Найменшу похибку мають перетворювачі при орієнтації вимірювальної хорди під кутом 0 *рад*, яка в межах 15-17 перерізів досягає мінімуму зі значеннями близькими до 0.В діапазоні 20-21 перерізів похибки приладу за різних орієнтацій вимірювальної хорди співпадають і наближаються до (0,5 – 0,6)%, за виключенням орієнтації напряму посилання УЗ сигналів $\theta = 3\pi / 4$ *рад*.

Висновки

Для ультразвукового перетворювача однопроменевої конфігурації результат вимірювання хордової швидкості, що визначає витрату, залежить від величини викривлення осьової симетрії профілю і зменшується з віддаленням контрольного перетину від місцевого опору. Величина похибки залежить від просторової орієнтації приладу відносно вертикальної вісі на технологічній мережі. Отримані результати дають чітку картину місць локального розміщення приладів на технологічній мережі за умов максимальної точності і мінімальної дії на вимірюване середовище. Це дозволяє можливість ефективно застосовувати перетворювачі без огляду на вимоги обов’язкового забезпечення прямих ділянок до і після приладів.

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A VIBRATION SENSOR AND A CUTTING SYSTEM MATHEMATICAL DESCRIPTION

From a unified point of Engineering Physics an attempt was made to give a mathematical description of a vibration sensor - accelerometer and an elastic cutting system in which the sensor is the source of the information signal.

Keywords: *engineering physics, vibration sensor, an information signal.*

Предпринята попытка с единых позиций инженерной физики дать математическое описание вибрационного датчика – акселерометра и упругой технологической системы резания, в которой этот датчик является источником информационного сигнала.

Ключевые слова: *инженерная физика, вибрационный датчик, информационный сигнал.*

Зроблена спроба з єдиних позицій інженерної фізики дати математичний опис вібраційного датчика - акселерометра і пружної технологічної системи різання, в якій цей датчик є джерелом інформаційного сигналу.

Ключові слова: *інженерна фізика, вібраційний датчик, інформаційний сигнал.*

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 block. 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 (figure 1).

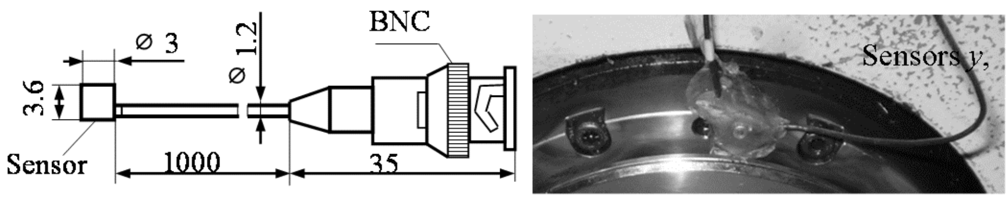

Figure 1 – AP2019 type accelerometer (left) and its mounting on the machine 500/V5 spindle block (right) along the machine *y* and *z* axes (*x* axis is not shown).

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. Besides, there are no theoretical studies directing to the vibration sensor dynamic error evaluation.

The purpose of this research is from a unified point of Engineering Physics to give a mathematical description of a vibration sensor - accelerometer and an elastic cutting system in which the sensor is the source of the information signal. The research is 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 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.

Main material with a substantiation of results. There are two main approaches in engineering physics modeling: "lumped"' physical system, such as configurations of masses, springs, and "dashpots" and the "distributed" one. The term "lumped"' comes from electrical engineering, and refers to lumped-parameter analysis, as opposed to distributed-parameter analysis. In general, a lumped-parameter approach is appropriate when the physical object has dimensions that are small relative to the wavelength of vibration. However a lumped-parameter approach is the most in cutting dynamics although a cutting system structure is a distributed one. There is the only one realm in which a lumped-parameter approach is appropriate well and this is the vibration sensor modeling. Firstly, the sensor is rectilinear one; secondly it has configurations of masses, springs, and "dashpots" which are selected by the designer especially for the measurement problem. Depending on the circumstances a displacement or a force is measured only. There is no another alternative in measurement technique [1]. That is why the question now is how to measure a displacement and its derivatives (velocity or acceleration), on the one hand, and the force, on the other hand. In the both cases a dynamic sensor model is used. This is one side of problem.

Under dynamic conditions, a sensor is described by its dynamic transfer function and may be characterized with a time-dependent characteristic which is called a dynamic characteristic. It means that the sensor does not respond instantly, it may indicate values of stimuli which are somewhat different from the real; i.e., the sensor responds with a dynamic error. The latter is a difference between static and dynamic errors. When a sensor is incorporated inside the control system which has its own dynamic characteristics, the combination may cause a delay in the appearance of a true value of a stimulus or even oscillations of the output value [2].

