444. Reduction of spindle vibrations in milling machine by active magnetic bearing

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Abstract. In this study, a three-dimensional dynamic model of a milling machine is proposed. The cutting forces of the face milling process were obtained according to the cutting parameters by means of computer simulations and experiment. The cutting forces excited the dynamic model of the system. Relative displacements of the contact point of the cutting tool and the workpiece were obtained by using forced vibration analysis. These displacements affected machining accuracy of the milling machine. Therefore, radial and axial electromagnetic bearings were designed for the active control of the system and they were adapted on the spindle of the milling machine. Thereby the electromagnetic force produced around the rotating spindle reduced vibration amplitude of the cutting tool. The system was operated with and without active control and both these cases were compared. It was revealed that active control diminished cutting tool vibrations and improved machining performance.

Keywords: vibration control, Machine tool, Spindle – Magnetic bearing design

Introduction

Machining is a very important production method for precision parts, which is commonly used in manufacturing industry. A lot of studies were devoted for increasing machine tool performance on cutting tools. So, the cutting tools, which were appropriate for high speeds were manufactured and cutting capacities of cutting tools were increased. Also, the developed technologies and control systems make possible to control machine tools precisely. However, vibrations which are very important problem between cutting tool and workpiece are generated and it reduces machining accuracy. This problem is particularly noteworthy in milling machines because of milling process cutting forces, which are inevitable and have variable characteristics. The response of the system for this kind of cutting operation affects cutting process and machining accuracy. A lot of studies were performed to solve this problem [1-3]. They consider the selection of optimal parameters and increasing of mechanical stiffness. But the optimization of the cutting parameters is inadequate because chatter arises for high cutting capacities. Therefore, studies were conducted in order to obtain new approaches regarding this problem [4,5]. At the same time, a stiffer structure means a larger section area for machine tool elements. This case is not desirable solution since the developed technology recommends lightness. Furthermore, machine tool bodies have a sufficient stiffness and they do not cause problem. The origin of problem is related to spindle vibrations. That is, the cutting forces cause the deflections of the cutting tool and this aggravatess machining accuracy. If the spindle vibrations are controlled, the problem can be significantly alleviated. Therefore, researches focused their investigations on this direction of the subject. Dohner J.L et al presented the experimental validation of the active control approach for mitigation of chatter in milling process. They implemented an active control system using magnetic bearings surrounding a spindle and cutting tool and raised the stability lobes of the machine tool [6]. Yousefi R. and Ichida Y. investigated effects of active magnetic bearing on surface roughness while machine tool was machining with ultra high speed [7]. Shhroder P. et al developed a genetic algorithm for the control of an active magnetic bearing. They compared new algorithm with conventional control algorithms and obtained improvements considerably [8]. Also, Cole M.O.T. et al developed a control algorithm. Their study was undertaken of possible system failure modes which are classified according to whether they are internal or external to magnetic bearing control system [9]. Ji J.C. and Hansen C.H. studied non-linear oscillations of a rotor in active magnetic bearing. They performed numerical simulations to verify the analytical predictions [10]. Zhang X. et al focused on improving the rotational accuracy of the magnetic bearing under the consideration of the unbalanced force of the spindle and the unbalanced magnetic pull of the motor [11]. As seen in most of the studies, magnetic damper was controlled measuring displacements of the spindle on-line. This case requires both expensive and high-precision equipment. Also there are some practical difficulties and problems in high-speed operations [6].

In this study, the control has been realized into open loop and as model-based. Therefore, the system does not need feedback and it is cheaper than the other systems. Therefore, it is expected that the system can be used widely for all types of machine tools in industry. The goal of this study is to reduce dynamic effects of the cutting forces on the spindle and enhance cutting performance by means of an Active Magnetic Bearing (AMB). The cutting forces were determined using both experimental and numerical approaches. The electromagnetic attraction forces were produced by the AMB according to the cutting forces. The study presents the model of the cutting forces, the experimental setup for measurement of cutting forces, the design of AMB and control system, the dynamic model of machine tool and directions for future work.

