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A CUTTING TOOL VIBRATION MECHANISM IN METALWORKING TECHNOLOGICAL SYSTEM

Представлено механізм виникнення вібрацій ріжучого інструменту на верстатах з ЧПК на основі існуючих фізичних уявлень про примусові, вільні і самозбудні коливання в технологічної системі різання, яка складається з взаємодіючих підсистем інструменту і заготовки.

Представлены возникновения вибраций режущего инструмента на станках с ЧПУ на основе существующих физических представлений о принудительных, свободные и самовозбуждающиеся колебания в технологической системе резания, которая состоит из взаимодействующих подсистем инструмента и заготовки.

A cutting tool vibration mechanism on CNC machines characteristic based on existing physical representations of the forced, free and self-excited oscillations occurrence in the technological cutting system consisting of interacting tool and workpiece subsystems is given.

Statement of the problem. To ensure reliable operation of advanced high-speed CNC machine a control system should provide not only precision programmable tool displacement relative to a workpiece but also diagnosis of the cutting technological system. The weakest link in the system is the cutting tool (CT) life which should be sufficient for reliable operation of the CNC machine for the desired cutting time.

The industrial cutting systems vibration problem is generally known, starting with the F.W. Taylor's works. Domestic researchers in this field, for example, A.I. Kashyrin, V.I. Dikushin, V.A. Kudinov and many others are also known. All of them paid much attention in their works to the physical principles of vibration when cutting hard and easily workable materials because an insight to the mechanism of vibration allows identifying appropriate ways to deal with this phenomenon.

Modern construction materials (stainless and heat resistant steels as well as alloys, titanium and its alloys, etc.) have high performance, but also they have a low machinability, which leads (because of the unpredictable influence on the process of cutting force and temperature factors) to low CT life. On the other hand for easily workable workpiece materials such as aluminum and its alloys a high cutting speed is currently using in high speed machining with increased feed and depth of cut. In both cases (i.e. hard and easily workable materials) a cutting vibration problem refers to the number of actual one in mechanical engineering since the appearance of vibration is usually associated with a CT life as well as a premature failure of the machine spindle unit. There are some exceptions connected with the controlled vibrations which improve the CT work such as these in vibrodrilling.

It is well known a necessity to increase the metal removal rate as well as machining production on the CNC machines. In order, however, to do this the so-called "chatter" arises and does this phenomenon a far more significant concern. That is why a manufacturer faces not only features of a machine and tool but also the dynamic characteristic of the spindle and work subsystems.

To avoid as the chatter as the other significant dynamic oscillations the most promising for use on modern CNC machines is small vibration sensors, such as AP2019 type. These sensors can be embedded in the various directions of the machine coordinate system. However, so far no reliable methods the CT state technological vibrodiagnostics which can be implemented on the basis of these sensors and available CNC system computational resources.

The purpose of this research is to develop the scientific premises for a CT state vibrodiagnostics automated system based on a USB type modular system NI CompactDAQ followed by programming the diagnostic algorithm (without any additional hardware) in modern CNC system having available computing resources.

Selection of unsolved parts of the problem. There are known technological methods of diagnosis by different estimating criteria for the CT state needed to solve a technological management task. They vary depending on the nature of selected physical parameters i.e. sources of information about the CT state: power, torque, cutting temperature, cutting vibrations (displacement, velocity, and acceleration), acoustic emission (sonic and ultrasonic), the parameters of quality of processing parts etc. [1].

In the physical dynamics there are two kinds of vibration: forced vibration and self-excited one. Forced vibrations are generated by the action of a periodic force, for example, due to an imbalance of the rotating spindle or CT edges interrupted operation (e.g. drill or mill edges). In this case, the vibration source (a spindle or CT edges) vibrates interacting with the technological system elements. As a result, the vibration frequency spectrum consist of the spindle and associated with it structural elements speed components as well as the rate of introduction into the machining material of the cutting edges. In order to understand the vibration self-excitation mechanism it is necessary to consider the nature of free vibrations in cutting [2] which arise, for example, when the cutting forces suddenly released, i.e. when the next CT edge is exited from the contact area. In this case sudden elimination of the impact of cutting forces on the machine takes place. These vibrations are characterized by their natural or own frequency which is known to be determined by the elastic system stiffness and its reduced mass [2].