Mathematically, a vibration sensor can be described by a differential equation whose order depends on the sensor's nature and design. There are three types of the equation depends on the relationship between the input $s(t)$ and the output $S(t)$: a zero-order, a first-order, and a second-order response.

In a control system theory the input-output relationship is described through a constant –coefficients linear equation. That is why the sensor's dynamic characteristics can be analyzed by evaluated such an equation with a zero-order, a first-order, or a second-order differential equation. The latter takes place for vibration sensors in measuring forces, displacement and their derivatives (velocity, acceleration).

A second-order differential equation describes a sensor that incorporates at least two energy storage components (a mass and a spring). The relationship between the input $s(t)$ and output $S(t)$ is represented by the equation

$$b_2 \frac{d^2 S(t)}{dt^2} + b_1 \frac{d\,S(t)}{dt} + b_0 S(t) = s(t), \tag{1}$$

where b_2, b_1, and b_0 are constant coefficients (constants).

Mathematical modeling of a sensor is a power tool in assessing its performance [2]. The modeling may address two issues: static and dynamic. The dynamic models may have several independent variables; however, one of them must be time. The resulting model is referred to as a lumped parameter model. As it was mentioned above the vibration physical laws may be interpreted most completely on the example of the vibration sensor mathematical description (see the name of the paper). This allows exhausting the vibration lumped-parameter modeling for all possible combinations of masses, springs, and "dashpots" as well as their individual quantities' measures. In other words, for the vibration analysis, a sensor is separated into simple lumped parameter elements and each of them is considered separately.

An example of a second-order sensor is an accelerometer that includes a mass and a spring, which are energy storing elements, and a "dashpot" (damper), which is energy dissipating one (figure 1).

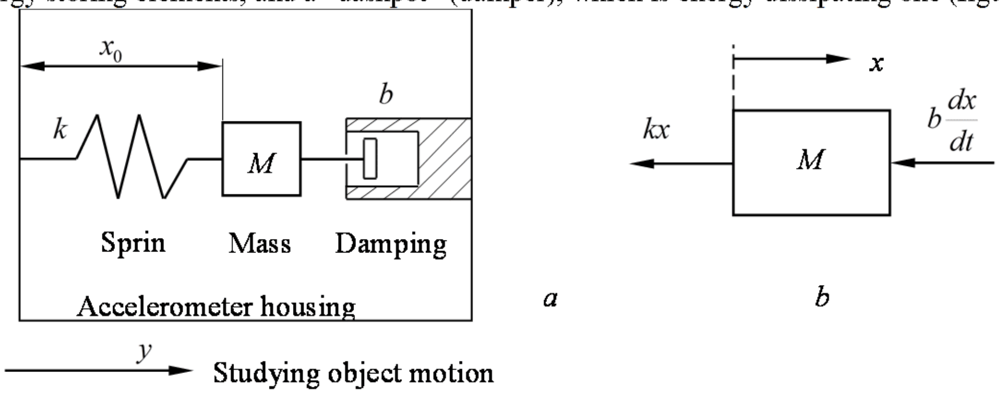

Figure 2 – Mechanical model of an accelerometer (*a*) and a free-body diagram of a mass (*b*) [2].

A monoaxial accelerometer consists of an inertia element M whose movement may be transformed into an electrical signal, for instance, on the basis of piezoelectric transformer. The mass M is supported by a spring with stiffness k and the mass movement is damped by a damping element with a coefficient b. Mass may be displaced with respect to the accelerometer housing only in horizontal direction x (figure 2, *b*). During operation, the accelerometer housing is subjected to acceleration $d^2 y / dt^2$, and the output signal is proportional to the mass M deflection of x_0. The system has the only one degree of freedom because the accelerometer mass is constrained to linear motion. Applying Newton's second law of motion gives

$$M\frac{d^2x}{dt^2}+b\frac{dx}{dt}+kx=M\frac{d^2y}{dt^2} \qquad (2)$$

The each term of the equation is the force and comparing the expressions (1) and (2) we obtain the following conclusion: $S(t)=x$; $s(t)=d^2y/dt^2$; $b_2=1$; $b_1=b/M$; $b_0=k/M$. The differential equation (2) is of a second order, which means that the sensor output signal may have a vibrating form. In order to get a desirable response, it is necessary selecting an appropriate coefficient b, and then the output signal may be brought to a critically damped state. Thus, once the equations describing the elements have been formulated, individual elements can be recombined to yield the mathematical model of the original sensor [2].

Using the Laplace transformation for the equation (2) we can get

$$Mp^2X(p)+bpX(p)+kX(p)=MA(p) \qquad (3)$$

where $X(p)$ and $A(p)$ are the Laplace transforms of $x(t)$ and d^2y/dt^2, respectively. Soling for $X(p)$, we obtain

$$X(p)=-\frac{MA(p)}{Mp^2+bp+k}. \qquad (4)$$

Introducing a conventional variable $\omega_0=\sqrt{k/M}$ and $2\xi\omega_0=b/M$, the equation (4) can be expressed as

$$X(p)=-\frac{A(p)}{p^2+2\xi\omega_0 p+\omega_0^2}, \qquad (5)$$

where ω_0 is accelerometer's angular natural frequency and ξ is the normalized damping coefficient.