The model of the cutting system

In this study, the asymmetric face milling was considered for the model of the cutting forces. The cutting force components between the cutting tool and the work piece are as follows [12]:

$$F_T = k_T \times A \qquad (1)$$

$$F_R = k_R \times F_T \qquad (2)$$

$$F_A = k_A \times F_T \qquad (3)$$

These forces correspond to one insert of the cutting tool. If these cutting force components transform Cartesian coordinate (see Fig. 1), the cutting force components in x, y and z directions can be obtained:

$$F_x = -F_T \ Sin\theta_i + F_R \ Cos\theta_i \qquad (4)$$

$$F_y = F_T \ Cos\theta_i + F_R \ Sin\theta_i \qquad (5)$$

$$F_z(i,\phi) = F_A(i,\phi) \qquad (6)$$

However, two or more inserts can cut simultaneously. In these cases, the cutting force components are:

$$\begin{Bmatrix} F_x(\phi) \\ F_y(\phi) \\ F_z(\phi) \end{Bmatrix} = \sum_{i=1}^{Z_c} \begin{bmatrix} -Sin(\theta_i(\phi)) & Cos(\theta_i(\theta)) & 0 \\ Cos(\theta_i(\phi)) & Sin(\theta_i(\phi)) & 0 \\ 0 & 0 & 1 \end{bmatrix} \begin{Bmatrix} F_T(i,\phi) \\ F_R(i,\phi) \\ F_A(i,\phi) \end{Bmatrix} \quad (7)$$

Where F_T, F_R and F_A are tangential, radial and axial forces respectively.

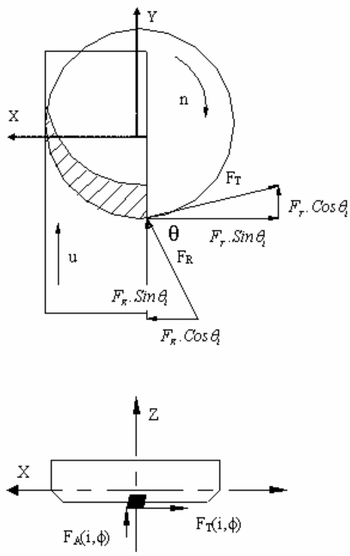


Fig. 1. Cutting force components

These cutting force components have variable characteristic. So the cutting system (workpiece – cutting tool – spindle) is affected by forced vibrations. Therefore, the relative displacement between the cutting tool and the workpiece takes place and thereby new cutting force components appear. The extra forces have been included to the components of the cutting force because of dynamic effects. So, the ultimate force components which affect the machine tool structure can be determined as follows [12]:

$$\begin{Bmatrix} DF_x(\phi) \\ DF_y(\phi) \\ DF_z(\phi) \end{Bmatrix} = \begin{Bmatrix} F_x(\phi) \\ F_y(\phi) \\ F_z(\phi) \end{Bmatrix} + \begin{Bmatrix} dF_x(\phi) \\ dF_y(\phi) \\ dF_z(\phi) \end{Bmatrix} \quad (8)$$

$dF_x(\phi)$, $dF_y(\phi)$ and $dF_z(\phi)$ are X direction force component of dynamic force at an angle ϕ, Y direction force component of dynamic force at an angle ϕ and Z direction force component of dynamic force at an angle ϕ. The computer simulation of the cutting process has been prepared by using these forces components. The cutting forces components can be obtained by the simulation of the cutting process resulting in numerical values and graphical form.

The experiment for the cutting force

An experimental setup has been built so that the model of cutting forces can be verified. The experimental equipment consists of a milling machine, a force transducer that has measuring capability in x, y and z directions, amplifiers, A/D converter and data acquisition card. The actual view and the schematic diagram of the experimental setup are presented in Fig. 2 and Fig. 3 respectively.