Main material with a substantiation of results. When a CT tooth enters into the machining material, the CT subsystem "spindle - tool holder - tool" will be deformed by the cutting forces. When these forces are released by the tooth exiting the material, the CT subsystem will vibrate with its own natural frequency. It is assumed that the workpiece subsystem "table - device - storage" stiffness is more than CT subsystem one and can be ignored. The vibration mentioned results a small waviness on the workpiece surface. If the following that is after the first tool tooth impact does not match the natural frequency of the CT subsystem, the chip thickness increases as well as the cutting force. This in turn causes a notable deformation of the system, which leads to a larger oscillation amplitude. The most disadvantageous condition is when the current vibration phase angle will be equal to 180° with respect to the surface waviness previously obtained. Thus, the self-excited vibration in the cutting zone (in domestic literature – auto oscillations) is the result of the unpredictable interaction of several factors. For example, it is so when the vibration phase from the CT edges is late 180° (π radians) from the previous track phase and a cutting power is sufficient to overcome the damping of oscillations. Such vibrations are called self-excited or chatter (a kind of rasp). Under these conditions the CT subsystem vibrates at its natural frequency (without an externally applied driving force), the cutting force is increased significantly, negatively affects the machining accuracy, CT life, and machine spindle unit longevity. This implies parametric and kinematic nature of self-excited oscillations in a system with positive feedback, which has a reserve of potential energy (spindle motor) and the way to deal with vibrations: either the gain coefficient to reduce or the oscillation phase to output of the positive feedback (180°) condition, or both mentioned simultaneously.

It is possible to avoid auto-oscillations in cutting as well as the chatter if the CT tooth exposure frequency is consistent with the natural frequency of the CT subsystem "spindle - tool holder - tool ". In other words, it is so when the surface waviness and the cutting vibrations are in phase (0°). By the way, the waviness may not be clearly visible, but appears to change the physical and mechanical properties of the surface layer. At this spindle speed, the chip thickness remains constant, cutting goes smoothly (quietly) and the cutter may go to a great depth without causing defects. This phenomenon is called "sweet spot» [2].

There are two basic approaches to determine spindle speed at which there is no vibration such as chatter or the like [2]. According to the first one the CT subsystem own frequency is found with the aid of using a vibrosensor (accelerometer) and an impact hammer. Then the system transfer function is determined for analytical forecasting oscillations by calculating the sweet spots. It is necessary for this to have a mathematical model of the cutting dynamic system. The second approach for determining the sweet spots is to perform cutting tests. This approach allows obtaining more accurate information but requires a large number of experiments that are carried out with different combinations of spindle speed and depth of cut. Experimentally determined cutting modes with the sweet spots is then programmed to provide a stable high-performance machine work for a certain combination of machine tool as well as the CT and workpiece subsystems. The best is to use both of these techniques to obtain more accurate information about the vibrations in the machine elastic system.

Described mechanism of self-excited vibrations during cutting may have features depending on the cutting process kind in mechanical engineering technology. For example, a processing on the lathe by a single turning cutter differs from a multiple edge milling by monolithic CT or assembled one. Vibration during drilling (core-drilling, reaming, counter-sinking etc.) depends on the torsional corresponding axial tools oscillations, especially for small diameter drills which change their length under the machining.

Technical systems modeling methods in the cutting dynamics as well as in the cutting thermophysics are divided into two broad classes depending on the adopted methodological concept: distributed or lumped system. In the first case (distributed system) processes are described by partial differential equations while in the second one (lumped systems) ordinary differential equations are used. The cutting dynamics usually uses the lumped system concept in which various assumptions are made and simplifying techniques are used (e.g., reduced and generalized parameters) to substitute a real distributed system by corresponding lumped system having one or more degree of freedom.

In this connection a vibration mechanism in the dynamic lumped systems is based on the equation of equilibrium of force components applied to some point belonging to a mechanical arrangement of any kind. One of the kinds is CNC machine tool, e.g. a machining center 500V/5.

