UDC: 621.941.01:534.1
DOI: 10.20535/2521-1943.2019.86.185849

Chatter suppression technologies for metal cutting

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Received: 14 June 2019 / Accepted: 05 September 2019

Background. *The cutting process is carried out in a closed elastic technological machining system and is always accompanied by vibrations. Vibrations arising during cutting, depending on the amplitude, can very slightly affect the machining result, and can lead to a catastrophic loss of stability of the whole process. In any case, all researchers agree that vibration is the factor that ultimately determines the productivity of the cutting process and the quality of the machined surface.*
Objective. *The aim of this study is to develop new technologies for selecting parameters for controlling the cutting speed to suppress chatter by passive methods, as well as to control the drive of the forming motion to suppress chatter by active methods.*
Methods. *The goal is achieved by creating new technologies aimed at the study of dynamic processes occurring in the cutting. It is noted that the mathematical model of the cutting process should be built taking into account the loop closed of the elastic technological machining system and the function of the delayed argument, which represents machining "on the trail". When studying the cutting process, four main groups of factors that influence its mathematical representation are taken into account, and three approaches are used to determine the stability diagram: frequency analysis, root analysis of the characteristic equation of motion of the system and the numerical method. The numerical method using the amplitude-frequency characteristics according to the corresponding stability criterion is considered to be the most effective.*
Results. *The results of theoretical studies are used in practice in the form of technologies for passive and active chatter reduction during cutting. A technology has been developed to suppress vibrations during face milling when controlling the spindle speed according to a harmonic law. An application program for simulating a process for determining the parameters of the control law is presented. For active control, a new technology is proposed, based on the use of a CNC machine drive with an additional closed system, introducing a harmonious signal into the channel of the shaping movement, the amplitude and phase of which are automatically adjusted using the coordinate-wise descent algorithm according to the criterion of the minimum amplitude of the motor current.*
Conclusions. *The technology of chatter suppression during face milling by controlling the spindle speed according to the harmonic law is limited by the speed of the spindle drive and its inertial characteristics. The active chatter control system uses a standard servo drive of the CNC machine, which has an additional closed loop for automatically searching for the amplitude and phase of the compensating control signal.*
Keywords: *chatter, cutting process, mathematical model, stability of the cutting process.*

Introduction

The cutting process is always carried out in an elastic closed technological machining system (TMS) and is necessarily accompanied by vibrations. The loop closure of TMS is manifested in the fact that the cutting force, which depends on the cutting mode, causes elastic deformations that change the parameters of the removed allowance layer, and this, in turn, changes the cutting force and so on.

Among the reasons for the self-oscillations, the nonlinearity of the cutting force characteristics is noted, the structure loop of dynamic system and the reason which the researchers rightly consider come main is the machining "along the trail".

The vibrations arising from cutting, depending on the amplitude, can have very little effect on the result of the machining, but can lead to a catastrophic loss of stability of the entire process. In any case, all researchers agree that chatter is the factor that ultimately determines the productivity of the cutting process and the quality of the machined surface.

Now it can be stated that the development of the science of cutting metals has reached a level where, in combination with the theoretical knowledge of modeling with modern computer technology with extensive process

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control capabilities provided by CNC machines, there are opportunities to control such a phenomenon. Therefore, recently, the number of studies devoted to the development of technologies capable of anticipating, identifying and suppressing chatter during cutting has sharply increased [1].

The purpose of such technologies is theoretical development to create mathematical models that can adequately represent cutting processes, on the basis of which practical applications are implemented to eliminate vibrations for various types of processing: turning, milling, grinding, etc. However, creating of adequate mathematical models and convenient criteria for predicting the vibration stability of the cutting process, which will allow you to create effective technologies for eliminating chatter, it is still relevant.

Analysis of literature data and problem statement

All researchers mainly distinguish three types of vibrations depending on the reasons for their occurrence: free vibrations of the system, forced vibrations and self-excited (regenerative) vibrations. When building a mathematical model of a process, four groups of factors are mainly taken into account, and a mathematical model is built with one, two, three or four degrees of freedom in the single-mass dynamic model representation [2, 3, 4].

The study on stability with the aim of building a stable area (the so-called "stability lobe diagram") in the coordinate depth (width) of cutting – cutting speed is performed by three methods (Fig. 1). The results of such studies provide the ability to assign cutting conditions that reduce the likelihood of vibrations during cutting. The use of numerical methods in the study of the model with the function of lateness in combination with the determination of stability by frequency criteria gives real results [5].

In practice, methods for eliminating vibration during cutting can be divided into two directions: passive (Passive Chatter Control – PCC) and active (Active Chatter Control - ACC).