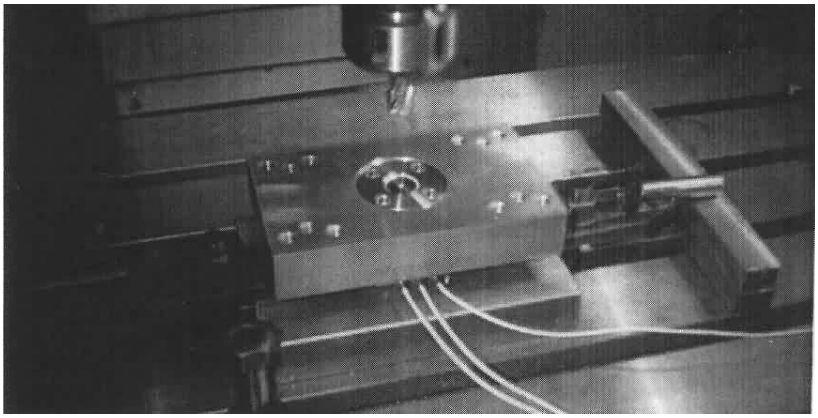


Fig. 2. Perspective view of the experimental set – up

The cutting experiments have been performed under the experimental conditions listed in Table 1.

Table 1. The cutting parameters

Cutting Parameters	Value
Workpiece	St 50
Spindle velocity	450 rpm
Uncutting thickness (D)	0.5 mm
Feed rate (f_t)	0.074 mm/insert
Workpiece wide (B)	27 mm
Cutting tool diameter (D)	50 mm
Insert number (Z_n)	3
Radial rake angle (γ_R)	14^o
Axial rake angle (γ_A)	8^o
Lead angle (γ_L)	20^0
Specific cutting pressure (k_T)	400
A dimensionless constant relating radial force to tangential force (k_R)	0.375
A dimensionless constant relating axial force to tangential force (k_A)	0.67

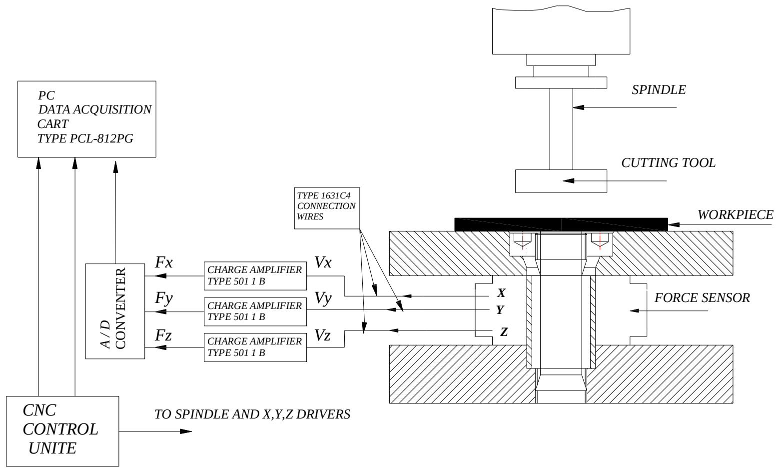


Fig. 3. Schematic diagram of the experimental setup.

Experimental cutting forces in x, y and z directions have been presented together with the theoretical results in Figs. 4-6. As these figures indicate the experimental and the simulated results are in good agreement. The results demonstrate that the theoretical cutting force values can be used to control AMB.

