motion. The motion parameters there were designated with the following indices: 1 for the active phase, 2.1, 2.2 and 2.3 for the passive phase.

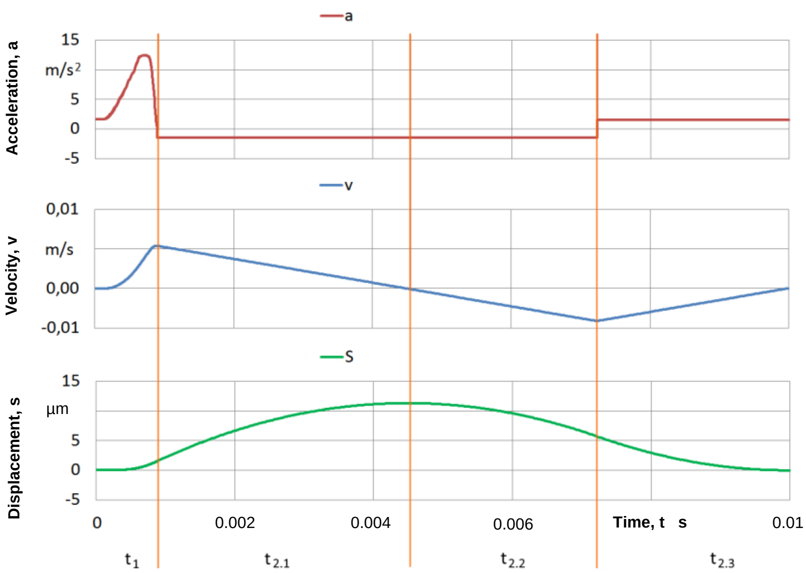

Fig. 3. Motion parameters

The graph Figure 3 shows the motion parameters (a – acceleration, v – velocity, s – displacement) of a shaker with an effective mass of 1kg as a function of time to compensate the X-projection of the cutting force for the following cutting parameters: a_p = 1mm, a_e = 1mm, n = 6000rpm, fz = 0.05mm. It is also shown on this chart, how the shaker's motion phases are divided and designated. The measurements were carried out with a modal analysis system.
Let's consider the active phase of motion. First the relation between acceleration along the X and Y axes and time was determined for each of the aforementioned effective masses, on the basis of Newton's second law, based on equation (1).

$$a_1(t) = -F_1(t)/m \tag{1}$$

Then, the dependencies between velocity and displacement of the effective mass along the corresponding axis and time were identified by numerical integration according to formulas (2).

$$\begin{aligned} v_1(t) &= \int_0^{t_1} a_1(t)dt \\ s_1(t) &= \int_0^{t_1} v_1(t)dt \end{aligned} \tag{2}$$

The following equations (see formulas 3-9) were created accordingly for the motion of the shaker in the passive phase.

$$a_{2.1} = v_{0\_2.1} t_{2.1} \tag{3}$$

$$s_{2.1}(t) = \frac{a_{2.1} t^2}{2} + v_{0\_2.1} t + s_{0\_2.1} \tag{4}$$

$$s_{2.2}(t) = \frac{a_{2.2} t_{2.2}^{\ 2}}{2} + s_{0\_2.2} \tag{5}$$

$$s_{2.3}(t) = \frac{a_{2.3} t_{2.3}^{\ 2}}{2} + s_{0\_2.3} \tag{6}$$

$$t_1 + t_{2.1} + t_{2.2} + t_{2.3} = 1/n \tag{7}$$

$$|a_{2.1}| = |a_{2.2}| = |a_{2.3}| \tag{8}$$

$$t_{2.2} = t_{2.3} \tag{9}$$

Where the time t_1 corresponds to the duration of cut in the simulation. When solving the equations, the following condition was assumed: the accelerations for all parts of the passive phase are equal in absolute value. This condition allows to perform braking and return to the initial condition of the shaker with a minimal inertial-dynamic force. Also, the durations of the second and third part of the passive phase were assumed as equal.

The motion parameters resulting from the system of equations (3 – 9) allow to determine the necessary displacement of the shaker as a function of time. The control voltage signal for the shaker could be created based on this function. Also the inertial force in the passive phase of the shaker's motion was determined on the basis of Newton's second law (formula 10).

$$F_p = a_{2.1} m \tag{10}$$

5. RESULTS OF CALCULATIONS AND EXPERIMENTS

A graph of the necessary stroke to create inertial forces to compensate the X and Y projections of the cutting force for the tool rotational speed 6000min^{-1} and the radial depth of cut 1mm, depending on the effective mass of the shaker and for different feed rates per tooth (index 1 – for 0.05mm, 2 – for 0.1mm and 3 – for 0.2mm) is shown on the Figure 4.