The methods of passive control include various types of dynamic vibration compensators, which are performed constructively as additional masses attached to the element of the machine, which directly performs the cutting process, through an elastic connection with damping. Sometimes the parameters of such devices can automatically adjust, adapt to the actual cutting conditions (Adaptive Turning Mass Damper – ATMD). In addition, passive methods also include controlling the frequency of rotation of the machine spindle (Spindle Speed Variation – SSV), using tools, usually milling cutters, with variable tooth pitch, etc. [7, 8].

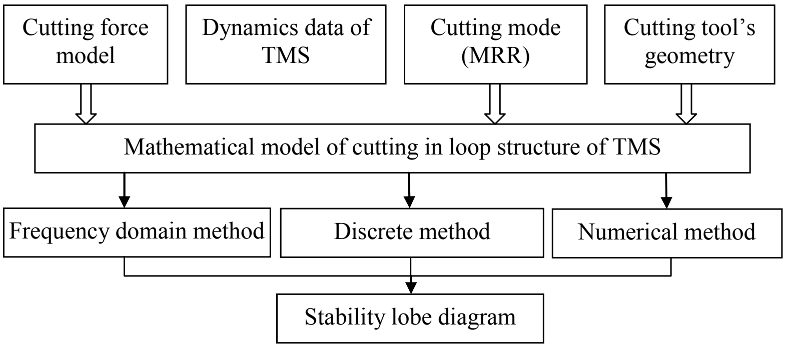


Fig. 1. The structure of the research stability of the cutting process

Here, it should be noted studies aimed at creating methods for the rational choice of parameters for harmonic modulation of the spindle speed of a lathe [9]. A program for modeling the turning process is proposed, in which you can select the amplitude and period of the harmonic signal modulating the spindle speed. Similar studies have been performed for the milling process [10]. It is noted that the choice of the amplitude and frequency parameter of the superimposed signal is one of the unsolved problems. Some research is being carried out in this area [11], although there is no exhaustive work on the experimental verification of their statements.

Recently, technologies for actively eliminating chatter are rapidly being introduced. All such technologies, one way or another, are based on the use of the principles of automatic control of feedback systems, when the signal from the vibration sensor is used to suppress the vibration itself. Among them, one can single out the use of piezoelectric and magnetostrictive drives, vibrational magnetic devices mounted on the spindle. It is noted that such technologies make it possible to increase the cutting depth by 50% [13, 14].

The purpose and objectives of the study

The aim of this study is to develop new technologies for selecting cutting speed control parameters for vibration suppression by passive methods (Passive Chatter Control), as well as to control the drive of a forming motion to suppress vibration by active methods (Active Chatter Control).

Passive Chatter Control

As already noted, the main task when using spindle speed control as a passive method of vibration suppression is to determine the parameters of the cutting speed control law. A technique is proposed for solving the problem, based on numerical modeling of the face milling process with modulation of the cutting speed.

The Mathematical model. Considering that control of the cutting speed changes the character of perturbations affecting the elastic TOC in time, the influence of such control on the excitation of vibrations can be established by a mathematical model. For this purpose, it suffices to consider a model in the form of a single – mass system with one degree of freedom in the direction of longitudinal feed [15]. In the functional diagram (Fig. 2, *a*), the cutting process is represented by a block, at the input of which the actual feed per tooth $(F_z)_a$, and at the output – the horizontal component P_x of the cutting force along the X axis. The component P_x causes elastic displacement δF_z, which, due to the closed loop of the TMS, provokes a change in the actual feed per the cutter tooth: $(F_z)_a = (F_z)_1 - \delta F_z$.

The peripheral component of the cutting force on one tooth can be determined by the formula:

$$P_o = C_p (F_z)^{1-k} (H)^{1-\mu} \sin^{1-k} \varphi, \tag{1}$$

where C_p, k and μ are the coefficient and exponents that depend on the mechanical properties of the material being processed, F_z is the actual feed to the tooth, H is the actual cutting depth, φ is the angular coordinate of the tooth.