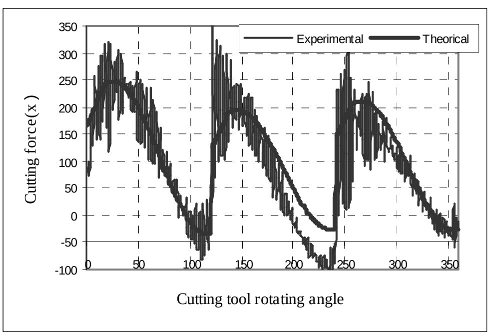


Fig. 4. Experimental and simulated cutting force component in x direction

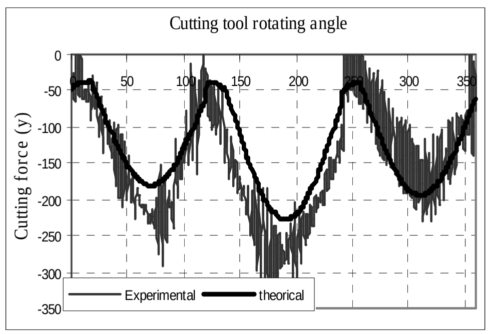


Fig. 5 Experimental and simulated cutting force components in y direction

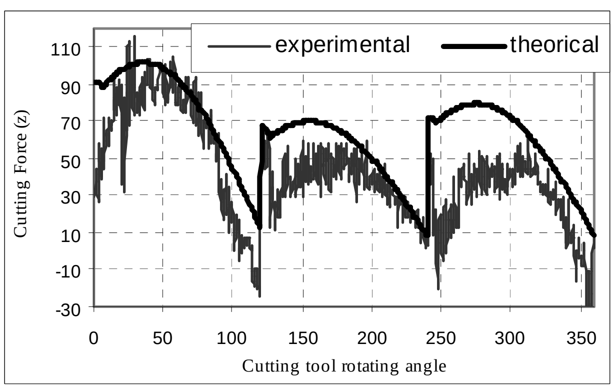


Fig. 6 Experimental and simulated cutting force components in z direction

Design of AMB and control system

Dynamic cutting forces cause spindle vibrations and cutting tool deflections resulting in lower machining accuracy of machine tool. Therefore, the spindle vibrations must be controlled. Spindle vibrations is a very common problem in machines and so far many active and passive control systems were used to reduce these vibrations. AMB that is very suitable for adaptive control and high speeds has been widely used as well. In this study, AMB has been used for reduction of spindle vibrations. Design of AMB and its control have been considered together. Here, a new approach that is based on the model of cutting force components has been presented for the control of AMB. This approach is not based on displacements of the spindle and this constitutes its distinction in comparison to existing approaches. This study has resulted in development of a useful new approach because there are problems while displacement sensors are settled and vibrations are measured at high speed. The configuration of the AMB is presented in Fig. 7 and the system diagram with AMB designed for the present study is given in Fig. 8.

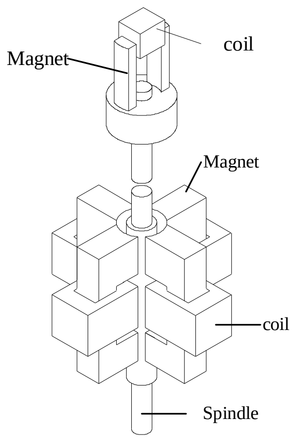


Fig. 7. Configuration of active magnetic bearing

Each power amplifier (A) that drivers a magnet coil (C) is of the switching type using pulse-width modulation to reduce power losses. There are two electromagnets, one of them being radial and the other - axial. The former consists of four radial poles and the latter has one pole. The electromagnet exerts a force approximately proportional to the square of the magnet flux presented at the pole face. It is also unstable in an open-loop mode since, as the deflection increases towards a magnet, so does the attractive force. The system can, however, be made stable feeding back a signal (V_H) by a Hall probe, which is proportional to the flux at the pole face. The balancing of the axial cutting force components in (z) direction is easier that the radial cutting force components (x) and (y) directions. Because, it is sufficient that an axial force which has the same magnitude with cutting force component in (z) direction is produced by magnetic bearing and is applied on the spindle in opposite direction of cutting force component in (z) direction. But, the magnitudes of the balancing radial forces in (x) and (y) direction are variable according to the position of the magnetic bearing on the spindle, because the radial cutting forces affect the system with bending moment on the spindle. Therefore, the bending moments must be balanced by electromagnetic bearing, i.e. the balancing forces of the electromagnetic bearing must be adjusted according to cutting force components and the electromagnetic bearing position on the spindle so that the effect of bending moment can be eliminated. This case is achieved by the controller. The electromagnetic force produced by electromagnets can be obtained as reported in [13].