General scientific problem of the mechanical oscillations in engineering systems with lumped parameters more fully discussed in the study of linear and torsional vibrations [3]. Analysis of this work can reveal some features of the elastic cutting system vibration mechanism.

Initially this work examines free (or natural) harmonic oscillations when the oscillations of a vibrating weight which can be substituted later by a reduced (concentrated) mass is maintained only by an spring force that is equal to the product of stiffness k (spring constant according to Timoshenko S.P.) and elastic displacement x, i.e. kx, where k is the force which produces a unit displacement. Equilibrium equation or differential equation of motion for an ideal mass-spring system which can be derived on the basis of Newton's principle is

$$\ddot{x} + p^2 x = 0, \tag{1}$$

where $p = \sqrt{\frac{k}{m}}$ or $p = \sqrt{\frac{g}{\delta_{st}}}$ is introduced notation for the natural or own frequency of the spring mechanical system; m is the reduced to a point mass, kg; g is the gravitational acceleration, m^2/s; $\delta_{st} = \frac{mg}{k}$ is the static deflection of an equivalent spring in an ideal mass-spring system (an elastic system).

Introducing the equation (1) here as well as the following equations below we assume that further it will be a kind of lumped (concentrated) elastic system which will be equivalent to the real distributed elastic system for any technical arrangement including a machine tool. The equation (1) is a homogeneous (without right part) linear differential equation of second order with constant coefficients and satisfied if $x = C_1 \cos pt$ and $x = C_2 \sin pt$, where C_1 and C_2 are arbitrary constants (constants of integration) a number of which is equal to the order of the differential equation (1). The general solution of this equation is the sum of the mentioned above components each of them is a particular solution of equation (1), that is

$$x = C_1 \cos pt + C_2 \sin pt \tag{2}$$

Taking into account that at the initial moment ($t = 0$) the vibrating mass has a displacement x_0 (from its equilibrium position) and moves at this moment with the velocity $\dot{x}_0$ it can be obtained $C_1 = x_0$ and $C_2 = \dot{x}_0 / p$. Therefore the equation (2) takes the form

$$x = x_0 \cos pt + \frac{\dot{x}_0}{p} \sin pt \tag{3}$$

It is important to observe that each vibration phenomenon can be represented both in vibrational and rotational forms. The equation (third) represents a sum of the two vector projections on the x axis. The later can be represented as

$$\frac{\dot{x}_0}{p}\sin pt = \frac{\dot{x}_0}{p}\cos\left[-\left(\frac{\pi}{2} - pt\right)\right] = \frac{\dot{x}_0}{p}\cos\left(\frac{\pi}{2} - pt\right).$$

Therefore

$$x = x_0 \cos pt + \frac{\dot{x}_0}{p}\cos\left(\frac{\pi}{2} - pt\right) \tag{4}$$

It is seen that even without both a viscous damping and a disturbing force the vibration consist of two parts; the first is proportional to $\cos pt$ and depends on the initial displacement x_0 and the second is proportional to $\cos\left(\frac{\pi}{2} - pt\right)$ and depends on the initial velocity $\dot{x}_0$ in the form of $\dot{x}_0 / p$. The two rotating vectors can be substituted by resulting one which rotates with the same angular velocity p around a fixed point. This velocity in contrast to the former is circular velocity p of vibration. If one resulting vector remains instead of the two rotating ones and it is equal to the geometrical sum of the previous vectors then the same solution like equation (third) or (forth) has the following form

$$x = \sqrt{x_0^2 + \frac{\dot{x}_0^2}{p^2}}\cos\left(pt - arctg\frac{\dot{x}_0}{x_0 p}\right) \tag{5}$$

Thus, the sum of projections of the two rotating vectors on the *x* axis is equal to one resulting vector projection. The angle $arctg\frac{\dot{x}_0}{x_0 p}$ is equal to one between the vector with amplitude x_0 and the resulting vector with amplitude $\sqrt{x_0^2 + \frac{\dot{x}_0^2}{p^2}}$ and is the phase shift between these vectors. It is seen that if at the initial moment ($t = 0$) the initial velocity $\dot{x}_0$ of the vibrating body is absent, i.e. $\dot{x}_0 = 0$, then the only one vector rotates with an amplitude x_0.