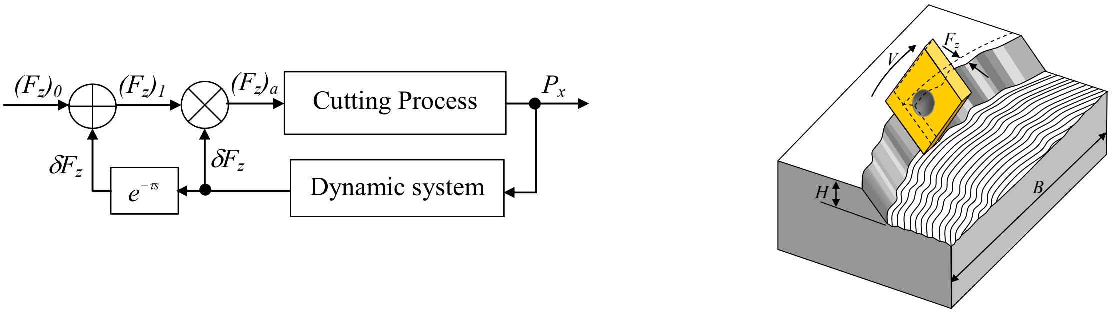
The radial component $P_r \approx 0.6P_o$, of the cutting force, which acts in the direction of the X axis, is determined from the geometric relationships of the location of the vectors of the components of the cutting force:

$$P_x = P_r + P_o = P_o (\cos \phi + 0,6 \sin \varphi). \tag{2}$$

In addition, it is necessary to take into account the machining “on the trail” when each next cutter tooth cuts off the allowance layer that was formed by the previous tooth (see Fig. 2, b):

$$(F_z)_a = (F_z)_0 - \delta F_z (1 - e^{-\tau s}), \tag{3}$$

where $(F_z)_0$ is the preset feed per cutter tooth, τ is the passage time of the angle between two adjacent teeth, s is the Laplace operator.



a *b*

Fig. 2. Face milling process: *a* - functional diagram, *b* - machining “on the trail”

The mathematical model of the motion of a dynamic system according to Fig. 2, *a* can be represented in a transfer function:

$$W(s) = \frac{\delta F_z(s)}{P_x(s)} = \frac{1/C}{T^2 s^2 + 2\xi T s + 1}, \tag{4}$$

where T is the period of self-oscillations, ξ is the damping coefficient of oscillations.

The Technology for cutting speed control. The control of cutting speed according to the harmonic law (Sinusoidal Spindle Speed Variation) is applied:

$$V(t) = V_0 + \delta V Cos(2\pi t / T), \tag{5}$$

where V_0 is the specified cutting speed, δV is the amplitude of the change in speed, T is the period of change.

This law of change in cutting speed is provided by a corresponding change in the frequency of rotation of the tool spindle of the milling machine:

$$\omega(t) = 2000V(t)/(60D_m), \qquad (6)$$

where D_m is the diameter of the mill. Note that, according to the dimensions adopted in the theory of cutting in equation (6), the cutting speed has a dimension of *m/min* and the cutter diameter is *mm*, so the rotation frequency will have a dimension of *rad/s*.

A change in the rotational speed of the cutter affects the actual feed to the mill tooth:

$$F_z(t) = \frac{F_0 2\pi}{Z\omega(t)60}, \qquad (7)$$

where F_0 is the feedrate specified in the control program (*mm/min*), Z is the number of teeth.

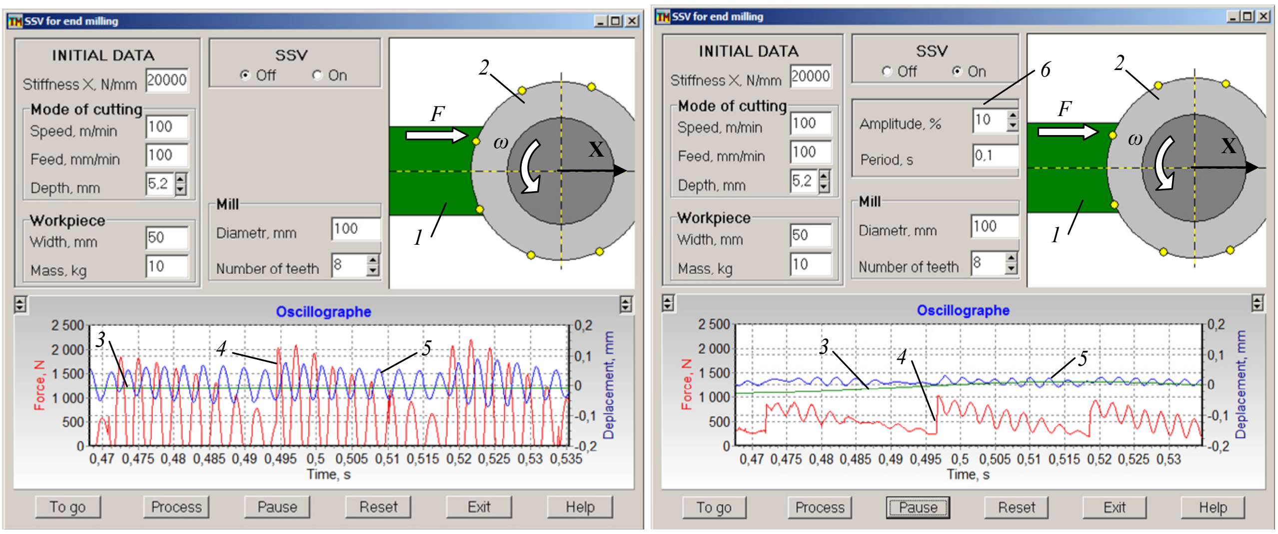
Therefore, a change in cutting speed affects not only the frequency of the pulses from the cutting force, but also the parameters of the allowance cut by each tooth. Thus, the wavy trace on the treated surface from the vibrations of the previous tooth changes not only its amplitude along and the location along the length of this surface. Therefore, this control eliminates fluctuations caused by monotonously repeating allowance changes at a constant cutting speed. Obviously, the control efficiency will depend on the given amplitude and period of fluctuations in the cutting speed.