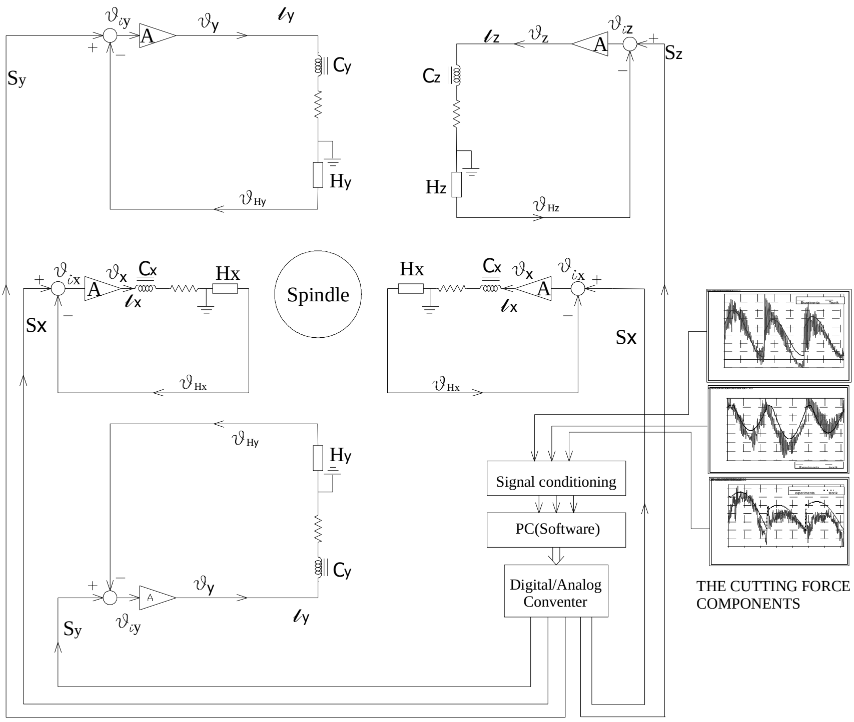


Fig. 8. Schematic plot of the AMB system

For any given coil and its series resistance, (R), the supply voltage, (V), inductance, (L), is given by:

$$V = L\frac{di}{dt} + iR \qquad (9)$$

Where (i) is the current passing thought the coil. Thus:

$$\frac{I}{V} = \frac{1/R}{\mathrm{L}(L/R)+1} \qquad (10)$$

Now, consider the electromagnet in (x) direction:

$$\vartheta_{ix} = S_x - \vartheta_{Hx} \qquad (11)$$

Where Sx is signal in D/A converter output and calculated by PC for the spindle stability. ϑix is the input voltage to the corresponding amplifier and ϑHx is proportional to the flux ϕ_x at the probe face. This flux is also inversely proportional to the gap Zx between the magnet face and the spindle while the flux is proportional to the current ix in the magnet coil. So, $\vartheta_{Hx} = K_x \phi_x$ and:

$$\vartheta_{ix} = S_x - K_x \phi_x = S_x - K_x K \frac{i_x}{Z_x} \qquad (12)$$

$$\vartheta_{ix} = S_x - \frac{K_x K}{Z_x} \frac{\vartheta_x / R_x}{(\mathrm{L}\, L_x / R_x + 1)} \qquad (13)$$

$$\vartheta_x = a\,\vartheta_{ix} \qquad (14)$$

Where (a) is the amplifier gain. Hence:

$$\vartheta_{ix} = S_x \Big/ \left[1 + \frac{K_x K \;\; a \;\; / R_x}{Z_x (\mathrm{L} L_x / R_x + 1)} \right] \qquad (15)$$

if equation (15) is replaced in equation (12)

$$\phi_x = S_x \Big/ \left[K_x + \frac{(\mathrm{L} L_x / R_x + 1) Z_x}{a \;\; K / R_x} \right] \qquad (16)$$

Hence, if (a) is large enough,

$$\phi_x \cong S_x / K_x \qquad (17)$$

The attraction force produced by the magnets can be obtained as follows:

$$F'_x = \frac{\phi_x^2}{2\mu A} \qquad (18)$$

Where μ is magnetic permeability coefficient (μ=μ0.μr=4π.10-7) and A is the cross-section of magnetic flux route. If equation (17) is replaced in equation (18):

$$F'_x \cong \frac{Sx^2}{K_x^2 \;\; 2\mu A} \qquad (19)$$

From this equation it is obvious that the attraction force (F'x) is proportion to the square of Sx because Kx, μ and A are constant parameters. Similar equations can be used for y and z axis. The attraction forces obtained by the system simulation have been presented in Fig. 9.