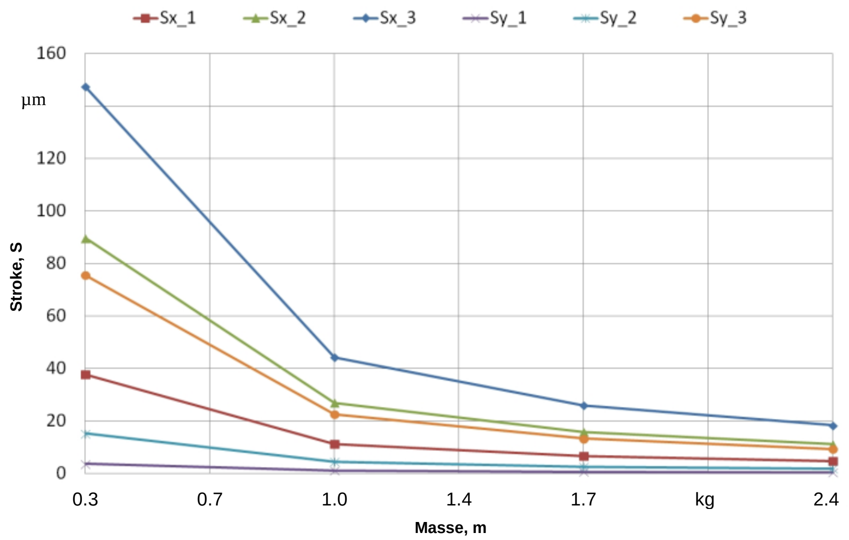

Fig. 4. The necessary stroke depending on the effective mass

The chart above shows that the amplitude of the necessary stroke of the shaker can be reduced by increasing its effective mass.

The following graphs allow us to estimate the influence of the tool rotational speed on the necessary stroke (Figure 5) and the inertial force during the passive phase of motion (Figure 6) of the shaker with the effective mass of 1kg for milling with the feed rates per tooth of 0.1mm for different radial depths of cut (index 1 – for 1mm, 2 – for 2mm and 3 – for 4mm).

These graphs show that piezo shakers with a smaller stroke can be used for higher tool rotational speeds. At the same time, with increasing tool rotational speeds, the value of the inertial force during the passive phase of motion increases significantly, which is undesirable.

When selecting the parameters of the piezo drive, it should also be taken into account that increasing the stroke of the drive also increases the overall dimensions of the shaker and at the same time lowers its maximum frequency of operation [10].

The graph below (Figure 7) shows the experimental results of the generation of the inertial-dynamic force F_{exp} and the calculated force F_{calc} as functions of time.