It is not possible to solve the problem of determining the optimal parameters of the control law in an analytical way, since the control acts on a non-linear dynamic system, which is described by a differential equation with a delayed argument (see Fig. 2, *a*).

The Simulation. Numerical modeling methods that use the 4th-order Runge-Kutta procedure to integrate the differential equation of the system, as well as the numerical procedure for determining the parameters of the geometric and force interaction of the cutter tooth with the machined surface, taking into account the trace from the previous tooth, allow us to solve the problem.

Fig. 3 shows the interface of the created simulation program. The parameters of the dynamic model as a single-mass system, cutting mode, and tool geometry are taken as initial data. An animation of the movements of the workpiece 1 and the cutter 2 is presented in the graphics window. The cutter teeth are conventionally shown with circles located on the periphery of the outer diameter of the mill. The cutter rotates with frequency ω, and the workpiece moves with feed *F*. During the simulation, the results are displayed on the oscilloscope screen: line 3 rotation frequency ω, line 4 – cutting force component P_x acting along the *X* axis, line 5 elastic movement δF_z in the feed direction.

At a constant rotational speed of the cutter spindle (Fig. 3, *a*), the specified parameters of the cutting process lead to a loss of stability, as evidenced by an increase in the amplitude of oscillations of the component P_x of the cutting force to 2000N and elastic displacements to 0.08mm after 0.5 s of simulation. Using the SSSV option with the amplitude and the oscillation period indicated in window 6 of the interface provides a stable cutting process with the same parameters as in the previous experiment: the amplitude of the cutting force component and elastic displacements is reduced by 2 times (Fig. 3, *b*).



Stiffness X, N/mm 20000
Speed, m/min 100
Feed, mm/min 100
Depth, mm 5,2
Width, mm 50
Mass, kg 10
SSV ● Off ○ On
Diametr, mm 100
Number of teeth 8

a

Stiffness X, N/mm 20000
Speed, m/min 100
Feed, mm/min 100
Depth, mm 5,2
Width, mm 50
Mass, kg 10
SSV ○ Off ● On
Amplitude, % 10
Period, s 0,1
Diametr, mm 100
Number of teeth 8

b

Fig. 3. Simulation program interface: *a* – constant cutting speed, *b* – variable cutting speed

An increase in the oscillation period of the spindle speed control law leads to a decrease in method efficiency. It should be expected that a significant limitation to the practical use of such control will be the dynamics characteristics of the spindle drive of the machine, as well as the thermal loads of its engine.

After checking the adequacy of the developed mathematical model in terms of correspondence of the dynamic parameters of the elastic system (stiffness and mass), it can be used to determine the optimal amplitude and period of oscillations of the spindle speed control law.

Active Chatter Control

This type of technology for suppressing vibration during cutting is becoming increasingly important [16]. However, the proposed technologies are based on the modernization of equipment, the use of additional servo systems with actuators and special vibration sensors with appropriate software. Therefore, a technological solution based on the use of a shaping movement control channel already available on a CNC machine for active influence on the cutting process can be very promising.

The idea is to measure the frequency of vibrations that occur during cutting in the TMS and automatically determine the necessary shape of the compensating signal at the input of the drive of the shaping movement, which controls the movement along the path of shaping, for example, feed. It is proposed to determine the frequency and amplitude of the vibrations arising during cutting by analyzing the current of the corresponding feed drive of the CNC machine and act on its input with an additional harmonic signal. Moreover, the amplitude and phase of this signal are automatically adjusted using the coordinate descent algorithm in the loop of the automatic drive control system.

Fig. 4 shows a block diagram of the drive of a CNC machine, which has additional software designed to perform feed control in order to actively suppress vibration.