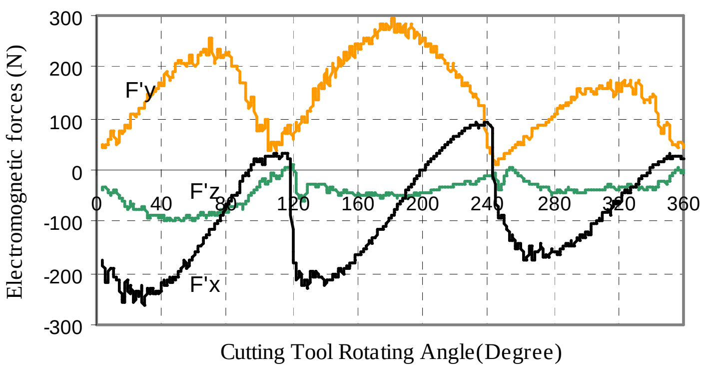


Fig. 9. Electromagnetic forces of AMB

Dynamic model of the system

Dynamic model of the system which consists of the spindle, cutting tool and the workpiece was built using mass, spring and damper elements. The model is 3D and has 6 degree of freedom (Fig. 10).

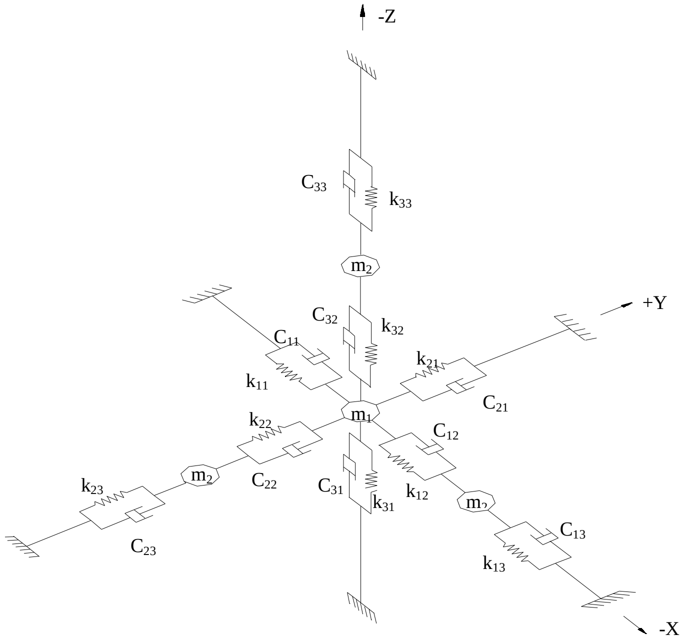


Fig. 10. Dynamic model of the system

The model was excited by the cutting force components that were applied on mass m_1. The compensating electromagnetic forces were applied on mass m_2. The equation of motion of the model is:

$$[M]\{\ddot{U}\}+[C]\{\dot{U}\}+[K]\{U\}=\{F\} \tag{20}$$

$$\{F(t)\}=\begin{Bmatrix} F_X \\ 0 \\ F_Y \\ 0 \\ F_Z \\ 0 \end{Bmatrix} \tag{21}$$

$$\{F^{'}(t)\}=\begin{Bmatrix} F_x \\ F_x^{'} \\ F_y \\ F_y^{'} \\ F_z \\ F_z^{'} \end{Bmatrix} \tag{22}$$

where [M], [C], [K] are mass, damping and stiffness matrices respectively , and vectors $\{U\}$, $\{\dot{U}\}$, $\{\ddot{U}\}$and $\{F\}$ are the displacements, velocities, accelerations and forces, respectively. {F(t)} is the external force without electromagnetic bearing and $\{F^{'}(t)\}$ is external force with electromagnetic bearing.