Then a disturbing force $Q\sin\omega t$ is applied to the elastic system that is "a harmonic force function" added to the system. This function has the amplitude Q (in Newtons) and circular frequency ω (radians per second) which has been imposed on the elastic system. In such a case instead of the equations (1) and (2) we obtain correspondingly the equilibrium equation and the general solution of the equation, i.e.

$$\ddot{x} + p^2 x = q\sin\omega t. \tag{6}$$

$$x = C_1\cos pt + C_2\sin pt + \frac{q\sin\omega t}{p^2 - \omega^2} \tag{7}$$

Taking into attention that $\frac{q}{p^2} = \frac{Q}{k}$, the later part of the equation (7) *takes the form*

$$x = \left(\frac{Q}{k}\sin\omega t\right)\left(\frac{1}{1 - \omega^2 / p^2}\right), \tag{8}$$

Where x is the part of total amount of displacement during vibration of a reduced mass, m; $p = \sqrt{\frac{k}{m}}$ is the natural frequency of the mechanical system, rad/ s;

In expression (8) the second multiplier taken in modulus is called **the magnification factor** which is given by the multiplier

$$\beta = \left|\frac{1}{1 - \omega^2 / p^2}\right|$$

It can be seen that the magnitude of β depends on the ratio ω / p which is obtained by dividing the frequency of the disturbing force ω by the frequency of free vibration p.

A positive sign for β, that is $\omega < p$, indicates coincidence driving directions of the reduced mass and the driving force (phase match) while a negative one, that is $\omega < p$, on the contrary means that these directions are opposite. In the first case, a positive feedback loop is created, in which the disturbing force is "swinging" an oscillation, in the second – a negative feedback when the disturbing force is "extinguishing" the oscillation. With increasing frequency ω the frequency ratio ω^2 / p^2 increases, the magnification factor β decreases sharply, asymptotically approaching zero. For example, if we assume $\omega / p = 3$ then β = 0,125. At the same time as the ω decreases, i.e. ω / p tends to zero, the factor β asymptotically tends to the unit value. Finally, when $\omega = p$ a resonance occurs in which the magnification factor β is increased sharply, i.e. $\beta \to \infty$.

Finally, to match the real actual mechanical system it is necessary to introduce the so-called equivalent viscous damping [3]. The presence of the damping forces (friction force between the dry or lubricated sliding surfaces, environmental resistance, internal friction in the elastic zone of material, etc.) in real technical systems *partly changes* described mechanism of vibration, bringing it closer to the real mechanism that takes place in practice. Damping forces cause dissipation (or loss) of energy and are complex in terms of their mathematical description. Therefore, in the classical theory of oscillations it is accepted to replace any resistance forces by the equivalent viscous damping in which the damping force $c\dot{x}$ is proportional to the velocity $\dot{x}$ of the reduced mass displacement where c is a coefficient of viscous damping that is equal to the value of the damping force per unit velocity.

Forced oscillations equation for the reduced mass with one degree of freedom (motion along a single coordinate) is now a major equation in the theoretical study of vibrations in the cutting technological sysnem. This equation consists of the elastic and damping forces which are equal to kx and $c\dot{x}$ respectively. The equation and its solution have the following form [3]

$$m\ddot{x} = -kx - c\dot{x} + Q\sin\omega t, \tag{9}$$

$$x = e^{-nt}\left(C_1 \cos p_d t + C_2 \sin p_d t\right) + M\cos\omega t + N\sin\omega t, \tag{10}$$

where $Q\sin\omega t$ is the harmonic force function that is similar to the above mentioned expression $Q\sin\omega t$, H; $n = c/2m$ – the analog of circular frequency caused by the *equivalent viscous damping*, rad / s; $p_d = \sqrt{p^2 - n^2}$ – the angular frequency of damped oscillations when the damping takes place, rad / s ; C_1, C_2 – integration constants, which is determined by the initial conditions for free oscillations; M, N – integration constants that is defined by substituting of the equation (10), to the exclusion of free oscillations, in the original equation (9).