During the operation of the drive due to the cutting process, vibrations occur, accompanied by fluctuations in the load torque T_l on the motor. These fluctuations due to the presence of a non-self-locking ball screw transmission provoke a servo response in the form of a change in the motor current. Such a signal can be identified to determine the frequency ω_c of the main harmonic, which will correspond to the frequency of the main harmonic of oscillations in the TMS.

The software, which implements the coordinate-wise descent algorithm, compiles a feedback channel to minimize errors in the amplitude of the oscillations of the motor current. The coordinate descent is carried out according to two parameters of the signal $V_c = ACos\,(\omega_c t + \varphi)$, which is added to the main control voltage V_0 at the input of the drive: in phase φ and in amplitude A.

A change in the control signal causes a change in the motor supply voltage and the feed motion with the necessary amplitude and in antiphase to the vibration signal in the TMS, which leads to their suppression, at least in the frequency of the main harmonic.

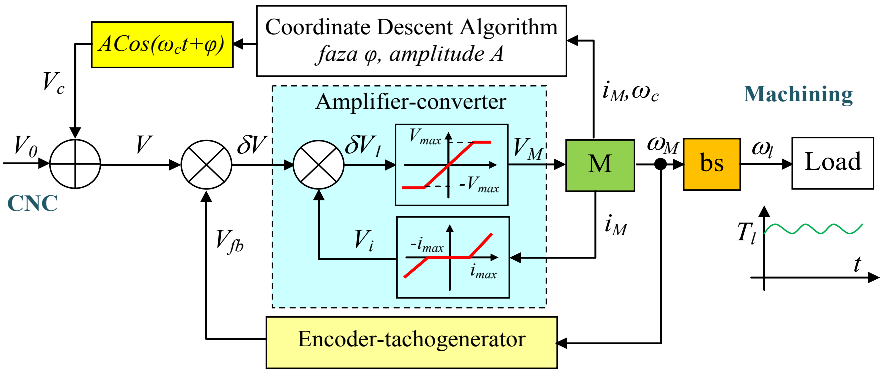


Fig. 4. Functional diagram of the drive

The Mathematical model. The standard complete CNC machine feed drive consists of an Amplifier Converter, an Motor, a tachogenerator encoder, a Ball Screw pair and a Load. The dynamic properties of the amplifier-converter are described by a first-order aperiodic link with a time constant T_{ac} and gain k_{ac}:

$$T_{ac}\frac{dV_M}{dt}+V_M=k_{ac}\delta V_1. \tag{8}$$

In addition, the amplifier-converter includes two non-linear links: with a characteristic of the type "saturation":

$$\begin{cases} if\ \delta V_1 k_{ac} > V_{\max},\ \ then\ V_M = V_{\max} \\ if\ \delta V_1 k_{ac} < -V_{\max},\ \ then\ V_M = -V_{\max} \end{cases} \tag{9}$$

and current limiting unit (Motor current) with the characteristic type "insensitivity":

$$\begin{cases} if\ i_M > i_{\max},\ then\ \delta V_1 = \delta V - k_i(i_M - i_{\max}) \\ if\ i_M < -i_{\max},\ then\ \delta V_1 = \delta V - k_i(i_M + i_{\max}) \end{cases}, \tag{10}$$

where i_M is the motor current (motor rotor current), k_i is the slope of the characteristic.

The mathematical model of a DC motor is represented by two equations – the Lagrange-Maxwell equations:

$$T_M = J_r \frac{d\omega_M}{dt} + k_f \omega_M + T_l, \tag{11}$$

where T_M is the engine torque, J_r is the moment of inertia of the engine rotor, ω_M is the angular velocity, k_f is the coefficient of the linearized dependence of friction on the engine speed, T_l is the load torque reduced to the motor shaft.

Another equation represents the voltage balance in the motor rotor circuit:

$$V_M = L\frac{di}{dt} + Ri + k_E \omega_M, \tag{12}$$

where L is the inductance of the motor armature circuit, R is the active resistance of the armature, k_E - counter-EMF coefficient (back electromotive force).