In this model m_1 denotes the mass of cutting tool and workpiece while m_2 denotes the mass of the spindle. The performance of the machine tool can be analyzed if this model is integrated with the model of the cutting force and the active magnetic bearing. The relative displacement in (z) direction between the cutting tool and workpiece that is the displacement of m_1 mass affects the surface quality of the workpiece directly whereas the displacements in (x) direction and (y) direction have no effect on the surface quality. Therefore, the surface roughness can be reduced to the relative displacement in (z) direction if it is assumed that all of the other parameters are constant. The block diagram of the integrated system designed for this reason is presented in Fig. 11.

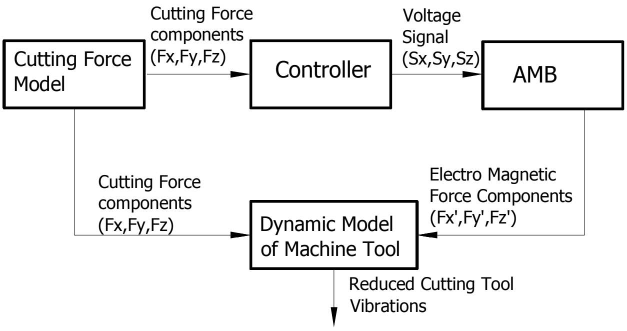
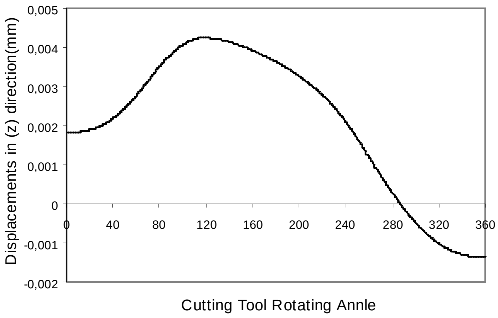
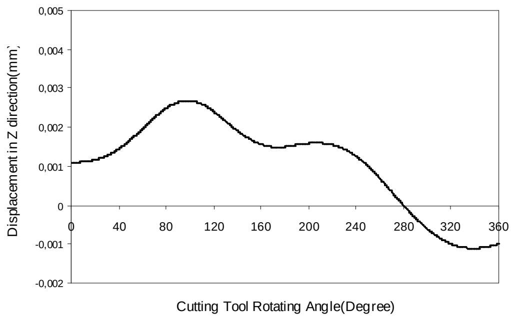


Fig. 11. Open-loop control of dynamic simulation of cutting system

The relative displacements in (z) direction (surface roughness) without the AMB and with AMB are presented in Fig. 12. There is a pronounced positive influence of AMB on dynamic performance of the cutting process as evident in Fig. 12. The relative displacement in (z) direction, which is surface roughness, is reduced approximately by 40% when AMB is active in the system.





Displacements without electromagnetic bearing

Displacements with electromagnetic bearing

Fig. 12. Relative displacements in (z) direction (surface roughness)

Conclusions and future work

In this study, cutting tests are realized for specific cutting parameters on a face milling machine. Furthermore, a theoretical model of cutting force was developed using the same cutting conditions. Experimental and theoretical results were compared thereby verifying the model. In addition a structural dynamic model of the milling machine was built considering structural parameters. A theoretical AMB was designed so that the spindle vibrations can be controlled. Finally, the simulation of milling process was conducted integrating dynamic model of the machine tool, the model of the cutting force and AMB. The results of the system obtained with and without AMB were compared revealing that AMB improves dynamic performance of the system considerably and results in better surface precision of the products. In the similar systems, AMB works only when cutting operation passes chatter limit, i.e. the deflection of the spindle reaches a certain value. Whereas in the present system AMB works during machine tool operation and prevents chatter generation as well as improves machining quality continuously. The structural parameters of the machine tool have constant values in general and so the cutting force variations depend only on cutting parameters, which can be sent to the controller of electromagnetic system from the control unit of the machine tool directly and the controller transmits voltage signal to AMB. Thus, AMB can be controlled without measuring relative displacements between cutting tool and workpiece. This is a distinctive quality of the proposed approach. This system needs less equipment in comparison to similar systems and therefore is less costly. We expect that the system will be extensively used and will contribute to increase of quality and productivity. Future studies include validation of the results of the dynamic performance using the system with AMB. Therefore the AMB designed for this study will be produced and tested on the actual machine tool under the same conditions.

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