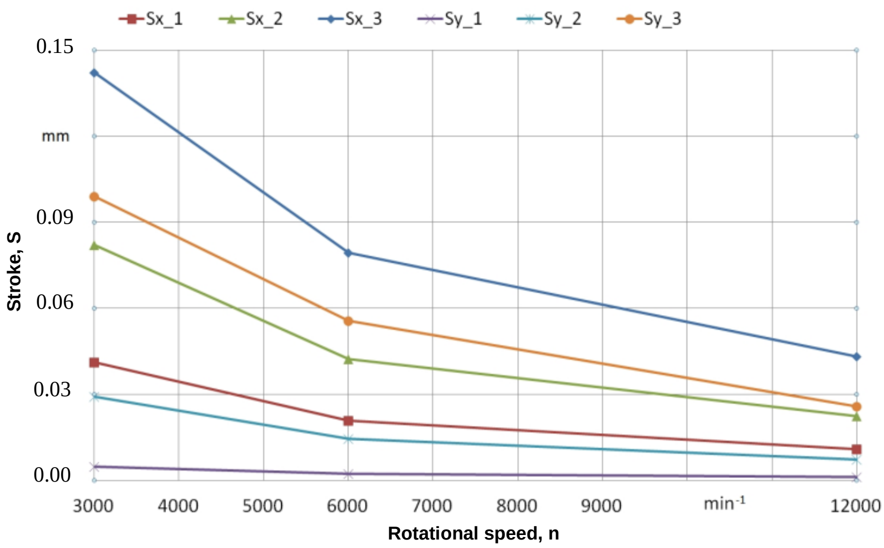

Fig. 5. The necessary stroke depending on the tool rotational speed

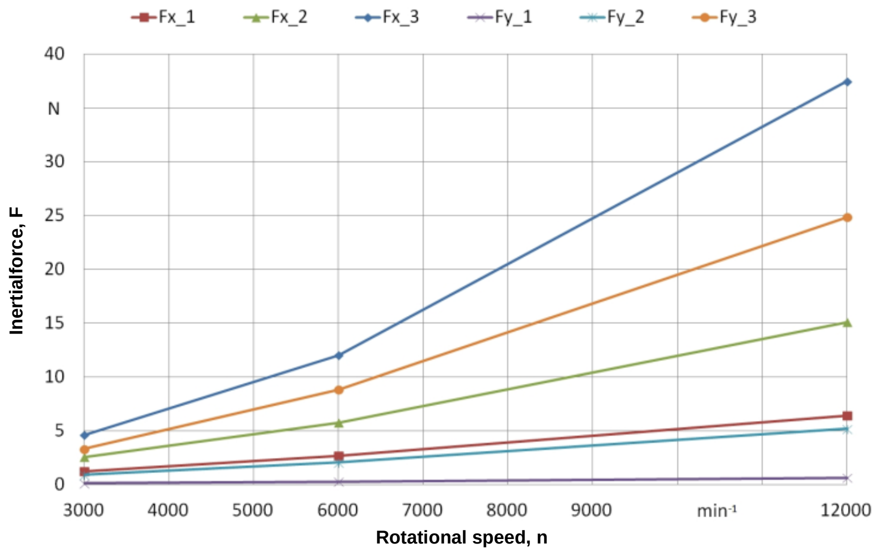

Fig. 6. Passive inertial force depending on the tool rotational speed

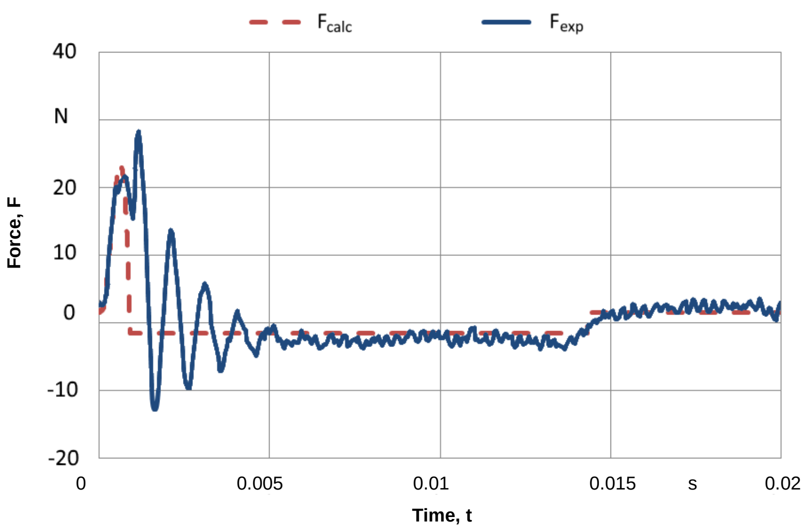

Fig. 7. Generated inertial-dynamic force

The force F_{calc} was calculated for a shaker with the effective mass of 0.74kg to compensate for the X-projection of the cutting force in milling for the following cutting parameters: $a_p = 1mm$, $a_e = 1mm$, $n = 3000rpm$, $f_z = 0.1mm$. In this experiment, a piezo actuator PI 216.K040 (with a working stroke of 200 microns) and an E-482 amplifier by Physik Instrumente were used. The magnitude of the generated force was measured with a dynamometer type 9272 by Kistler.

The experimental results show that the self-oscillations of the shaker arise during the generation of the inertial-dynamic forces when using the control signals described in this paper. These oscillations are caused by the complex influence of mechanical and electrical processes in the shaker. The oscillations reach maximum amplitude when the actuator moves from the active to the passive phase of motion at that time, the force decrease reaches its maximum value. As a result, the generated forces deviate significantly from the calculated ones.

6. CONCLUSIONS

This paper investigated the possibility of using piezo drives to compensate for the components of cutting forces in milling processes in the feed and the normal directions. It showed the influence of the milling parameters (width of cut, feed per tooth, rotational speed) and the effective mass of the shaker on the value of the necessary stroke of the piezo shaker and on the inertial force during the passive phase of its movement.

The proposed approach was based on the generation of the inertial-dynamic forces, which correspond exactly to cutting forces in milling. The self-oscillations of the shaker cause deviations of the values of the generated forces from the desired values. Therefore, it can only be applied if measures are taken to reduce the self-oscillations of the shakers during the generation of inertial forces. Hence, it is necessary to develop a new method of controlling piezo actuator movement during force generation. Moreover, there are limitations to the application of this approach, depending on the tool rotational speed. This is due to the fact that the inertial-dynamic forces increase in the passive phase of the shaker movement with increasing tool rotational speed.

7. ACKNOWLEDGEMENTS

The research work reported here was supported by the Deutsche Forschungsgemeinschaft (German Research Foundation).

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Received: November 01, 2018 / Accepted: June 15, 2019 / Paper available online: June 20, 2019 © International Journal of Modern Manufacturing Technologies.