In addition, the motor torque is proportional to the rotor current:

$$T_M = k_M i_M. \tag{13}$$

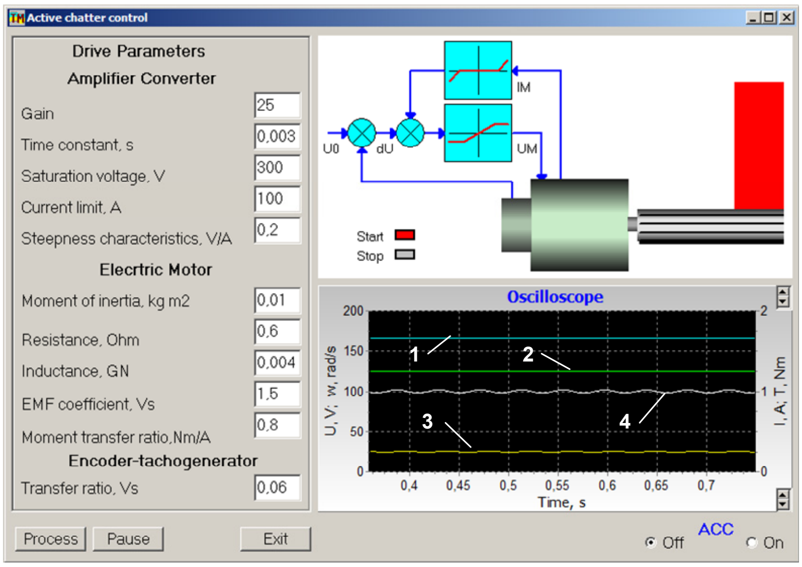
The torque of load resistance, reduced to the motor shaft, is determined by the equation:

$$T_l = J_l \frac{d\omega_M}{dt} k_{bc}^2 + k_{fl} \omega_M k_{bc}^2, \tag{14}$$

where J_l is the moment of inertia of the load; k_{fl} is the linearized coefficient of the dependence of the friction moment of the load on the rotation speed, k_{bc} is the gear ratio of the ball-screw pair.

The mathematical model of a complete electric drive thus compiled, together with the load (8) – (14), is a system of third-order nonlinear differential equations and can be studied only by numerical methods. For this, the mathematical model is represented in the form of a system of variable states and is solved by the standard fourth-order Runge-Kutt integration procedure.

The Simulation. To test the possibility of functioning of the proposed system, an application program was created, the interface of which in the initial state is shown in Fig. 5. We study a complete (standard) servo feed machine with CNC, the structural diagram of which is presented in the graphics window of the application program. The drive is represented by third-order differential equations with two nonlinearities: current limitation and voltage saturation. On the motor shaft conditionally shows the transmission of motion (ball-screw) and a red rectangle – the load from the cutting forces. Feedback is provided by an encoder (tachogenerator) signal. Thus, the drive is loop-closed and stabilizes the feed at a given level.



Drive Parameters

Amplifier Converter

Gain 25

Time constant, s 0,003

Saturation voltage, V 300

Current limit, A 100

Steepness characteristics, V/A 0,2

Elecrtric Motor

Moment of inertia, kg m2 0,01

Resistance, Ohm 0,6

Inductance, GN 0,004

EMF coefficient, Vs 1,5

Moment transfer ratio,Nm/A 0,8

Encoder-tachogenerator

Transfer ratio, Vs 0,06

ACC ● Off ○ On

Fig. 5. Simulation program interface

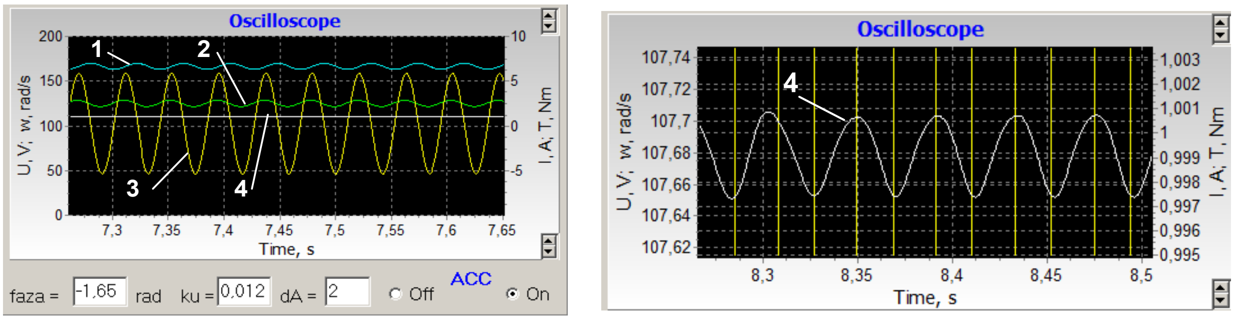
A harmonic signal is artificially introduced into the simulated system, simulating the occurrence of vibrations in the TOC during cutting as a change in the moment on the engine. The oscilloscope window displays the simulation results in time: line 1 – motor rotation speed (rad/s – left scale), line 2 – motor supply voltage (Volt – left scale), line 3 – current (Ampere – right scale), white line 4 – indignation (Nm – right scale).