The equation (10) corresponds to the case of "precritical" damping, in which the degree of damping does not prevent the periodic oscillations. Otherwise, free movement is not periodic one because the viscous resistance is so large ($n > p$), that the reduced mass does not oscillate, but only returns to its equilibrium position. This is so-called "overdamping" which leads to aperiodic motion of the lumped mass. The critical value of the damping coefficient c_{cr} is found from the condition $n = p$ and equal to $c_{cr} = 2\sqrt{km}$.

To determine C_1 and C_2 it is assumed that in the initial moment of time, that is at $t = 0$, the reduced mass is displaced during its oscillation from the equilibrium position, for example, by an amount of x_0 and has at this time point the initial velocity $\dot{x}_0$. Then it is obtained

$$C_1 = x_0; \qquad C_2 = \frac{\dot{x}_0 + nx_0}{p_d}. \tag{11}$$

After *damped free vibrations* cease to exert influence on the process, in accordance with equation (10) the forced oscillations of the following form is set

$$x = M\cos\omega t + N\sin\omega t. \tag{12}$$

It is seen from the equation (12) that the forced oscillation consists of the two terms. One of them is proportional to $\cos\omega t$ with the M constant and the other is proportional to $\sin\omega t$ with the N constant.

Substituting (12) into the original equation (9) follows [3]

$$M = \frac{q\,(p^2 - \omega^2)}{(p^2 - \omega^2) + 4n^2\omega^2} \quad \text{and} \quad N = \frac{q\,(2n\omega)}{(p^2 - \omega^2) + 4n^2\omega^2},$$

wherein $q = Q / m$ is the acceleration, that is the inertia force component, which can be controlled by the vibrosensor, m/s^2.

The above analysis of the physical essence of mechanical vibrations in an elastic mechanical system such as any arrangement including a machine tool allows you to reframe the principles of equivalence and generalized parameters for the development of embedded system of technological diagnostics.

Conclusions

1. For over a long historical period (more than a hundred years) in the theoretical study of vibrations into cutting technological system the lumped system concept is used, while the real cutting system is the distributed one.

2. Even without both a viscous damping and a disturbing force the vibration consist of two parts; the first is proportional to $\cos pt$ and depends on the initial displacement x_0 and the second is proportional to $\cos\left(\frac{\pi}{2} - pt\right)$ and depends on the initial velocity $\dot{x}_0$ in the form of $\dot{x}_0 / p$.

3. In a system with a natural vibration frequency p presence of damping leads to reduction of the natural frequency to a level of $p_d < p$, that is viscous damping has got a high pass filter property.

4. In a system with "precritical" damping a reduced mass movement consists of two components: a damped free (it undergoes transient and disappears) and forced periodic (it is operating indefinitely).

5. The emergence of “chatter” is a manifestation of mechanical resonance in the *elastic system with damping*, but differs from it by the mechanism of the influence of the previous traces of machining, for example, such as a pre-formed waviness.

6. For the occurrence of oscillations in a mechanical elastic system of the machine must create two conditions: antiphase, i.e. 180°, to the previous traces and the required magnification factor that is sufficient for the positive feedback occurrence.

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CUTTING TOOL VIBRODIAGNOSTICS SYSTEM ON THE MACHINING CENTER

Представлено характеристику способу технологічної вібродіагностики ріжучого інструмента на верстатах з ЧПК, та результати експериментальних досліджень вібродіагностичної системи на основі вимірювального комплексу NI CompactDAQ при свердлінні отворів малого діаметра (до 5 мм) на обробному центрі мод. 500V/5.

Представлена характеристика способа технологической вибродиагностики режущего инструмента на станках с ЧПУ, и результаты экспериментальных исследований вибродиагностического системы на основе измерительного комплекса NI CompactDAQ при сверлении отверстий малого диаметра (до 5 мм) на обрабатывающем центре мод. 500V / 5.

Characteristic of the cutting tools technological vibrodiagnostics way on CNC machines and the results of experimental studies of the vibrodiagnostic system in drilling small holes (up to 5 mm in diameter) on the basis of measuring complex NI CompactDAQ built in the machining center 500V/5 type *are given.*

Introduction. This work is a continuation of experimental studies [1] for drilling small holes on the machining center 500V/5 type with SIEMENS SINUMERIC 840D CNC (nominal and maximum spindle