In its turn, in terms of the inverse Laplace transform operator, we obtain

$$x(t)=L^{-1}\left[G(p)A(p)\right] \qquad (6)$$

Here

$$G(p)=-\frac{1}{p^2+2\xi\omega_0 p+\omega_0^2} \qquad (7)$$

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

$$\ddot{x}+p^2x=0, \qquad (8)$$

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 (8) 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 (8) 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 (8). 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

$$x=C_1\cos pt+C_2\sin pt \qquad (9)$$

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 (9) takes the form

$$x = x_0 \cos pt + \frac{\dot{x}_0}{p} \sin pt \tag{10}$$

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

$$x = x_0 \cos pt + \frac{\dot{x}_0}{p} \cos\left(\frac{\pi}{2} - pt\right) \tag{11}$$

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

$$x = \sqrt{x_0^2 + \frac{\dot{x}_0^2}{p^2}} \cos\left(pt - arctg \frac{\dot{x}_0}{x_0 p}\right) \tag{12}$$

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 (8) and (9) we obtain correspondingly the equilibrium equation and the general solution of the equation, i.e.

$$\ddot{x} + p^2 x = q \sin \omega t. \tag{13}$$

$$x = C_1 \cos pt + C_2 \sin pt + \frac{q \sin \omega t}{p^2 - \omega^2} \tag{14}$$

Taking into attention that $\frac{q}{p^2} = \frac{Q}{k}$, the later part of the equation (14) takes the form

$$x = \left(\frac{Q}{k} \sin \omega t\right)\left(\frac{1}{1 - \omega^2 / p^2}\right), \tag{15}$$

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 $\beta =$ 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 system. 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]

$$m\ddot{x} = -kx - c\dot{x} + Q\sin\omega t, \tag{16}$$

$$x = e^{-nt}\left(C_1 \cos p_d t + C_2 \sin p_d t\right) + M\cos\omega t + N\sin\omega t, \tag{17}$$

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 (17), to the exclusion of free oscillations, in the original equation (16).

The equation (17) 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

$$C_1 = x_0; \qquad C_2 = \frac{\dot{x}_0 + nx_0}{p_d}. \tag{18}$$

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

$$x = M\cos\omega t + N\sin\omega t. \tag{19}$$

It is seen from the equation (19) 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 (19) into the original equation (16) 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. The lumped approach in engineering physics is more suitable for a vibration sensor mathematical description then for machine elastic system 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.

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ЗАКОНОМЕРНОСТИ ОБРАЗОВАНИЯ ПРИЖОГОВ ЗАКАЛКИ ПРИ ШЛИФОВАНИИ ПОДШИПНИКОВЫХ СТАЛЕЙ ЗА СЧЕТ ДИФФУЗИОННОГО МЕХАНИЗМА МАРТЕНСИТ-ПЕРЛИТ-АУСТЕНИТ

У статті наведено результати досліджень законів утворення структури аустеніту в поверхневому шарі деталі при шліфуванні Це явище одержало назву шліфувальні припаленя. Структура цих припалень являє собою аустеніт високої твердості. Така структура поверхневого шару значно знижує довговічність і експлуатаційний ресурс деталі. Показано, що це перетворення може відбуватися за рахунок трьох різних механізмів - дифузійного, бездифузійного при нормальному тиску і бездифузійного при високих тисках. При утворенні аустеніту по дифузійному механізму при нормальному тиску інструментальна точка A_{C1} підвищується. Це підвищення тим більше, чим вище швидкість нагрівання. Одержаний аустеніт піддається інтенсивному наклепу, після чого охолоджується зі швидкостями вище критичних швидкостей гарту. Наведено, що залишковий аустеніт має перекручену наклепом кристалічну гратку. Дифузійне перетворення займає досить значне місце Швидкісна відпустка мартенситу відбувається при швидкостях нагріву близько сотні тисяч градусів в секунду. Таким чином, при дуже високих швидкостях нагріву можливе утворення перліту, який при підвищенні температури понад A_{C3} буде дифузійного перетворюватися на аустеніт. Чим вище температура, тим швидше завершується дифузійний процес перерозподілу вуглецю в аустеніт.

Ключові слова: *Аустеніт, поверхневий шар, дифузійний механізм утворення.*

В статье приведены результаты исследований законов образования структуры аустенита в поверхностном слое детали при шлифовании Это явление получило название шлифовочные прижоги. Структура этих прижогов представляет собой аустенит аномально высокой твердости. Такая