The harmonic signal of the disturbing moment corresponding to the main harmonic of the vibrations arising in the system is represented by line 1 in the oscilloscope window and its amplitude is 0.02 Nm. Such load fluctuations provoke fluctuations in the motor current (line 3, fig.5) due to the closed loop of the servo drive along the loop of the tachogenerator encoder, just as it happens with real cutting.

After the Active Vibration Reduction (ACC – On) option is enabled, the motor current analysis module determines the frequency of the main harmonic of the excitation (25 Hz) and the search for the compensation signal parameters begins. The search is performed automatically in the coordinate descent algorithm in two coordinates: phase and amplitude. With each iteration, the direction of movement along the coordinates is determined by the criterion for reducing the amplitude of the oscillations of the motor current, and hence the amplitude of the moment from the vibrations of the system.

When ACC is turned on, additional windows open on the interface, in which the values of the phase and amplitude of the compensating signal in each iteration are displayed. The results of the system functioning are presented in the oscilloscope window, which are shown in Fig. 6, where the same notation is used as in Fig. 5.

It can be seen that the compensating signal caused changes in the voltage and current of the motor, and the speed fluctuations that occur in anti-phase to the exciting signal led to a decrease in its amplitude by a factor of 10 (0.002 Nm, see Fig. 6, *b*).



faza = -1,65 rad ku = 0,012 dA = 2 ○ Off ● On

a *b*

Fig. 6. The reaction of the system: a – control signals, b – an enlarged fragment of the moment of disturbance

The search was performed in 93 iterations, the motions in Fig. 7 are represented by lines 1 – by phase, line 2 – by the amplitude of the compensating signal, the change in the oscillation amplitude is represented by line 3. When modeling, it was found that the effectiveness of the system to chatter eliminate with a high frequency entirely depends on its speed reaction.

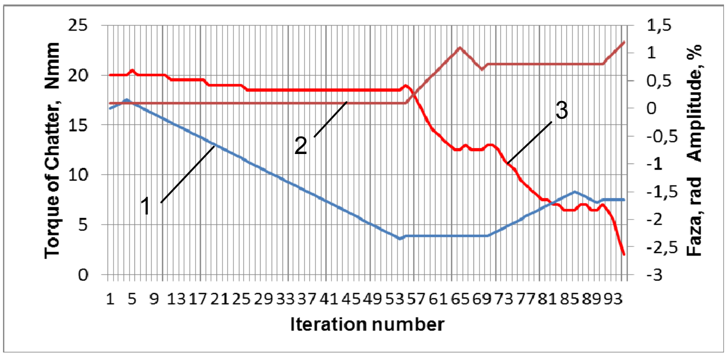


Fig. 7. Search paths

Conclusions

1. An analysis of current trends in the development of the science of metal cutting shows that the improvement of the control systems of CNC machines, computational modeling methods has led to the possibility of solving the most important problem of mechanical processing – eliminating fluctuations in TOC. It is shown that the determination of the optimal parameters of passive and active technologies is possible when modeling cutting processes, where it is

necessary to take into account the loop closing of the elastic technological system, its dynamic characteristics, geometric and force interaction of the tool with the workpiece, and machining along the trace.

2. The technology of vibration suppression during face milling by controlling the spindle speed according to the harmonic law is presented. The parameters of the control law must be determined when modeling the process on the created application program. The control efficiency is limited by the speed reaction of the spindle drive and its inertial characteristics.

3. A new system of active control of the drive of the forming movement of a CNC machine is proposed. The system uses a standard drive, in which an additional closed loop is integrated to automatically search for the amplitude and phase of the compensating control signal. The frequency of this signal is automatically determined by the frequency of the main harmonic of the feed motor current, and the amplitude and phase are searched by the coordinate descent algorithm according to the criterion of the minimum oscillation amplitude. The effectiveness of the control also depends on the speed reaction of the feed drive of the CNC machine.

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Технології усунення вібрації при різанні металів

Ю. В. Петраков

Проблематика. *Процес різання здійснюється в замкнутій пружній технологічній обробній системі і завжди супроводжується вібраціями. Виникаючі при різанні вібрації, в залежності від амплітуди, можуть зовсім незначно впливати на результат обробки, а можуть привести до катастрофічної втрати стійкості всього процесу. У будь-якому випадку все дослідники сходяться на тому, що саме вібрації є фактором, який, в кінцевому підсумку, визначає продуктивність процесу різання і якість обробленої поверхні.*

Мета дослідження. *Метою даного дослідження є розробка нових технологій вибору параметрів управління швидкістю різання для придушення вібрацій пасивними методами, а також управління приводом формоутворюючого руху верстату з ЧПК для придушення вібрацій активними методами.*

Методика реалізації. *Поставлена мета досягається шляхом створення нових технологій, спрямованих на дослідження динамічних процесів, що відбуваються в процесі різання. Відзначено, що математична модель процесу різання повинна будуватися з урахуванням замкнутості пружної технологічної обробної системи і функції запізнюючого аргументу, який* *представляє обробку по сліду. При дослідженні процесу різання враховуються чотири основні групи факторів, що впливають на його математичне уявлення, а для визначення діаграми стійкості використовуються три підходи: частотний аналіз, аналіз коренів характеристичного рівняння руху системи і чисельний метод. Найбільш результативним вважається чисельний метод з використанням амплітудно-частотних характеристик за відповідним критерієм стійкості.*

Результати дослідження. *Результати теоретичних досліджень використовуються на практиці у вигляді технологій пасивного і активного усунення вібрацій при різанні. Розроблено технологію усунення вібрацій при торцевому фрезеруванні при управлінні частотою обертання шпинделя за гармонічним законом. Представлена прикладна програма моделювання процесу для визначення параметрів закону управління. Для активного управління пропонується нова технологія, заснована на використанні приводу верстата з ЧПУ з додатковою замкнутою системою, що вносить в канал формоутворюючого руху гармонічний сигнал, амплітуда і фаза якого автоматично підналаштовуються з використанням алгоритму покоординатного спуску за критерієм мінімуму амплітуди струму двигуна.*

Висновки. *Технологія усунення вібрацій при торцевому фрезеруванні управлінням частотою обертання шпинделя за гармонічним законом обмежується швидкодією приводу шпинделя і його інерційними характеристиками. Система активного усунення вібрацій використовує стандартний сервопривод верстата з ЧПУ, в який вбудований додатковий замкнутий контур автоматичного пошуку амплітуди і фази компенсуючого сигналу управління.*

Ключові слова: *вібрації, процес різання, математична модель, стійкість процесу різання.*

Технологии подавления вибраций при резании металлов

Ю. В. Петраков

Проблематика. *Процесс резания осуществляется в замкнутой упругой технологической обрабатывающей системе и всегда сопровождается вибрациями. Возникающие при резании вибрации, в зависимости от амплитуды, могут совсем незначительно влиять на результат обработки, а могут привести к катастрофической потере устойчивости всего процесса. В любом случае все исследователи сходятся на том, что именно вибрации является фактором, который, в конечном итоге, определяет производительность процесса резания и качество обработанной поверхности.*

Цель исследования. *Целью настоящего исследования является разработка новых технологий выбора параметров управления скоростью резания для подавления вибраций пассивными методами, а также управления приводом формообразующего движения для подавления вибраций активными методами.*

Методика реализации. *Поставленная цель достигается путем создания новых технологий, направленных на исследование динамических процессов, происходящих в процессе резания. Отмечено, что математическая модель процесса резания должна строиться с учетом замкнутости упругой технологической обрабатывающей системы и функции запаздывающего аргумента, который представляет обработку по следу. При исследовании процесса резания учитываются четыре основные группы факторов, влияющих на его математическое представление, а для определения диаграммы устойчивости используются три подхода: частотный анализ, анализ корней характеристического уравнения движения системы и численный метод. Наиболее результативным считается численный метод с использованием амплитудно-частотных характеристик по соответствующему критерию устойчивости.*

Результаты исследования. *Результаты теоретических исследований используются на практике в виде технологий пассивного и активного подавления вибраций при резании. Разработана технология подавления вибраций при торцевом фрезеровании при управлении частотой вращения шпинделя по гармоническому закону. Представлена прикладная программа моделирования процесса для определения параметров закона управления. Для активного управления предлагается новая технология, основанная на использовании привода станка с ЧПУ с дополнительной замкнутой системой, вносящей в канал формообразующего движения гармоничный сигнал, амплитуда и фаза которого автоматически подстраиваются с использованием алгоритма покоординатного спуска по критерию минимума амплитуды тока двигателя.*

Выводы. *Технология подавления вибраций при торцевом фрезеровании управлением частотой вращения шпинделя по гармоническому закону ограничивается быстродействием привода шпинделя и его инерционными характеристиками. Система активного подавления вибраций использует стандартный сервопривод станка с ЧПУ, в который встроен дополнительный замкнутый контур автоматического поиска амплитуды и фазы компенсирующего сигнала управления.*

Ключевые слова: *вибрации, процесс резания, математическая модель, устойчивость процесса резания.*

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